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# KENT'S
# MECHANICAL ENGINEERS' HANDBOOK

## *Design. Shop Practice*

*Founded by the late*

WILLIAM KENT, M.E., Sc.D.

**ELEVENTH EDITION, REWRITTEN**

BY

ROBERT THURSTON KENT, M.E.
*Editor-in-Chief*

*With the Collaboration of a Staff of Specialists*

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# PUBLISHER'S PREFACE

In making plans for new editions of our handbooks in mechanical engineering and in electrical engineering, it soon became clear that engineering science and practice had developed to such an extent that handbooks were growing beyond all practical bounds. They had become both bulky and inconvenient and contained much duplicated material. In order to solve the problems presented by these conditions, the editors of our various handbooks were asked to serve as an advisory editorial board.

This board recommended, first, that the fundamental material underlying all engineering be published in a separate volume, and, second, that the existing handbooks as they are revised be issued in several volumes containing material closely related to the specialized branches of engineering. As a result of these recommendations, the Wiley Engineering Handbook Series has been initiated, which in the beginning will comprise the following: Eshbach's "Handbook of Engineering Fundamentals"; Kent's "Mechanical Engineers' Handbook" in two volumes, viz., "Power" and "Design and Shop Practice"; Pender's "Electrical Engineers' Handbook" in two volumes, viz., "Electric Power" and "Communication and Electronics."

This division has also made it possible to devote more space to the various topics so that the entire new series of handbooks contains more complete information on all topics than heretofore has been possible. It is our hope that this new plan will give engineers information that is more useful, more complete, and in more convenient form.

John Wiley & Sons, Inc.

# PREFACE TO THE ELEVENTH EDITION

## March 1938.

The division of this handbook into two volumes has permitted a more thorough treatment of the material in the earlier editions and the inclusion of many new subjects. The development in the past ten years of many new materials of engineering and the improvement in tools, machinery and processes have made many of the data in the earlier editions obsolete. The book, therefore, has practically been rewritten. In the selection of material, the same principle has been applied that has governed the ten preceding editions, namely, that the book should cover the practice of mechanical engineering. The editor, and his collaborators, have presented the data that have been useful to them in their work as practicing engineers.

In order to make this volume complete in itself. some duplication of material in the other volumes of this series was necessary; for example, the mathematical tables and some of the data on the materials of engineering. This duplication has been kept to a minimum, and in all cases the material has been revised to accord more closely with the needs of the mechanical engineer.

The authorship of all special articles has been indicated in the text. Where no author appears, the material was prepared by the editor-in-chief. Every effort has been made to show all sources of material, and to give credit where credit is due.

The publishers and the editor-in-chief extend their thanks to all who have assisted in the preparation of this work, to the collaborators, to the many industrial organizations, and to the individuals who have offered valuable criticism and suggestions.

Robert Thurston Kent.

# ABSTRACT FROM PREFACE TO THE FIRST EDITION, 1895.

More than twenty years ago the author began to follow the advice given by Nystrom: "Every engineer should make his own pocket-book, as he proceeds in study and practice, to suit his particular business." The manuscript pocket-book thus begun, however, soon gave place to more modern means for disposing of the accumulation of engineering facts and figures, viz., the index rerum, the scrap-book, the collection of indexed envelopes, portfolios and boxes, the card catalogue, etc. Four years ago, at the request of the publishers, the labor was begun of selecting from this accumulated mass such matter as pertained to mechanical engineering, and of condensing, digesting, and arranging it in form for publication. In addition to this, a careful examination was made of the transactions of engineering societies, and of the most important recent works on mechanical engineering in order to fill gaps that might be left in the original collection, and insure that no important facts had been overlooked.

Some ideas have been kept in mind during the preparation of the Pocket-book that will, it is believed, cause it to differ from other works of its class. In the first place it was considered that the field of mechanical engineering was so great, and the literature of the subject so vast, that as little space as possible should be given to subjects which especially belong to civil engineering.

Another idea prominently kept in view by the author has been that he would not assume the position of an "authority" in giving rules and formulæ for designing, but only that of compiler, giving not only the name of the originator of the rule, where it was known, but also the volume and page from which it was taken, so that its derivation may be traced when desired. When different formulæ for the same problem have been found they have been given in contrast, and in many cases examples have been calculated by each to show the difference between them. In some cases these differences are quite remarkable. Occasionally the study of these differences has led to the author's devising a new formula, in which case the derivation of the formula is given.

Much attention has been paid to the abstracting of data of experiments from recent periodical literature, and numerous references to other data are given. In this respect the present work will be found to differ from other Pocket-books.

William Kent.

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# SYMBOLS AND ABBREVIATIONS

## Greek Letters

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Α | α | Alpha | Η | η | Eta | Ν | ν | Nu | Τ | τ | Tau |
| Β | β | Beta | Θ | ϑ θ | Theta | Ξ | ξ | Xi | Υ | υ | Upsilon |
| Γ | γ | Gamma | Ι | ι | Iota | Ο | ο | Omicron | Φ | φ | Phi |
| Δ | δ | Delta | Κ | κ | Kappa | Π | π | Pi | Χ | χ | Chi |
| Ε | ϵ | Epsilon | Λ | λ | Lambda | Ρ | ρ | Rho | Ψ | ψ | Psi |
| Ζ | ζ | Zeta | Μ | μ | Mu | Σ | σ ς | Sigma | Ω | ω | Omega |

## Mathematical Signs and Abbreviations

$+$ plus (addition).
$+$ positive.
$-$ minus (subtraction).
$-$ negative.
$\pm$ plus or minus.
$\mp$ minus or plus.
$=$ equals.
$\geqq$ equals or greater than.
$\leqq$ equals or is less than.
$\approx$ approximately equals.
$\times$ multiplied by.
$ab$ or $a.b = a \times b$.
$\div$ divided by.
$/$ divided by.
$\frac{a}{b} = a/b = a \div b$. $15/16 = \frac{15}{16}$.
$0.2 = \frac{2}{10}$; $0.002 = \frac{2}{1000}$.
$\sqrt{\ }$ square root.
$\sqrt[3]{\ }$ cube root.
$\sqrt[4]{\ }$ 4th root.
: is to, : : so is, : to (proportion).
2 : 4 : : 3 : 6, 2 is to 4 as 3 is to 6.
: ratio; divided by.
2 : 4, ratio of 2 to 4 = 2/4.
$>$ greater than.
$<$ less than.
° degrees, arc or thermometer.
′ minutes or feet.
″ seconds or inches.
′ ″ ‴ accents to distinguish letters, as $a'$, $a''$, $a'''$.
$a_1$, $a_2$, $a_3$, $a_b$, $a_c$, read $a$ sub 1, $a$ sub $b$, etc.
( ) [ ] { } —— parentheses, brackets, braces, vinculum; denoting that the numbers enclosed are to be taken together; as, $(a + b)c = \overline{4 + 3} \times 5 = 35$.
$a^2$, $x^3$, $a$ squared, $a$ cubed.
$a^n$, $a$ raised to the $n$th power.
$a^{2/3} = \sqrt[3]{a^2}$, $a^{3/2} = \sqrt{a^3}$.
$a^{-1} = \frac{1}{a}$, $a^{-2} = \frac{1}{a^2}$.
$10^9$ = 10 to the 9th power = 1,000,000,000.
$\sin a$ = the sine of $a$.
$\sin^{-1} a$ = the arc whose sine is $a$.
$\sin a^{-1} = \frac{1}{\sin a}$.
log = logarithm.
$\log_e$ or hyp log = hyperbolic logarithm.
% per cent.
∠ angle.
△ triangle.
sin, sine.
cos, cosine.
tan, tangent.
sec, secant.
versin, versed sine.
cot, cotangent.
cosec, cosecant.
covers, co-versed sine.

In Algebra, the first letters of the alphabet, $a$, $b$, $c$, $d$, etc., are generally used to denote known quantities, and the last letters, $w$, $x$, $y$, $z$, etc., unknown quantities.

*Abbreviations and Symbols commonly used.*

$d$, differential (in calculus).
$\int$, integral (in calculus).
$\int_b^a$, integral between limits $a$ and $b$.
Δ, delta, difference.
Σ, sigma, sign of summation.
$\pi$, pi, ratio of circumference of circle to diameter = 3.14159.
$g$, acceleration due to gravity = 32.16 ft. per second per second.

## Abbreviations

| | |
|---|---|
| abs. | absolute |
| A.C. | alternating-current |
| A.I.E.E. | American Institute of Electrical Engineers |
| A.I.M.E. | American Institute of Mining and Metallurgical Engineers |
| amp. | amperes |
| A.I. and S.I. | American Iron and Steel Institute |
| approx. | approximate |
| A.P.I. | American Petroleum Institute |
| A.R.A. | American Railway Association |
| A.R.E.A. | American Railway Engineering Association |
| A.S.A. | American Standards Association |
| A.S.C.E. | American Society of Civil Engineers |

# SYMBOLS AND ABBREVIATIONS

| | |
|---|---|
| A.S.H.V.E. | American Society of Heating and Ventilating Engineers |
| A.S.M.E. | American Society of Mechanical Engineers |
| A.S.R.E. | American Society of Refrigerating Engineers |
| A.S.S.T. | American Society for Steel Treating |
| A.S.T.M. | American Society for Testing Materials |
| atmos | atmosphere |
| avoir | avoirdupois |
| A.W.G. | American wire gage |
| $B$., $b$ | breadth |
| B. & S | Brown & Sharpe wire gage |
| bar | barometer, barometric |
| bbl | barrel |
| Be | Baumé |
| B.G. | Birmingham gage (tube and sheet) |
| B.Hp | brake horsepower |
| B.M. | board measure |
| Bur. of Std. | Bureau of Standards |
| B.t.u. | British thermal unit(s) |
| Bu | bushel(s) |
| Bull | bulletin |
| B.W.G. | Birmingham wire gage |
| C | one hundred |
| C., cent | centigrade |
| c.c. | cubic centimeter(s) |
| c.f.m. | cubic feet per minute |
| c.g.s. | centimeter-gram-second |
| c.m. | circular mil |
| cal | calorie(s) |
| cg | centigram(s) |
| cir | circular |
| cl | centiliter(s) |
| col | column |
| cu. cm | cubic centimeter(s) |
| cu. ft | cubic feet |
| cu. in | cubic inch(es) |
| cu. m | cubic meter(s) |
| cu. yd | cubic yard(s) |
| cyl | cylinder |
| $D$., $d$ | diameter, depth |
| db | decibel(s) |
| D.C. | direct current |
| def | definition |
| deg | degree(s) |
| diam | diameter |
| $e$ | base of Naperian system of logarithms |
| E.E.I. | Edison Electrical Institute |
| E. Hp | electrical horsepower |
| e.m.f. | electromotive force |
| eff | efficiency |
| ext | external |
| F | Fahrenheit |
| $F$ | force |
| $f, \lambda, \mu$ | coefficient of fricton |
| f.Hp | friction horsepower |
| f.p.m. | feet per minute |
| ft | foot, feet |
| ft.-lb | foot-pound(s) |
| $g$ | acceleration due to gravity |
| g.p.m. | gallons per minute |
| gal | gallon(s) |
| gm | gram(s) |
| gm.-cal | gram-calorie(s) |
| gr | grain(s) |
| $H$, $h$ | height |
| Hg | mercury |
| hor | horizontal |

| | |
|---|---|
| Hp | horsepower |
| Hp.-hr | horsepower-hour(s) |
| hr | hour(s) |
| hyd | hydraulic |
| hyp | hyperbolic |
| hyp. log | hyperbolic or Naperian logarithm |
| $I$ | current, electric; moment of inertia |
| I.Hp | indicated horsepower |
| Imp | Imperial |
| in | inch(es) |
| in.-lb | inch-pound(s) |
| Inst. C.E., I.C.E. | Institute of Civil Engineers |
| Inst. M.E., I.M.E. | Institution of Mechanical Engineers |
| int | internal |
| $J$ | mechanical equivalent of heat |
| $j$ | joule(s) |
| kg | kilogram(s) |
| kg.-cal | kilogram-calorie(s) |
| kg.-m | kilogram-meter(s) |
| kl | kiloliter(s) |
| km | kilometer(s) |
| kva | kilovolt-ampere(s) |
| kw | kilowatt(s) |
| kw.-hr | kilowatt-hour(s) |
| $L$, $l$ | length |
| l | liter |
| lat | latitude |
| lb | pound(s) |
| lin | linear |
| log | logarithm |
| log$_e$ | Naperian logarithm |
| log$_{10}$ | common logarithm |
| M | one thousand |
| m | meter(s) |
| m.-kg | meter-kilogram |
| m.m.f | magnetomotive force |
| m.p.h | miles per hour |
| max | maximum |
| mi | mile(s) |
| min | minimum; minute |
| mm | millimeter |
| $N$ | number (in mathematical tables) |
| nat | natural |
| N.E.L.A. | National Electric Light Association (now Edison Electrical Institute) |
| N.E.M.A. | National Electrical Manufacturers Association |
| No | number |
| O.D. | outside diameter |
| oz | ounce(s) |
| $P$ | pressure |
| p., pp | page, pages |
| p.f | power factor |
| Proc | Proceedings |
| pt | pint(s) |
| $Q$ | quantity or volume |
| qt | quart(s) |
| $R$, $r$ | radius |
| r.m.s. | square root of mean square |
| r.p.m., or rev. per min. | revolutions per minute |
| Ry | railway |
| S.A.E. | Society of Automotive Engineers |
| S.W.G. | Standard (British) wire gage |
| sat | saturated |
| sec | second(s) |
| sp | specific |

# SYMBOLS AND ABBREVIATIONS

| | |
|---|---|
| sp. gr. | specific gravity |
| sp. ht. | specific heat |
| sq | square |
| sq. cm. | square centimeter(s) |
| sq. ft. | square foot, square feet |
| sq. in. | square inch(es) |
| sq. km. | square kilometer(s) |
| std | standard |
| T.S. | tensile strength |
| temp | temperature |
| Trans | Transactions |
| U.S. | United States |
| U.S.S.G. | United States Standard gage |
| ult | ultimate |
| *V*, *v* | velocity; volume |
| vel | velocity |
| vol | volume |
| *vs* | versus |
| *W*, *w* | weight |
| W. & M. | Washburn and Moen wire gage |
| W.I. | wrought iron |
| wt | weight |
| yd | yard(s) |
| y.p. | yield point |
| yr | year(s) |

## Periodicals and Publications

*Am. Electrician*, American Electrician.*

*Am. Lumberman*, American Lumberman, Chicago.

*Am. Mach.*, American Machinist, New York.

*Am. Soc. for Metals*, American Society for Metals, Cleveland.

*Auto. Ind.*, Automotive Industries, New York.

*Blast Furnace and Steel Plant*, Blast Furnace and Steel Plant, Pittsburgh.

*Bull.* Am. Inst. of Min. Eng., Bulletin American Institute of Mining Engineers, New York.

*Bull.* Amer. Ry. Eng. Assoc., Bulletin American Railway Engineers Association, Chicago.

*Bulletin* A.S.A., Bulletin American Standards Association, New York.

*Bull.* Engg. Exp. Sta., Univ. of Ill., Bulletin Engineering Experiment Station, University of Illinois, Urbana, Ill.

*Bull.* School of Engg. Research, Univ. of Toronto, Bulletin of Engineering Research, University of Toronto, Toronto, Canada.

*Bull.* U. S. Bureau of Mines, Bulletin United States Bureau of Mines, Washington, D. C.

*Cass. Mag.*, Cassier's Magazine.*

*Chem. and Met. Engg.*, Chemical and Metallurgical Engineering, New York.

*Civil Engg.*, Civil Engineering, New York.

*Coal Age*, Coal Age, New York.

*Construction Methods*, Construction Methods and Equipment, New York.

*Demag Nachrichten*, Duisburg, Germany.

*El. Rev.*, Electrical Review, Chicago.

*Elec. Jour.*, Electrical Journal, Pittsburgh.

*Electrochem. and Met. Ind.*, Now Chemical and Metallurgical Engineering, New York.

*Engg.*, Engineering, London.

*Engg. Min. Jour.*, Engineering and Mining Journal, New York.

*Eng. News.*, *Engg. News-Record*, Engineering News-Record, New York.

*Foundry Trade Journal*, London.

*Fordertechnik*, Wittenberg, Germany.

*Forging, Stamping and Heat Treating*, now Heat Treating and Forging, Pittsburgh.

*Grits and Grinds*, Norton Co., Worcester, Mass.

*Heat Treating and Forging*, Pittsburgh.

*Indust. Eng.*, Industrial Engineering.*

*Industrial Heating*, Industrial Heating, Pittsburgh.

*Ind. & Engg. Chem.*, Industrial and Engineering Chemistry, New York.

*Iron Age*, Iron Age, New York.

*Iron and Coal Trade Rev.*, Iron and Coal Trade Review, London.

*Jour.* A.W.W.A., Journal American Water Works Association, New York.

*Jour.* Acoustical Soc., Journal Acoustical Society of America, New York.

*Jour.* Am. Ceramic Soc., Journal American Ceramic Society, Columbus, Ohio.

*Jour.* Am. Welding Soc., Journal American Welding Society, New York.

*Jour.* Franklin Inst., Journal Franklin Institute, Philadelphia.

*Jour.* Engg. Inst. of Canada, Journal Engineering Institute of Canada, Montreal, Que.

*Jour.* I. and S., Journal Iron and Steel Institute, London.

*Jour.* Inst. of Metals, Journal Institute of Metals, London.

*Jour.* Op. Soc. of Am., Journal Optical Society of America, New York.

*Journal*, S.A.E., Journal Society of Automotive Engineers, New York.

*Machy.*, *Machinery*, Machinery, New York.

*Matl. Handling and Distrib.*, Material Handling and Distribution, now Mill and Factory, New York.

*Mech. Engg.*, Mechanical Engineering, New York.

*Metal Industry*, London.

*Metal Industry*, New York.

*Metal Progress*, Cleveland.

*Metals and Alloys*, New York.

*Metallurgia*, Manchester, England.

*National Directory of Commodity Specifications*, Government Printing Office, Washington, D.C.

*National Safety News*, Chicago.

*Power*, New York.

*Proc.* A.C.I., Proceedings American Concrete Institute, Philadelphia.

*Proc.* A.S.C.E., Proceedings American Society of Civil Engineers, New York.

*Proc.* A.S.T.M., Proceedings American Society for Testing Materials, Philadelphia.

*Proc.* Am. Ry. Eng. Assoc., Proceedings American Railway Engineers Association, Chicago.

*Proc.* I.M.E., Proceedings Institution of Mechanical Engineers, London.

*Proc.* Inst. C. E., Proceedings Institute of Civil Engineers, London.

*Product Engg.*, Product Engineering, New York.

*Rwy. Age*, Railway Age, New York.

*Revue de Metallurgie*, Paris.

*Safety*, New York.

*Specialized Journal*, Specialized Journal, British Institute of Metals, London.

*Stahl u. Eisen*, Stahl und Eisen, Dusseldorf, Germany.

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* Out of Print.

*Steel*, Cleveland. Formerly Iron Trade Review.
*Steel Maker*, Pittsburgh.
*Tech. Bull.* U. S. Bureau of Stds., Technical Bulletin, U.S. Bureau of Standards, Washington.
*The Brass World*, New York.
*The Fdry.*, *The Foundry*, Cleveland.
*The Locomotive*, Hartford, Conn.
*The Mainspring*, The Mainspring, Wallace Barnes Co., Bristol, Conn.
*Trans.* A.F.A., *Trans.* Am. Fdrymen. Assoc., Transactions American Foundrymen's Association, Chicago.
*Trans.* A.I.E.E., Transactions American Institute of Electrical Engineers, New York.
*Trans.* A.I.M.E., *Trans.* A.I.M. & M.E., Transactions American Institute of Mining and Metallurgical Engineers, New York.
*Trans.* A.S.C.E., Transactions American Society of Civil Engineers, New York.
*Trans.* A.S.H.V.E., Transactions American Society of Heating and Ventilating Engineers New York.
*Trans.* A.S.M., Transactions American Society of Metals, Cleveland.
*Trans.* A.S.M.E., Transactions American Society of Mechanical Engineers, New York.
*Trans.* Cer. Soc., Transactions Ceramic Society, Stoke-on-Trent, England.
*Trans.* Electrochem. Soc., Transactions Electrochemical Society, New York.
*Trans.* Ill. Engg. Soc., Transactions Illinois Engineering Society, Urbana, Ill.
*Zeit.*, V.D.I., Zeitschrift des Verienes deutscher Ingenieure, Berlin.
*Zeitschrift fur Metalkunde*, Berlin.

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* Out of Print

This book is divided into sections, each section carrying its independent sequence of page numbers. For example, 3-15 indicates Section 3, page 15.

# Section 1

# GENERAL PROPERTIES OF MATERIALS

# TABLE OF CONTENTS

## GENERAL PROPERTIES OF MATERIALS

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# GENERAL PROPERTIES OF MATERIALS

## 1. THE CHEMICAL ELEMENTS AND COMPOUNDS

Table 1 gives a list of the chemical elements, together with their atomic numbers and their atomic weights. The atomic weights are the 1934 weights published by the Committee on Atomic Weights of the International Union of Chemistry. Table 2, from Peele's Mining Engineers' Handbook, gives the chemical composition of the more important industrial compounds together with their molecular weights and specific gravity. Other properties of many of these compounds are discussed elsewhere in this volume and in Kent's Mechanical Engineers' Handbook—Power, comprising Vol. 2 of this series. For a complete discussion of the combinations and reactions of the more important of the chemical elements, and of chemical calculations, see Eshbach, Handbook of Engineering Fundamentals, Sec. 10, comprising Vol. 1 of this series.

**Table 1.—The Chemical Elements**

From report of Committee on Atomic Weights of Int. Union of Chemistry, 1934

| Element | Symbol | Atomic Number | Atomic Weight | Element | Symbol | Atomic Number | Atomic Weight | Element | Symbol | Atomic Number | Atomic Weight |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Actinium... | Ac | 89 | ....... | Holmium..... | Ho | 67 | 163.5 | Radon..... | Rn | 86 | 222. |
| Aluminum.. | Al | 13 | 26.97 | Hydrogen..... | H | 1 | 1.0078 | Rhenium.. | Re | 75 | 186.31 |
| Antimony.. | Sb | 51 | 121.76 | Indium....... | In | 49 | 114.76 | Rhodium.. | Rh | 45 | 102.91 |
| Argon...... | A | 18 | 39.944 | Iodine........ | I | 53 | 126.92 | Rubidium.. | Rb | 37 | 85.44 |
| Arsenic..... | As | 33 | 79.91 | Iridium....... | Ir | 77 | 193.1 | Ruthenium | Ru | 44 | 101.7 |
| Barium..... | Ba | 56 | 137.36 | Iron.......... | Fe | 26 | 55.84 | Samarium.. | Sm | 62 | 150.43 |
| Beryllium... | Be | 4 | 9.02 | Krypton...... | Kr | 36 | 82.7 | Scandium.. | Sc | 21 | 45.10 |
| Bismuth.... | Bi | 83 | 209.00 | Lanthanum... | La | 57 | 138.92 | Selenium... | Se | 34 | 78.96 |
| Boron...... | B | 5 | 10.82 | Lead......... | Pb | 82 | 207.22 | Silicon..... | Si | 14 | 28.06 |
| Bromine.... | Br | 35 | 79.916 | Lithium...... | Li | 3 | 6.940 | Silver..... | Ag | 47 | 107.880 |
| Cadmium... | Cd | 48 | 112.41 | Lutecium..... | Lu | 71 | 175.0 | Sodium.... | Na | 11 | 22.997 |
| Calcium.... | Ca | 20 | 40.08 | Magnesium... | Mg | 12 | 24.32 | Strontium.. | Sr | 38 | 87.63 |
| Carbon..... | C | 6 | 12.00 | Manganese.... | Mn | 25 | 54.93 | Sulphur.... | S | 16 | 32.06 |
| Cerium..... | Ce | 58 | 140.13 | Masurium.... | Ma | 43 | ........ | Tantalum.. | Ta | 73 | 181.4 |
| Cesium..... | Cs | 55 | 132.91 | Mercury...... | Hg | 80 | 200.61 | Tellurium.. | Te | 52 | 127.61 |
| Chlorine.... | Cl | 17 | 35.457 | Molybdenum.. | Mo | 42 | 96.0 | Terbium... | Tb | 65 | 159.2 |
| Chromium.. | Cr | 24 | 52.01 | Neodymium... | Nd | 60 | 144.27 | Thallium.. | Tl | 81 | 204.39 |
| Cobalt..... | Co | 27 | 58.94 | Neon......... | Ne | 10 | 20.183 | Thorium... | Th | 90 | 232.12 |
| Columbium. | Cb | 41 | 93.3 | Nickel........ | Ni | 28 | 58.69 | Thulium... | Tm | 69 | 169.4 |
| Copper..... | Cu | 29 | 63.57 | Nitrogen...... | N | 7 | 14.008 | Tin....... | Sn | 50 | 118.70 |
| Dyprosium. | Dy | 66 | 162.46 | Osmium...... | Os | 76 | 191.5 | Titanium.. | Ti | 22 | 47.90 |
| Erbium.... | Er | 68 | 165.20 | Oxygen....... | O | 8 | 16.00 | Tungsten.. | W | 74 | 184.0 |
| Europium.. | Eu | 63 | 152.0 | Palladium.... | Pd | 46 | 106.7 | Uranium... | U | 92 | 238.14 |
| Fluorine.... | F | 9 | 19.00 | Phosphorus... | P | 15 | 31.02 | Vanadium. | V | 23 | 50.95 |
| Gadolinium. | Gd | 64 | 157.30 | Platinum..... | Pt | 78 | 195.23 | Xenon..... | Xe | 54 | 131.3 |
| Gallium.... | Ga | 31 | 69.72 | Polonium..... | Po | 84 | (210) | Ytterbium. | Yb | 70 | 173.04 |
| Germanium. | Ge | 32 | 72.60 | Potassium.... | K | 19 | 39.096 | Yttrium... | Y | 39 | 88.92 |
| Gold....... | Au | 79 | 197.2 | Praseodymium | Pr | 59 | 140.92 | Zinc....... | Zn | 30 | 65.38 |
| Hafnium... | Hf | 72 | 178.6 | Protoactinum. | Pa | 91 | ........ | Zirconium.. | Zr | 40 | 91.22 |
| Helium..... | He | 2 | 4.002 | Radium...... | Ra | 88 | 225.97 |  |  |  |  |

## 2. PROPERTIES OF MATERIALS

**METALS.**—The properties of the metals used in engineering are given in Table 3 and the notes accompanying it. Many of these metals, particularly the ferrous metals, and the more generally used non-ferrous alloys also are discussed in great detail in other sections of this book, as indicated in the notes.

**NON-METALLIC MATERIALS.**—For properties of non-metallic materials, see pp. 5–03 to 5–38.

**WOOD.**—For properties of wood, see p. 5–20.

### Table 2.—The More Important Industrial Compounds

| Name | Formula | Mol. wt. | Sp. gr. |
|---|---|---|---|
| Acetic acid | $HC_2H_3O_2$ | 60.03 | 1.06 |
| Alcohol (grain) | $C_2H_5\cdot OH$ | 46.05 | 0.785 |
| " (wood) | $CH_3\cdot OH$ | 32.03 | 0.791 |
| Alumina | $Al_2O_3$ | 102.2 | 3.86 |
| Alum, ammonium | $Al_2(SO_4)_3\cdot (NH_4)_2SO_4\cdot 24\,H_2O$ | 906.95 | 1.64 |
| " potassium | $Al_2(SO_4)_3\cdot K_2SO_4\cdot 24\,H_2O$ | 949.06 | 1.76 |
| Ammonia | $NH_4OH$ | 35.03 | 0.88 |
| Am chlor (sal-ammoniac) | $NH_4Cl$ | 53.50 | 1.52 |
| Ammonium nitrate | $NH_4NO_3$ | 80.05 | 1.73 |
| " sulphate | $(NH_4)_2SO_4$ | 132.14 | 1.77 |
| Arsenious oxide (white arsenic) | $As_4O_6$ | 395.84 | 3.74 |
| Barium carbonate | $BaCO_3$ | 197.37 | 4.27 |
| " sulphate (blanc fixe) | $BaSO_4$ | 233.44 | 4.40 |
| Boric acid | $H_3BO_3$ | 62.02 | 1.43 |
| Calcium acetate (acet. of lime) | $Ca(C_2H_3O_2)_2.\,H_2O$ | 176.13 | ...... |
| Calcium carbide | $CaC_2$ | 64.07 | 2.22 |
| " carbonate | $CaCO_3$ | 100.07 | 2.85 |
| " oxide (quicklime) | $CaO$ | 56.07 | 3.30 |
| " hydroxide (slaked lime) | $Ca(OH)_2$ | 74.09 | 2.08 |
| " phosphate (phos. of lime) | $Ca_3(PO_4)_2$ | 310.28 | 3.18 |
| " sulphate (plaster paris) | $CaSO_4$ | 136.14 | 2.96 |
| Carbon tetrachloride | $CCl_4$ | 153.84 | 1.58 |
| " disulphide | $CS_2$ | 76.14 | 1.29 |
| Cupric arsenite (paris green) | $CuHAsO_3$ | 187.54 | ...... |
| Cupric oxide | $CuO$ | 79.57 | 6.37 |
| " sulphate (bluestone) | $CuSO_4\cdot 5H_2O$ | 249.72 | 2.28 |
| Ferric oxide | $Fe_2O_3$ | 159.68 | 5.18 |
| Ferrous oxide | $FeO$ | 71.84 | ...... |
| " sulphate (copperas) | $FeSO_4\cdot 7H_2O$ | 278.02 | 1.90 |
| Hydrochloric acid | $HCl$ | 36.47 | ...... |
| Hydrogen peroxide | $H_2O_2$ | 34.02 | ...... |
| Lead acetate (sugar of lead) | $Pb(C_2H_3O_2)_2\cdot 3\,H_2O$ | 379.20 | 2.50 |
| Lead carbonate (white lead) | $2PbCO_3\cdot Pb(OH)_2$ | 775.31 | ...... |
| Lead monoxide (litharge) | $PbO$ | 223.10 | 9.37 |
| " oxide (red lead) | $Pb_3O_4$ | 685.30 | 9.10 |
| " sulphate | $PbSO_4$ | 303.17 | 6.23 |
| Magnesium carbonate | $MgCO_3$ | 84.32 | 3.04 |
| " sulph. (epsom salt) | $MgSO_4\cdot 7\,H_2O$ | 246.50 | 1.68 |
| Manganese dioxide | $MnO_2$ | 86.93 | 5.03 |
| Mercuric chlor. (corrosive sublimate) | $HgCl_2$ | 271.52 | 5.40 |
| Nitric acid | $HNO_3$ | 63.02 | 1.53 |
| Phosphoric acid | $HPO_3$ | 80.05 | 2.30 |
| Potassium carbonate (potash) | $K_2CO_3\cdot 2\,H_2O$ | 174.23 | 2.04 |
| Potassium chlorate | $KClO_3$ | 122.56 | 2.34 |
| " chloride | $KCl$ | 74.56 | 1.99 |
| " chromate | $K_2CrO_4$ | 194.20 | 2.73 |
| " cyanide | $KCN$ | 65.11 | 1.52 |
| " ferricyanide | $K_3Fe(CN)_6$ | 329.20 | 1.81 |
| " ferrocyanide | $K_4Fe(CN_6)\cdot 3\,H_2O$ | 422.35 | 1.85 |
| " hydroxide (caustic potash) | $KOH$ | 56.11 | 2.04 |
| " nitrate (saltpeter) | $KNO_3$ | 101.11 | 2.10 |
| " permanganate | $KMnO_4$ | 158.03 | 2.70 |
| " sulphate | $K_2SO_4$ | 174.27 | 2.66 |
| Silver nitrate (lunar caustic) | $AgNO_3$ | 169.89 | 4.35 |
| Sodium borate (borax) | $Na_2B_4O_7\cdot 10\,H_2O$ | 382.16 | 1.69 |
| " carbonate (soda) | $Na_2CO_3$ | 106.00 | 2.47 |
| " bicarbonate | $NaHCO_3$ | 84.01 | 2.20 |
| " chloride (salt) | $NaCl$ | 58.46 | 2.17 |
| " cyanide | $NaCN$ | 49.01 | ...... |
| " hydroxide (caustic soda) | $NaOH$ | 40.01 | 2.13 |
| " nitrate (Chile saltpeter) | $NaNO_3$ | 85.01 | 2.27 |
| " silicate (water glass) | $Na_2Si_4O_9$ | 303.20 | ...... |
| " sulphate (Glauber salts) | $Na_2SO_4\cdot 7\,H_2O$ | 268.18 | ...... |
| " sulphite | $Na_2SO_3\cdot 7\,H_2O$ | 252.18 | 1.59 |
| " thiosulphate | $Na_2S_2O_3\cdot 5H_2O$ | 248.22 | 1.73 |
| Sulphur dioxide | $SO_2$ | 64.07 | ...... |
| Sulphuric acid | $H_2SO_4$ | 98.09 | ...... |
| Tartaric acid | $H_2C_4H_4O_6$ | 150.05 | 1.75 |
| Tin chloride | $SnCl_4$ | 260.84 | 2.28 |
| Vanadium oxide | $V_2O_5$ | 182.00 | 3.36 |
| Zinc chloride | $ZnCl_2$ | 136.29 | 2.91 |
| " oxide (zinc white) | $ZnO$ | 81.37 | 5.78 |
| " sulphate | $ZnSO_4\cdot 7\,H_2O$ | 287.55 | 1.97 |

## Notes for Table 3

1. **Aluminum.**—The lightest of all useful metals except beryllium and magnesium, soft ductile and malleable. Color, white with a bluish cast. Very non-corrosive. Strength varies widely, depending on the alloying materials and treatment. For further properties, see p. 4–25 to 4–49.

2. **Antimony.**—A brittle metal of bluish-white color, and highly crystalline or laminated structure. Burns in air with a bluish-white flame. Chiefly used for manufacturing such alloys as type-metal (antimony 1, lead 4), brittania (antimony 1, tin 9), and various anti-friction metals. Commercial antimony contains minute quantities of silver, arsenic, tin, lead, copper, iron, nickel, cobalt and sulphur.

3. **Arsenic.**—A chemical element midway between the metals and non-metals. It occurs in nature in combination with many metals and with sulphur. Arsenic compounds are used in the laboratory for titrating iodine solutions. Used in industry in dyeing, and vermin- and insect-poisoning compounds. Also used in medicine. Its soluble compounds are extremely poisonous.

4. **Barium.**—A metal used in the manufacture of pigments and in pyrotechnic compounds. Also has been used in the commercial preparation of oxygen.

5. **Beryllium.**—A light metal of silvery luster used in the production of light alloys. Will not decompose in water at any temperature below 212° F.

*(Continued on page 1-08)*

## Table 3.—Properties of the Useful Metals

| Substance | Specific Gravity | Density, lb. per cu. ft. | Melting Point, deg. F. | Latent Heat of Fusion, B.t.u. | Specific Heat | | Thermal Coefficient of Linear Expansion per deg. F.* | Thermal Conductivity, B.t.u., per hr. per sq. ft. per deg. F. per in. | Electrical Resistivity, microhm-cm. | Temperature Coefficient of Resistivity | Tensile Strength, 1000 lb. per sq. in. | Elongation in 2 in. percent | Reduction of Area, percent | Modulus of Elasticity, millions | Shearing Modulus, millions | Brinell Hardness No. | See p. 1-04, Note No. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | B.t.u. per lb. per deg. F. | Temp., deg. F. | | | | | | | | | | | |
| Aluminum....... | 2.70 | 168.56 | 1217.7 | 138.24 | 0.2089<br>.2250<br>.2739 | 32<br>61–579<br>932 | $0.0_{4}1233$[3]<br>$.0_{4}1285$[5]<br>$.0_{4}1750$[4] | 1475[9]<br>1560[10] | 2.828[14]<br>3.86[11]<br>8.0[18] | 0.0039[12]<br>.0040[11]<br>.0050[23] | 9[43]<br>19–44[44]<br>13–68[46] | 60[43]<br>0.5–13[44]<br>3–35[46] | .......<br>.......<br>....... | 10[43]<br>.......<br>....... | 3.87[43]<br>......<br>...... | 15[43, 45]<br>40–150[43, 45]<br>23–116[45, 46] | 1 |
| Antimony....... | 6.62 | 413.2 | 1166.0 | 70.2 | .0489<br>.0503<br>.0520 | 59<br>212<br>392 | $.0_{5}587$[3] | 128[9]<br>115 | 41.7[14] | .0036[14] | 1.5[47] | .......... | ....... | 11.3[47] | 2.86[47] | .......... | 2 |
| Arsenic.......... | 5.73 | 357.7 | 1562. | ....... | .0822[1]<br>.0861[2] | 32–212<br>32–212 | $.0_{5}3105$[5] | ........ | 35.[9] | .0042[14] | .............. | .......... | ....... | ....... | ...... | 147[45] | 3 |
| Barium.......... | 3.50 | 218.5 | 1562. | ....... | .068 | −121 to +68 | ............... | ........ | 9.8[14] | .0033[14] | .............. | .......... | ....... | ....... | ...... | .......... | 4 |
| Beryllium....... | 1.80 | 112.4 | 2336. | 572.4 | ......... | ....... | $.0_{5}6778$[14] | ........ | .............. | ........... | .............. | .......... | ....... | ....... | ...... | .......... | 5 |
| Bismuth......... | 9.79 | 610.6 | 519.8 | 22.75 | .0302 | 68–212 | $.0_{5}731$[3] | 56[12]<br>47[11] | 119.0[9]<br>160.2[12] | .004[14] | .............. | .......... | ....... | ....... | ...... | .......... | 6 |
| Brass and bronze. | 7.57 to 8.95 | 472–558 | 1625–2235 | ....... | 0.086–0.092 | ....... | $0.0_{4}167$–$0.0_{4}214$ | 350–1712 | 4.21–16.3[14] | ........... | 22–125 | 0–60 | 6–50 | 14–16 | ...... | 40–240 | 7 |
| Cadmium........ | 8.65 | 540.0 | 609.6 | 24.59 | .0570 | 212 | $.0_{4}1755$[3] | 648[9]<br>624[11] | 7.54[12]<br>9.82[11]<br>34.1[19] | .0038[14] | 12[48] | 52 | 80 | 10[48] | 3.5[48] | 21–24[45, 48] | 8 |
| Calcium......... | 1.55 | 96.8 | 1490. | 140.4 | .170 | 32–358 | $.0_{4}1389$[37] | ........ | 4.6[14] | .0036[24] | 8.7[48] | 6[49] | ....... | ....... | ...... | 42[45] | 9 |
| Chromium...... | 6.93 | 432.6 | 2939. | 125.9 | .1039<br>.1121<br>.1872 | 32<br>212<br>1112 | $.0_{5}3778$[36] | ........ | 2.6[9] | ........... | .............. | .......... | ....... | ....... | ...... | 91[45] | 10 |
| Cobalt.......... | 8.71 | 543.8 | 2696. | 115.2 | .1452<br>.204 | 932<br>1832 | $.0_{5}687$[5] | ........ | 9.7[14] | .0066[3] | 34[48]<br>96[50] | 5[50] | 8[50] | ....... | ...... | 124[48, 51] | 11 |
| Copper.......... | 8.89 | 555.0 | 1981±5 | 75.6 | .0942<br>.0951<br>.1259 | 212<br>59–450<br>1652 | $.0_{5}926$[3] | 2680[9]<br>2605[10] | 1.724[14, 20]<br>1.77[14, 21]<br>2.92[10, 22]<br>4.10[18] | .0039[14, 20]<br>.0038[14, 21]<br>.0038[11]<br>.0042[18] | 20–30[48]<br>30–40[52]<br>51[53]<br>50–70[54] | 4 | ....... | 16[43] | ...... | 103[45] | 12 |
| Gold............ | 19.31 | 1204.9 | 1945. | 28.6 | .0316 | 32–212 | $.0_{5}817$[3] | 2070[9]<br>2030[11] | 2.22[9]<br>2.44[14] | .0034[14]<br>.0025[11, 20] | 25[48]<br>37[54] | ....... | ....... | ....... | ...... | .......... | 13 |
| Indium.......... | 7.31 | 456.1 | 311. | ....... | .0570 | 32–212 | $.0_{4}2317$[5] | ........ | 8.37[9] | ........... | 15.9 | .......... | ....... | ....... | ...... | .......... | 14 |
| Iridium......... | 22.43 | 1399.7 | 4262† | 47.0 | .0323 | 54–212 | $.0_{5}489$[3] | 4091[5] | 6.10[9] | .0041[3] | .............. | .......... | ....... | ....... | ...... | .......... | 15 |

*See footnotes at end of table.*

*Table continued on following page.*

**Table 3.—Properties of the Useful Metals—*Continued***

| Substance | Specific Gravity | Density, lb. per cu. ft. | Melting Point, deg. F. | Latent Heat of Fusion, B.t.u. | Specific Heat | | Thermal Coefficient of Linear Expansion per deg. F.* | Thermal Conductivity, B.t.u., per hr. per sq. ft. per deg. F. per in. | Electrical Resistivity, microhm-cm. | Temperature Coefficient of Resistivity | Tensile Strength, 1000 lb. per sq. in. | Elongation in 2 in. percent | Reduction of Area, percent | Modulus of Elasticity, millions | Shearing Modulus, millions | Brinell Hardness No. | See p. 1–04, Note No. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | B.t.u. per lb. per deg. F. | Temp., deg. F. | | | | | | | | | | | |
| Iron (99.98%)... | 7.87 | 491.3 | 2786 | 116.97 | ‡ | ....... | $.0_5 6611$[3] | 468[12]<br>438[11] | 10.[14]<br>17.8[25] | .0050[14]<br>.0068[11] | 40. | 40 | 80 | 29.7 | 10. | 50–90 | 16 |
| " , cast, gray.. | 7.03–7.13 | 438.7 to 444.9 | 1900 to 2200 | 41.4 | 0.1189 | 68–212 | $.0_5 589$[5] | 350[9]<br>220[10] | 74.6 to 98.0 | ...........<br>........... | 18–50 | ......... | ....... | ....... | ...... | 132–270 | |
| " , wrought... | 7.80–7.90 | 486.7 to 493.0 | 2700 to 2900 | ..... | .1152 | 59–212 | $.0_5 633$[6] | 423[9]<br>360[10] | 160 | ........... | 50± | 21[55] | 30 | 28.2 | ...... | ......... | |
| Lead........... | 11.341 | 707.7 | 621± 4.5 | 9.65 | .0319 | 61–493 | $.0_4 1516$[3] | 241[12]<br>235[11] | 28.0[26]<br>36.9[27] | .0039[14] | 1.7–3.0 | ......... | ....... | 2.1–2.5 | 0.78 | 4.2 | 17 |
| Magnesium...... | 1.74 | 108.6 | 1203.8 | 126.0 | .2492 | 68–212 | $.0_4 1450$[5] | 1090[9]<br>1015[10] | 4.35[9]<br>5.99[25]<br>11.9[18] | .0038[9]<br>.0045[11]<br>.0036[23] | 27 | 15 | ....... | 6.25 | ...... | 37[45] | 18 |
| Manganese...... | 7.20 | 449.5 | 2246. | ....... | .1211 | 68–212 | $.0_4 1267$[36] | ........ | 5.±[24] | ........... | ............. | ......... | ....... | ....... | ...... | ......... | 19 |
| Mercury........ | 13.546 | 845.3 | −37.98 | 5.07 | .32<br>.3284 | −121 to +68<br>212 | $.0_3 1010$[39] | 58[9]<br>55[13] | 95.78[14]<br>103.25[11] | .00088[9]<br>§ | ............. | ......... | ....... | ....... | ...... | ......... | 20 |
| Molybdenum.... | 10.21 | 636.8 | 4595† | ....... | .0647 | 68–212 | $.0_5 2722$ | 1000[9]<br>965[11] | 5.7[14] | .0033[28] | 148–154[56]<br>42–125[52] | 4.7–5.7[56]<br>3.15–20[52] | 65–68[56]<br>0–66[52] | ....... | ...... | ......... | 21 |
| Nickel.......... | 8.85 | 552.5 | 2645 | 8.35 | .109<br>.1128<br>.1299<br>.1608 | 64–212<br>212<br>932<br>1832 | $.0_5 566$[5] | 408[9]<br>390[10] | 7.8[14]<br>11.12[9]<br>60.2[18] | .006[14]<br>.0043[11]<br>.0030[23] | 60–70[48]<br>70–85[52, 57]<br>95–115[58]<br>80–95[59]<br>100–105[60] | 15–35<br>40–50<br>2–8<br>25–35<br>15–20 | 30–50<br>55–75<br>.......<br>40–60<br>....... | .......<br>.......<br>.......<br>.......<br>....... | ......<br>......<br>......<br>......<br>...... | 75–95[45]<br>85–125[45]<br>.........<br>215–240[45]<br>......... | 22 |
| Palladium....... | 12.01 | 749.2 | 2820±9 | 65.34 | .0714 | 32–2309 | $.0_5 653$[5] | 489[12] | 11.[14] | .0033[14] | 39[21] | ......... | ....... | 17 | 6.4 | 49 | |
| Platinum........ | 21.38 | 1334.1 | 3191±9 | 48.96 | .0319<br>.0359 | 68–212<br>68–2372 | $.0_5 499$[5] | 482[9]<br>500[11] | 10.96[9]<br>14.85[11] | .0037[9]<br>.003[14] | 34.1[52]<br>53.[21] | 50[52]<br>18[21] | .......<br>....... | .......<br>24.2[21] | ......<br>9.3[21] | 24[45, 52]<br>13[21] | 23 |
| Potassium....... | 0.8702 | 54.3 | 144.1 | 28.26 | .170 | −301 to +68 | $.0_4 4607$[7] | 675[14] | 6.1[9] | .0055[9] | ............. | ......... | ....... | ....... | ...... | 0.037[61] | 24 |
| Rhodium........ | 12.45 | 776.6 | 3542† | ....... | .0580 | 50–207 | ,$.0_5 472$[5] | 610[15] | 4.69[9] | .0043[9] | ............. | ......... | ....... | 42.5 | ...... | 139[48] | |
| Silver.......... | 10.50 | 655.5 | 1760.9 | 37.92 | .0559<br>.05663<br>.05987 | 32–212<br>212<br>63–945 | $.0_4 1067$[5] | 2850[9]<br>2600[10] | 1.468[9, 22]<br>1.629[12] | .0038[14]<br>.0036[11] | 15.4[48]<br>41.2[62]<br>44.1–51.2[21] | 41. | 37.7 | 11.2[21] | 3.8 | 42[45] | 25 |
| Sodium.......... | 0.9712 | 60.6 | 207.5 | 57.06 | .253 | −301 to +68 | $.0_3 12554$[8] | 950[9]<br>835[11] | 4.3[9]<br>5.4[30] | .0054[14] | ............. | ......... | ....... | ....... | ...... | 0.07[63] | 26 |

*See footnotes at end of table.*

*Table continued on following page.*

### Table 3.—Properties of the Useful Metals—*Continued*

| Substance | Specific Gravity | Density, lb. per cu. ft. | Melting Point, deg. F. | Latent Heat of Fusion, B.t.u. | Specific Heat | | Thermal Coefficient of Linear Expansion per deg. F.* | Thermal Conductivity, B.t.u., per hr. per sq. ft. per deg. F. per in. | Electrical Resistivity, microhm-cm. | Temperature Coefficient of Resistivity | Tensile Strength, 1000 lb. per sq. in. | Elongation in 2 in. per cent | Reduction of Area, percent | Modulus of Elasticity, millions | Shearing Modulus, millions | Brinell Hardness No. | See p. 1–04, Note No. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | B.t.u. per lb. per deg. F. | Temp., deg. F. | | | | | | | | | | | |
| Steel | 7.60–<br>7.80 | 474–486 | 2550± | ...... | 0.117± | ...... | $0.0_5734^5$<br>$0.0_56087^{7,20}$ | $310^9$<br>$300^{10}$ | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 27 |
| Tantalum | 16.6 | 1036.3 | 5252† | ...... | .033<br>.043 | −301 to +68<br>2552 | $.0_53639^3$ | $376^9$<br>$505^{16}$ | $15.5^{14}$ | $.0031^{14}$ | $132^{21}$ | ...... | ...... | 27 | ...... | ...... | 28 |
| Tellurium | 6.25 | 390.2 | 845.6 | 13.1 | .0483 | 59–212 | $.0_42048^3$ | $41.5^{41}$ | $200{,}000^{31}$ | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| Tin | 7.29 | 455± | 449.4 | 25.2 | .0551 | 70–228 | $.0_41275^3$ | $450^9$<br>$425^{11}$ | $13.0^9$<br>$11.5^{14}$<br>$18.2^{32}$ | $.0042^{14}$ | $4^{48}$<br>$5^{64}$<br>$10^{65}$ | $35^{48}$<br>......<br>...... | ......<br>......<br>...... | $4^{48}$<br>$5.7–7.8^{64}$ | 2.4 | $14.^{45}$ | 29 |
| Titanium | 4.50 | 280.9 | 3263† | ...... | .1125 | 32–212 | ...... | ...... | $3.2^{24}$ | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| Tungsten | 19.0 | 1186± | 6152† | ...... | .0336<br>.0337 | 32–212<br>1832 | $.0_52466^{40}$ | $1381^{12}$<br>$723^{17}$ | $5.51^{14}$<br>$41.4^{33}$ | $.0045^{12}$<br>$.0089^{34}$ | $50–215^{66}$<br>$264–700^{65}$ | ...... | ...... | 57 | ...... | ...... | 30 |
| Vanadium | 5.60 | 349.6 | 3128 | ...... | .1153 | 32–212 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ... |
| Zinc | 7.14 | 445.8 | 786.9 | 50.63 | .0931<br>.0951<br>.1040 | 68–212<br>212<br>572 | $.0_41653^3$ | $775^9$<br>$750^{11}$ | $5.75^9$<br>$8.00^{27}$<br>10.37 | $.0037^{14}$ | $18–35^{64}$ | ...... | ...... | 12 | 5 | 39 | 31 |
| Zirconium | 6.53 | 407.7 | 3092† | ...... | .0660 | 32–212 | ...... | ...... | 170± | $.0012^{42}$ | ...... | ...... | ...... | ...... | ...... | ...... | ... |

* A subscript after a character indicates the number of times it is repeated; thus $0.0_38 = 0.0008$.

† Value uncertain; temperatures above melting point of platinum may be 100° F. in error.

‡ Dr.-Ing. P. Oberhoffer, in *Zeit.* des Vereines Deutscher Ingenieure (*Eng. Digest*, Sept., 1908), describes some experiments on the specific heat of nearly pure iron. The following mean specific heats were obtained:

| Temp., deg. F. | 500 | 600 | 800 | 1000 | 1200 | 1300 |
|---|---|---|---|---|---|---|
| Specific Heat | 0.1228 | 0.1266 | 0.1324 | 0.1388 | 0.1462 | 0.1601 |
| Temp., deg. F. | 1500 | 1800 | 2100 | 2400 | 2700 | |
| Specific Heat | 0.1698 | 0.1682 | 0.1667 | 0.1662 | 0.1666 | |

The specific heat increases steadily between 500° and 1200° F. Then it increases rapidly to 1400° F., after which it remains nearly constant.

§ $R_t = R_0(1 + 0.0_3494t + 0.0_5555t^2)$

[1] Gray arsenic
[2] Black arsenic
[3] Mean coefficient, 32°–212° F.
[4] At 1112° F.
[5] At 104° F.
[6] From 0°–212° F.
[7] 32–122° F.
[8] 32–194° F.
[9] At 32° F.
[10] At 400° F.
[11] At 212° F.
[12] At 64° F.
[13] At 120° F.
[14] At 68° F.
[15] At 63° F.
[16] At 3092° F.
[17] At 2912° F.
[18] At 752° F.
[19] At 604° F.
[20] Annealed
[21] Hard drawn
[22] Electrolytic
[23] At 932° F.
[24] At ordinary temperatures
[25] At 209° F.
[26] At 195° F.
[27] At 376° F.
[28] At 77° F.
[29] At 217° F.
[30] At 131° F.
[31] At 67° F.
[32] At 96.6° F.
[33] At 2240° F.
[34] At 1832° F.
[35] At 98.4° F.
[36] From 68–212° F.
[37] From 32–68° F.
[38] From 32–352° F.
[39] Cubical expansion at 68° F.
[40] At 80.6° F.
[41] At 113° F.
[42] From −112° to +86° F.
[43] Pure metal
[44] Cast alloys
[45] 500 kg. load, 10 mm. ball
[46] Wrought alloys
[47] Wire
[48] Cast
[49] In 5 cm.
[50] Drawn wire
[51] 1600 kg. load, 10 mm. ball
[52] Annealed
[53] Hard sheet
[54] Cold rolled or drawn
[55] Elongation in 8 in.
[56] Swaged and drawn, hot
[57] Forged or hot rolled
[58] Sheet and strip, full hard
[59] Rod, normalized
[60] Tubing as drawn
[61] 100 mm. ball, 1.6 kg. load
[62] Hard rolled
[63] 10 mm. ball, 3.2 kg. load
[64] Rolled
[65] Drawn
[66] Swaged

**6. Bismuth.**—Color, light red. Highly crystalline, and so brittle that it can readily be pulverized. Expands in cooling. Tribe has shown that this expansion does not occur until after solidification. Bismuth is the most diamagnetic element known, a sphere of it being repelled by a strong magnet. Chiefly used in fusible alloys for electric fuses, plugs for automatic fire alarms and automatic sprinkler systems, and as stereotype metal in the printing industries.

**7. Brass and Bronze.**—Alloys of copper, tin, zinc and other metals, covering a wide range of properties and uses, depending on the proportions of the alloying metals and upon the method of manufacture and treatment. See pp. 4–18 to 4–33.

**8. Cadmium.**—Color bluish-white, lustrous, with a fibrous fracture. Used as an ingredient in some fusible alloys with lead, tin, and bismuth. Alloys of cadmium are used for the fusible tips of automatic sprinklers, safety fusible plugs for boilers, cadmium cliché metal for stereotype plates, and solders in which cadmium is substituted for part or all of the tin commonly used. A fraction of 1% cadmium is used as a deoxidizer for bronze telephone and telegraph wires and cables in Europe. It is also used as a deoxidizer in making nickel alloys. It is a better rust preventive than nickel and it may be used in combination with nickel in nickel plating, the first coat consisting of cadmium and the second coat of tarnish-proof nickel.

**9. Calcium.**—A metal found in nature in the form of carbonate, limestone, marble, chalk, dolomite, gypsum, etc. Metallic calcium is used in an alloy for anti-friction metals for bearings. The carbonate is used in the preparation of the pigment whiting used in the paint and paper industries. Calcium bicarbonate is the cause of temporary hardness in water and calcium sulphate causes permanent hardness.

**10. Chromium.**—A steel-white metal, harder than iron, copper, or nickel, and capable of taking a brilliant polish. In the massive form it is not affected by moisture, air, and only very slightly by dilute acids. Now produced on a large scale for use in stainless and alloy steels. Also used as a protective surface for metals in the form of plate. Chromuim compounds are used in the manufacture and application of dyes and pigments, tanning hides, and photography.

**11. Cobalt.**—A silver-white metal, slightly harder than iron or nickel. Magnetic, in all temperatures up to 2100° F., then becoming non-magnetic. When polished is not appreciably affected by exposure to air or water. Dissolves in nearly all dilute acids, evolving hydrogen. Is not attacked by alkali solutions or fused alkalies. Used in the manufacture of ferrous and non-ferrous alloys. Steel containing cobalt and tungsten is used in permanent magnets for telephony. Non-ferrous alloys containing over 50% cobalt with varying amounts of tungsten, chromium, and occasionally molybdenum are used in high-speed cutting tools (see p. 21–25).

**12. Copper.**—See p. 4–09.

**13. Gold.**—The most malleable and ductile of all metals. One ounce Troy may be beaten so as to cover 160 sq. ft. of surface. The average thickness of gold leaf is 1/282000 in. or 100 sq. ft. per ounce. One grain may be drawn into a wire 500 ft. long. Ductility is destroyed by the presence of 1/2000 part of lead, bismuth or antimony. Gold is hardened by the addition of silver or copper. By jewelers, the fineness of gold is expressed in carats, pure gold being 24 carats, $^{3}/_{4}$ fine, 18 carats, etc.

**14. Indium.**—A white, lustrous metal, very soft and ductile. Has high surface stability at ordinary temperatures, but oxidizes and burns at temperatures above its melting point, especially if finely divided. Can be electroplated, simultaneously with other elements, from a sulphuric acid bath containing sodium citrate. Its addition to other metals increases their surface stability and hardness until the In is in excess. Of use in the automotive, electrical, jewelry, silverware, and dental industries. See article by W. S. Murray, *Ind. & Engg. Chem.*, June, 1932.

**15. Iridium.**—One of the rarer metals. Has a white luster, resembling that of steel. Is quite brittle when cold, but at white heat is somewhat malleable. Is extremely infusible and almost absolutely inoxidizible. For uses of iridium, methods of manufacturing it, etc., see W. L. Dudley, Iridium Industry, *Trans.*A.I.M.E., 1884.

**16. Iron.**—For discussion of the properties of iron, see p. 2–03; of cast iron, see p. 2–03, and Foundry Practice, p. 20–03; of wrought iron, see p. 2–04; of steel, see pp. 2–12 to 2–48.

**17. Lead.**—Brittle just below the melting point, with a crystalline fracture, malleable and ductile. Tenacity so low that it can be drawn into wire only with difficulty. Elasticity is low, and the metal flows under very slight stress. Rendered brittle, and easily fusible by the presence of antimony and arsenic. Penn. R. R. specifications call for not less than 99 $^{1}/_{2}$% metallic lead in No. 1 grade, and not less than 97 $^{1}/_{2}$% metallic lead for No. 2 grade. Shrinkage in casting, 0.3125 in. per ft.

**18. Magnesium.**—Color, brilliant silver-white. Slightly malleable when cold, but very malleable and ductile at 660° F. Can be cast in iron molds or dry sand, and extruded into shapes or wire at 1022° F. Is stable in air at ordinary temperatures, but combines readily with oxygen and nitrogen at temperatures above its melting point. Is nearly non-corrosive, a film of oxide forming in the presence of moisture which prevents further oxidation. It is attacked by the acids $H_2SO_4$, $HNO_3$ and HCl, but is not attacked by alkalies. At red heat, it reduces CO, $CO_2$, $SO_2$, $SiO_2$ and most metallic oxides and many hydrocarbons. In form of filings, wire or ribbon it burns in air with a dazzling brilliancy, and is used for flash and signal lights. Its heat of combustion is 10,710 B.t.u. per lb. Alloys readily with aluminum, copper, nickel, zinc, tin, cadmium, silver, silicon and other non-ferrous metals. Principally used as a deoxidizer of non-ferrous metals, and for alloying with aluminum; 5% Mg added to aluminum gives about as much increase of strength and hardness as 10% of copper. Also used for laboratory reactions. See also p. 4–55.

**19. Manganese.**—The pure metal is not used in the arts, but alloys of manganese and iron (spiegeleisen, containing below 25% Mn, and ferro-manganese, containing 25 to 90% Mn) are used in the manufacture of steel. Metallic manganese when alloyed with iron, oxidizes rapidly in air. In steel manufacture it removes oxygen from the bath of steel.

**20. Mercury.**—A silver-white metal, liquid at temperatures above −38° F. Unchangeable as gold, silver and platinum in the atmosphere at ordinary temperatures, but oxidizes to the red oxide when near its boiling-point. Easily tarnished by sulphur fumes, also by dust, from which it may be freed by straining through cloth. No metal, except iron or platinum, should be allowed to touch mercury. The smallest portions of tin, lead, zinc and even copper to a less extent, cause it to tarnish and lose its perfect liquidity.

**21. Molybdenum.**—A silvery-white, malleable metal, softer than steel. Its tensile strength, drawn into wire, is about $^1/_2$ that of tungsten or steel wire of equivalent diameter. Is resistant to acids; concentrated $H_2SO_4$ and HCl attack it slowly, but moderately dilute $HNO_3$ and *aqua regia* dissolve it rapidly. Is attacked by fused, but not by liquid, caustic alkalies. Is not affected by air at ordinary temperatures, but oxide forms at a dull red heat. Used as a support for electric lamp filaments, windings for electric resistance furnaces, X-ray apparatus, wireless telegraphy plates, and for alloying with steels to produce high strength. Steels with 1% Mo are used for permanent magnets, rustless steels and high-speed tools. See also p. 4–60.

**22. Nickel.**—Color, silvery-white with a strong luster, not tarnishing in air. Ductile, hard, magnetic and as tenacious as iron. Loses its magnetic quality at 662° F. Can be welded to itself or to iron at a white heat. Resists cold sulphuric and hydrochloric acid, but is soluble in nitric acid or *aqua regia*. Chiefly used in alloys with copper, as German-silver, nickel-silver, etc., and also in the manufacture of steel to increase its hardness and strength; also used for nickel-plating. See also pp. 4–49 to 4–54.

**23. Platinum.**—Color, whitish steel-gray; malleable, very ductile, and as unalterable by ordinary agencies as gold. Fused and refined, it is as soft as copper. Fusible only by the oxy-hydrogen blowpipe or strong electric currents. Combined with iridium, it forms an alloy of great hardness, which has been used for gun-vents, and for standard weights and measures. The most important uses of platinum in the arts are for vessels for chemical laboratories and manufactories, and for the connecting wires in incandescent electric lamps, and for electrical contact points. Cubical expansion less than that of any other metal except the rare metals, and almost the same as glass.

**24. Potassium.**—A metal forming the basis of many fertilizers.

**25. Silver.**—Whitest of all metals, very malleable and ductile.

**26. Sodium.**—One of the alkali metals used in the preparation of important organic compounds, dyes, medicinal substances and perfumes. Becomes an important reducing agent when alloyed with mercury. Sodium and potassium hydroxides are known as caustic alkalies or lye.

**27. Steel.**—A compound of iron and carbon, containing up to 1.7 carbon. Alloyed with other metals, it covers a wide range of properties and uses. For properties, uses and treatment of steel, see pp. 2–12 to 2–48. For properties and treatment of tool steel, see p. 21–25.

**28. Tantalum.**—Has a modulus of elasticity of 27,000,000. The pure metal can be hammered into thin sheets or drawn into fine wire. Has the hardness of soft steel, and a tensile strength higher than steel or platinum. The coefficient of expansion is less than that of platinum, and it can be sealed into glass. At temperatures below 572° F., pure tantalum will resist all acids except fluoric.

**29. Tin.**—White, lustrous, soft, malleable. Not sensibly volatile when at ordinary temperature. Malleability decreases with increase of temperature, until at about 390° F. it is very brittle. The presence of 1 to 2% of copper or lead increases its hardness. Chief uses are for coating of sheet-iron (called tin-plate) and for making alloys with copper and other metals. Commercial tin contains copper, iron, bismuth and other impurities.

**30. Tungsten.**—A bright, steel-gray metal, hard but so ductile that it may be drawn into very fine wire. Its chief use is for filaments of incandescent electric lamps. It is also used as an alloy in high-speed tool-steel, as it increases the hardness by keeping the carbon in solution. Its presence in high-carbon steels increases the magnetic retentivity, and it therefore is used in the steel of permanent magnets. See also p. 4–58.

**31. Zinc.**—Color, bluish-white; ductile and malleable. Volatilizes and burns in air when melted, with bluish-white fumes of zinc oxide. Compressive strength about 20,000 lb. per sq. in. Shrinkage in casting 0.3125 in. per ft. Is practically non-corrosive in air, a thin film of carbonate of zinc forming upon it. Its principal uses are for coating iron surfaces, called "galvanizing," and for making brass and other alloys. For additional properties of zinc and zinc alloys see pp. 4–10 to 4–13.

## 3. SPECIFIC GRAVITY

**THE SPECIFIC GRAVITY** of a solid or liquid substance is the ratio of the mass of a given volume of the substance to the mass of an equal volume of water at some standard temperature. The temperature commonly used in engineering work is 60° F. The temperature used by physicists is 4° C. In the metric system the specific gravity is equal to the density, *i.e.*, to the weight in grams of one cubic centimeter.

The specific gravity of gases is taken as the ratio of the masses of equal volumes of the given gas and of air or hydrogen, usually at 32° F. and 29.921 in. of mercury barometric pressure (0° C. and 760 mm. pressure). It is necessary to state which basis is used, *i.e.*, air = 1, or hydrogen = 1.

**Determination of Specific Gravity of Solids.**—Determine the weight of the body in air; then submerge it in water and determine its weight. If $W$ = weight in air, and $w$ = weight when submerged,

$$\text{Specific gravity} = W/(W - w).$$

If the body is lighter than water, attach to it a weight, $w_1$, sufficient to sink it. Then

$$\text{Specific gravity} = (W - w_1)/\{(W - w_1) - w\}.$$

Given the specific gravity, referred to water at 60° F.,

$$\text{Weight per cu. ft.} = \text{sp. gr.} \times 62.36.$$
$$\text{Weight per cu. in.} = \text{sp. gr.} \times 0.036088.$$
$$\text{Specific gravity} = \text{weight per cu. ft.} \times 0.016036.$$

**THE HYDROMETER.**—The hydrometer is an instrument for determining the density of liquids. It usually is made of glass, and consists of three parts: (1) the upper part, a graduated stem or fine tube of uniform diameter; (2) a bulb, or enlargement of the tube, containing air; and (3) a small bulb at the bottom, containing shot or mercury which causes the instrument to float in a vertical position. The graduations are figures representing either specific gravities, or the numbers of an arbitrary scale, as the Baumé, A.P.I., Twaddell or Beck.

**The Baumé Hydrometer Scale** is based on the following formulas:

For liquids lighter than water

$$\text{Specific gravity} = 140/(130 + \text{degrees Baumé})$$

For liquids heavier than water

$$\text{Specific gravity} = 145/(145 - \text{degrees Baumé})$$

The relation of the two Baumé scales to the specific gravity, together with the corresponding weights per U. S. gallon, is given in Table 4.

**The A.P.I. Hydrometer Scale** was adopted to avoid the confusion incident to two Baumé scales, and is recommended by the American Petroleum Institute, the U. S. Bureau of Mines and the U. S. Bureau of Standards for use in the petroleum and oil industries. It is based on the formula

$$\text{Degrees A.P.I.} = 141.5/(\text{Sp. gr. } 60°/60° \text{ F.}) - 131.5.$$

The relation of the A.P.I. scale to specific gravity and the corresponding weights per U. S. gallon are given in Table 5.

**Beck Hydrometer Scale.**—The relation between specific gravity and the Beck hydrometer scale is expressed by the formulas:

For liquids heavier than water, $\text{Sp. Gr.} = 170 \div (170 - \text{Beck degrees})$

For liquids lighter than water, $\text{Sp. Gr.} = 170 \div (170 + \text{Beck degrees})$

**Twaddell Hydrometer Scale.**—The relation between specific gravity and the Twaddell hydrometer scale, used only for liquids heavier than water, is expressed by the formula

$$\text{Sp. Gr.} = (5 \times \text{Twaddell degrees} + 1000) \div 1000.$$

**The Salinometer** is a hydrometer used for determining the density of brine solutions. The indications are 4 times as large as those on the corresponding Baumé scale.

**Table 4.—Specific Gravities at 60°/60° F. Corresponding to Degrees Baumé**

| Deg. Bé. | Liquids Lighter than Water | | Liquids Heavier than Water | | Deg. Bé. | Liquids Lighter than Water | | Liquids Heavier than Water | | Deg. Bé. | Liquids Lighter than Water | | Liquids Heavier than Water | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Sp. Gr. | Lb. per U. S. gal. | Sp. Gr. | Lb. per U. S. gal. | | Sp. Gr. | Lb. per U. S. gal. | Sp. Gr. | Lb. per U. S. gal. | | Sp. Gr. | Lb. per U. S. gal. | Sp. Gr. | Lb. per U. S. gal. |
| 0 | ...... | ..... | 1.0000 | 8.328 | 34 | 0.8537 | 7.108 | 1.3063 | 10.878 | 68 | 0.7071 | 5.886 | 1.8831 | 15.682 |
| 1 | ...... | ..... | 1.0069 | 8.385 | 35 | .8485 | 7.065 | 1.3182 | 10.977 | 69 | .7035 | 5.856 | 1.9079 | 15.888 |
| 2 | ...... | ..... | 1.0140 | 8.444 | 36 | .8434 | 7.022 | 1.3303 | 11.078 | 70 | .7000 | 5.827 | 1.9333 | 16.104 |
| 3 | ...... | ..... | 1.0211 | 8.503 | 37 | .8383 | 6.980 | 1.3426 | 11.180 | 71 | .6965 | 5.798 | 1.9595 | 16.317 |
| 4 | ...... | ..... | 1.0284 | 8.564 | 38 | .8333 | 6.939 | 1.3551 | 11.285 | 72 | .6931 | 5.769 | 1.9863 | 16.541 |
| 5 | ...... | ..... | 1.0357 | 8.625 | 39 | .8284 | 6.898 | 1.3679 | 11.391 | 73 | .6897 | 5.741 | 2.0139 | 16.771 |
| 6 | ...... | ..... | 1.0432 | 8.687 | 40 | .8235 | 6.857 | 1.3810 | 11.500 | 74 | .6863 | 5.712 | 2.0423 | 17.007 |
| 7 | ...... | ..... | 1.0507 | 8.750 | 41 | .8187 | 6.817 | 1.3942 | 11.610 | 75 | .6829 | 5.685 | 2.0714 | 17.250 |
| 8 | ...... | ..... | 1.0584 | 8.814 | 42 | .8140 | 6.777 | 1.4078 | 11.723 | 76 | .6796 | 5.657 | 2.1015 | 17.500 |
| 9 | ...... | ..... | 1.0662 | 8.879 | 43 | .8092 | 6.738 | 1.4216 | 11.838 | 77 | .6763 | 5.629 | 2.1324 | 17.757 |
| 10 | 1.0000 | 8.328 | 1.0741 | 8.944 | 44 | .8046 | 6.699 | 1.4356 | 11.955 | 78 | .6731 | 5.602 | 2.1642 | 18.022 |
| 11 | 0.9929 | 8.269 | 1.0821 | 9.011 | 45 | .8000 | 6.661 | 1.4500 | 12.075 | 79 | .6699 | 5.576 | 2.1970 | 18.295 |
| 12 | .9859 | 8.211 | 1.0902 | 9.079 | 46 | .7955 | 6.623 | 1.4647 | 12.197 | 80 | .6667 | 5.549 | 2.2137 | 18.435 |
| 13 | .9790 | 8.153 | 1.0985 | 9.148 | 47 | .7910 | 6.586 | 1.4796 | 12.321 | 81 | .6635 | 5.522 | 2.2656 | 18.867 |
| 14 | .9722 | 8.096 | 1.1069 | 9.217 | 48 | .7865 | 6.548 | 1.4948 | 12.448 | 82 | .6604 | 5.497 | 2.3016 | 19.166 |
| 15 | .9655 | 8.041 | 1.1154 | 9.288 | 49 | .7821 | 6.511 | 1.5104 | 12.578 | 83 | .6573 | 5.471 | 2.3387 | 19.476 |
| 16 | .9589 | 7.986 | 1.1240 | 9.360 | 50 | .7778 | 6.476 | 1.5263 | 12.710 | 84 | .6542 | 5.445 | 2.3771 | 19.795 |
| 17 | .9524 | 7.931 | 1.1328 | 9.433 | 51 | .7735 | 6.440 | 1.5426 | 12.846 | 85 | .6512 | 5.420 | 2.4167 | 20.125 |
| 18 | .9459 | 7.877 | 1.1417 | 9.508 | 52 | .7692 | 6.404 | 1.5591 | 12.984 | 86 | .6482 | 5.395 | 2.4576 | 20.466 |
| 19 | .9396 | 7.825 | 1.1508 | 9.583 | 53 | .7650 | 6.369 | 1.5761 | 13.125 | 87 | .6452 | 5.370 | 2.5000 | 20.819 |
| 20 | .9333 | 7.772 | 1.1600 | 9.660 | 54 | .7609 | 6.334 | 1.5934 | 13.269 | 88 | .6422 | 5.345 | 2.5439 | 21.184 |
| 21 | .9272 | 7.721 | 1.1694 | 9.738 | 55 | .7568 | 6.300 | 1.6111 | 13.417 | 89 | .6393 | 5.320 | 2.5893 | 21.562 |
| 22 | .9211 | 7.670 | 1.1789 | 9.817 | 56 | .7527 | 6.266 | 1.6292 | 13.567 | 90 | .6364 | 5.296 | 2.6364 | 21.954 |
| 23 | .9150 | 7.620 | 1.1885 | 9.897 | 57 | .7487 | 6.233 | 1.6477 | 13.721 | 91 | .6335 | 5.272 | 2.6852 | 22.361 |
| 24 | .9091 | 7.570 | 1.1983 | 9.979 | 58 | .7447 | 6.199 | 1.6667 | 13.879 | 92 | .6306 | 5.248 | 2.7359 | 22.783 |
| 25 | .9032 | 7.522 | 1.2083 | 10.062 | 59 | .7407 | 6.166 | 1.6860 | 14.041 | 93 | .6278 | 5.225 | 2.7885 | 23.221 |
| 26 | .8974 | 7.473 | 1.2185 | 10.147 | 60 | .7368 | 6.134 | 1.7059 | 14.206 | 94 | .6250 | 5.201 | 2.8431 | 23.676 |
| 27 | .8917 | 7.425 | 1.2288 | 10.233 | 61 | .7330 | 6.102 | 1.7262 | 14.375 | 95 | .6222 | 5.178 | 2.9000 | 24.150 |
| 28 | .8861 | 7.378 | 1.2393 | 10.320 | 62 | .7292 | 6.070 | 1.7470 | 14.548 | 96 | .6195 | 5.155 | 2.9592 | 24.643 |
| 29 | .8805 | 7.332 | 1.2500 | 10.409 | 63 | .7254 | 6.038 | 1.7683 | 14.725 | 97 | .6167 | 5.132 | 3.0208 | 25.156 |
| 30 | .8750 | 7.286 | 1.2609 | 10.500 | 64 | .7216 | 6.007 | 1.7901 | 14.907 | 98 | .6140 | 5.110 | 3.0851 | 25.691 |
| 31 | .8696 | 7.241 | 1.2719 | 10.592 | 65 | .7179 | 5.976 | 1.8125 | 15.094 | 99 | .6114 | 5.088 | 3.1522 | 26.250 |
| 32 | .8642 | 7.196 | 1.2832 | 10.686 | 66 | .7143 | 5.946 | 1.8354 | 15.285 | 100 | .6087 | 5.066 | 3.2222 | 26.833 |
| 33 | .8589 | 7.152 | 1.2946 | 10.781 | 67 | .7107 | 5.916 | 1.8590 | 15.481 | | | | | |

**Table 5.—Specific Gravities at 60°/60° F. Corresponding to Degrees A.P.I.**

| Deg. A.P.I. | Sp. Gr. | Lb. per U. S. Gal. | Deg. A.P.I. | Sp. Gr. | Lb. per U. S. Gal. | Deg. A.P.I. | Sp. Gr. | Lb. per U. S. Gal. | Deg. A.P.I. | Sp. Gr. | Lb. per U. S. Gal. | Deg. A.P.I. | Sp. Gr. | Lb. per U. S. Gal. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 1.0000 | 8.328 | 29 | 0.8816 | 7.341 | 47 | 0.7927 | 6.600 | 65 | 0.7201 | 5.994 | 83 | 0.6597 | 5.491 |
| 11 | 0.9930 | 8.270 | 30 | .8762 | 7.296 | 48 | .7883 | 6.563 | 66 | .7165 | 5.964 | 84 | .6566 | 5.465 |
| 12 | .9861 | 8.212 | 31 | .8708 | 7.251 | 49 | .7839 | 6.526 | 67 | .7128 | 5.934 | 85 | .6536 | 5.440 |
| 13 | .9792 | 8.155 | 32 | .8654 | 7.206 | 50 | .7796 | 6.490 | 68 | .7093 | 5.904 | 86 | .6506 | 5.415 |
| 14 | .9725 | 8.099 | 33 | .8602 | 7.163 | 51 | .7753 | 6.455 | 69 | .7057 | 5.874 | 87 | .6476 | 5.390 |
| 15 | .9659 | 8.044 | 34 | .8550 | 7.119 | 52 | .7711 | 6.420 | 70 | .7022 | 5.845 | 88 | .6446 | 5.365 |
| 16 | .9593 | 7.989 | 35 | .8498 | 7.076 | 53 | .7669 | 6.385 | 71 | .6988 | 5.817 | 89 | .6417 | 5.341 |
| 17 | .9529 | 7.935 | 36 | .8448 | 7.034 | 54 | .7628 | 6.350 | 72 | .6953 | 5.788 | 90 | .6388 | 5.316 |
| 18 | .9465 | 7.882 | 37 | .8398 | 6.993 | 55 | .7587 | 6.316 | 73 | .6919 | 5.759 | 91 | .6360 | 5.293 |
| 19 | .9402 | 7.830 | 38 | .8348 | 6.951 | 56 | .7547 | 6.283 | 74 | .6886 | 5.731 | 92 | .6331 | 5.269 |
| 20 | .9340 | 7.778 | 39 | .8299 | 6.910 | 57 | .7507 | 6.249 | 75 | .6852 | 5.703 | 93 | .6303 | 5.246 |
| 21 | .9279 | 7.727 | 40 | .8251 | 6.870 | 58 | .7467 | 6.216 | 76 | .6819 | 5.676 | 94 | .6275 | 5.222 |
| 22 | .9218 | 7.676 | 41 | .8203 | 6.830 | 59 | .7428 | 6.184 | 77 | .6787 | 5.649 | 95 | .6247 | 5.199 |
| 23 | .9159 | 7.627 | 42 | .8155 | 6.790 | 60 | .7389 | 6.151 | 78 | .6754 | 5.622 | 96 | .6220 | 5.176 |
| 24 | .9100 | 7.578 | 43 | .8109 | 6.752 | 61 | .7351 | 6.119 | 79 | .6722 | 5.595 | 97 | .6193 | 5.154 |
| 25 | .9042 | 7.529 | 44 | .8063 | 6.713 | 62 | .7313 | 6.087 | 80 | .6690 | 5.568 | 98 | .6166 | 5.131 |
| 26 | .8984 | 7.481 | 45 | .8017 | 6.675 | 63 | .7275 | 6.056 | 81 | .6659 | 5.542 | 99 | .6139 | 5.109 |
| 27 | .8927 | 7.434 | 46 | .7972 | 6.637 | 64 | .7238 | 6.025 | 82 | .6628 | 5.516 | 100 | .6112 | 5.086 |
| 28 | .8871 | 7.387 | | | | | | | | | | | | |

**SPECIFIC GRAVITY OF VARIOUS SUBSTANCES.**—The specific gravity of the useful metals is given in Table 3. Table 6 gives the specific gravity of various solids, except the metals and woods, and Tables 7 and 8 give the specific gravity of liquids and gases respectively. For specific gravity of the various woods, see p. 5–21.

### Table 6.—Specific Gravity and Weight of Solid Substances, Except Metals and Wood

| Substance | Specific Gravity | Average Weight, lb. per cu. ft. | Substance | Specific Gravity | Average Weight, lb. per cu. ft. |
|---|---|---|---|---|---|
| Asbestos | 2.1–2.8 | 153 | Gypsum | 2.08–2.4 | 140 |
| Ashes | ....... | 43 | Hornblende | 3.2–3.52 | 210 |
| Asphaltum | 1.39 | 87 | Ice | 0.88–0.92 | 55–57 |
| Barytes | 4.50 | 281 | Leather | 0.86–1.02 | 59 |
| Basalt | 2.7–3.2 | 184 | Lime, quick, in bulk | 0.8–0.96 | 55 |
| Bauxite | 2.55 | 159 | Limestone | 2.3–2.9 | 160 |
| Bluestone | 2.2–2.5 | 147 | Magnesia, carbonate | 2.4 | 150 |
| Brick, soft | 1.6 | 100 | Magnesite | 3.0 | 187 |
| " common | 1.79 | 112 | Marble | 2.56–2.88 | 170 |
| " hard | 2.0 | 125 | Masonry, dry, rubble | 2.24–2.56 | 150 |
| " pressed | 2.16 | 135 | " dressed | 2.24–2.88 | 160 |
| " fire | 2.24–2.4 | 145 | Mica | 2.80 | 175 |
| " sand-lime | 2.18 | 136 | Mortar | 1.44–1.6 | 95 |
| Brickwork, mortar | 1.6 | 100 | Mud | 1.7–1.8 | 111 |
| " cement | 1.79 | 112 | Paper | 0.70–1.15 | 58 |
| Borax | 1.7–1.8 | 109 | Paraffin | 0.87–0.91 | 56 |
| Cement, American, natural | 2.8–3.2 | 187 | Peat | 0.65–0.85 | 47 |
| " portland | 3.05–3.15 | 190 | Phosphate rock | 3.2 | 200 |
| " " loose | 1.44 | 92 | Pitch | 1.15 | 72 |
| " " barreled | 1.84 | 115 | Plaster-of-Paris | 1.5–1.8 | 103 |
| " slag | 1.9–2.3 | 130 | Porcelain | 2.3–2.5 | 156 |
| Chalk | 1.8–2.6 | 137 | Porphyry | 2.6–2.9 | 172 |
| Clay | 1.92–2.4 | 137 | Pumice | 0.37–0.90 | 40 |
| Coal, anthracite | 1.4–1.8 | 97 | Riprap, limestone | 1.3–1.4 | 80–85 |
| " bituminous | 1.2–1.5 | 84 | " sandstone | 1.4 | 90 |
| " lignite | 1.1–1.4 | 78 | " shale | 1.7 | 105 |
| " charcoal | 0.27–0.58 | 18 | Rubber, caoutchouc | 0.92–0.96 | 59 |
| Coke | 0.27 | 30.8 | " manufactured | 1.0–2.0 | 95 |
| Concrete | 1.92–2.48 | 144 | Salt | 0.78–1.25 | 49–78 |
| Cork | 0.22–0.26 | 15 | Saltpeter | 1.07 | 67 |
| Dolomite | 2.9 | 181 | Sand | 1.44–1.76 | 100 |
| Earth, dry, loose | 1.2 | 75 | " wet | 1.89–2.07 | 125 |
| " " packed | 1.5 | 93 | Sandstone | 2.24–2.4 | 145 |
| " moist, loose | 1.3 | 81 | Serpentine | 2.4–2.7 | 165 |
| " " packed | 1.6 | 100 | Shale | 2.6–2.9 | 172 |
| " mud, flowing | 1.7 | 106 | Slag, bank | 1.1–1.2 | 69 |
| " " packed | 1.8 | 112 | " " screenings | 1.5–1.9 | 107 |
| Emery | 4.0 | 250 | " machine | 1.5 | 96 |
| Feldspar | 2.5–2.6 | 159 | " sand | 0.8–0.9 | 53 |
| Glass, common | 2.5–2.75 | 164 | Slate | 2.72–2.88 | 175 |
| " crystal | 2.90–3.00 | 184 | Soapstone | 2.65–2.8 | 170 |
| " flint | 3.2–4.7 | 188 | Starch | 1.53 | 96 |
| " plate | 2.45–2.72 | 161 | Stone, various | 2.16–3.4 | 135–200 |
| Gneiss | 2.4–2.7 | 165 | " crushed | 1.6 | 100 |
| Granite | 2.5–3.1 | 179 | Sulphur | 1.93–2.07 | 125 |
| Graphite | 1.9–2.3 | 126 | Talc | 2.6–2.8 | 169 |
| Gravel, dry, loose | 1.4–1.7 | 90–105 | Tar, bituminous | 1.20 | 75 |
| " " packed | 1.6–1.9 | 100–120 | Terra-cotta | 1.9 | 119 |
| " wet | 1.9 | 120 | Tile | 1.76–1.92 | 115 |
| Greenstone | 2.8–3.2 | 187 | Trap rock | 2.72–3.4 | 185 |
| | | | Wool | 1.32 | 82 |

### Table 7.—Specific Gravity of Liquids

Water at 39.1° F. (Sp. gr. = 1.000) weighs 62.43 lb. per cu. ft.

| Liquid | Specific Gravity | Liquid | Specific Gravity | Liquid | Specific Gravity |
|---|---|---|---|---|---|
| Acid, acetic | 1.06 | Carbon disulphide | 1.293 | Oil, neatsfoot | 0.913–0.917 |
| " carbolic | 0.96 | Chloroform | 1.480 | " olive | .918 |
| " fluoric | 1.50 | Ether | 0.736 | " palm | .905 |
| " muriatic | 1.20 | Gasoline | 0.66–0.75 | " petroleum | .78–0.88 |
| " nitric | 1.54 | Glycerine | 1.260 | " rape | .92 |
| " phosphoric | 1.56 | Kerosene | 0.78–0.82 | " resin | .955 |
| " sulphuric, 87% | 1.80 | Lye, soda, 66% | 1.70 | " sperm | .88 |
| Alcohol, ethyl, 100% | 0.807 | Naphtha | 0.76 | " turpentine | .873 |
| " methyl, 100% | 0.810 | Oil, cottonseed | .93 | " whale | .92 |
| Ammonia, 27.9% | 0.891 | " lard | .92 | Water, 39.1° F. | 1.00 |
| Benzine | 0.69 | " linseed, boiled | .942 | " 212° F. | 0.958 |
| Bromine | 3.187 | " mineral | .92 | " sea | 1.02–1.03 |

Table 8.—Specific Gravity and Physical Properties of Gases

| Gas | Chemical Symbol | Molecular Weight | Specific Gravity or Density at 32° F. and Atmospheric Pressure | | Wt. and Vol. at 32° F. and Atmospheric Pressure | | Specific Heat at Constant Pressure | $\frac{C_p}{C_v}$ * |
|---|---|---|---|---|---|---|---|---|
| | | | H = 1 | Air = 1 | Lb. per cu. ft. | Cu. ft. per lb. | | |
| Acetylene | $C_2H_2$ | 26.0156 | 12.909 | 0.8987 | 0.07255 | 13.784 | 0.350 | 1.29 |
| Air | | 28.9521 | 14.364 | 1.000 | .080728 | 12.387 | 0.238 | 1.40 |
| Ammonia | $NH_3$ | 17.0314 | 8.490 | 0.5911 | .04772 | 20.956 | 0.520 | 1.32 |
| Argon | A | 39.9940 | 19.842 | 1.3814 | .11518 | 8.682 | 0.124 | 1.66 |
| Chlorine | Cl | 70.9140 | 35.133 | 2.4460 | .19746 | 5.064 | 0.121 | 1.32 |
| Carbon dioxide | $CO_2$ | 44.0000 | 21.830 | 1.5198 | .12269 | 8.151 | 0.217 | 1.30 |
| Carbon monoxide | CO | 28.0000 | 13.892 | 0.9672 | .07808 | 12.807 | 0.243 | 1.40 |
| Ethylene | $C_2H_4$ | 28.0312 | 13.907 | 0.9682 | .07816 | 12.794 | 0.404 | 1.21 |
| Helium | He | 4.002 | 1.986 | 0.1383 | .01116 | 89.606 | 1.25 | 1.66 |
| Hydrochloric acid | HCl | 36.4648 | 18.091 | 1.3152 | .10617 | 9.419 | 0.191 | 1.40 |
| Hydrogen | H | 2.0156 | 1.000 | 0.0696 | .00562 | 177.936 | 3.400 | 1.41 |
| Methane | $CH_4$ | 16.0317 | 7.954 | 0.5538 | .04470 | 22.371 | 0.593 | 1.32 |
| Methyl chloride | $CH_3Cl$ | 50.4804 | 25.045 | 1.7436 | .14076 | 7.104 | 0.24 | 1.20 |
| Nitric oxide | NO | 30.008 | 14.888 | 1.0365 | .08367 | 11.939 | 0.231 | 1.40 |
| Nitrogen | N | 28.016 | 13.900 | 0.9677 | .07812 | 12.801 | 0.244 | 1.41 |
| Nitrous oxide | $N_2O$ | 44.016 | 21.838 | 1.5204 | .12274 | 8.147 | 0.226 | 1.31 |
| Oxygen | O | 32.0 | 15.876 | 1.1053 | .08923 | 11.207 | 0.217 | 1.40 |
| Sulphur dioxide | $SO_2$ | 64.06 | 31.782 | 2.2127 | .17863 | 5.598 | 0.154 | 1.25 |

* This column gives the ratio of specific heat at constant pressure to specific heat at constant volume.

## 4. HARDNESS

Hardness of a substance may be defined as its resistance to penetration by another substance. Several methods of determining hardness are in commercial use. These methods are described below.

**BRINELL'S METHOD.**—J. A. Brinell, a Swedish engineer, in 1900 published information about a method for determining the relative hardness of steel, which has come into extensive use. A hardened steel ball, 10 mm. diam. (0.3937 in.), is forced with a pressure of 3000 kg., or 500 kg. for soft metals, into a flat surface on the sample to be tested and allowed to remain on iron and steel for at least 10 seconds, and on other metals for at least 30 sec., to make a slight spherical indentation, whose diameter is measured with a microscope. The hardness is defined as the quotient of the pressure divided by the area of the indentation. From the measurement the *hardness number* is calculated by the equation

$$B = P/(\pi D/2)(D - \sqrt{D^2 - d^2}),$$

where $P$ = load = 3000 kg.; $D$ = diam. of ball = 10 mm.; $d$ = diam. of indentation, mm.; $B$ = Brinell hardness number. Table 3 gives the hardness numbers corresponding to the different values of $d$. A complete table of these numbers is given in A.S.T.M. Standards, 1933 E10–27, pp. 943–944.

There is a well defined approximate relationship between the Brinell numbers and the ultimate strength of the steel. Abbott (*Proc.* A.S.T.M. vol. xv, Pt. II, 1915), as the result of a large number of tests of different kinds of steel, proposed the equation $s_u = 0.70B - 26$, where $s_u$ = ultimate strength in thousands of lb. per sq. in.; $B$ = Brinell hardness number. For many steels, however, the strength is as well represented by the relation $s_u = 0.50B$.

**THE ROCKWELL METHOD** is similar to the Brinell method, in that it forces a spherical point into the metal to be tested and measures the penetration of the point. For metals of average hardness, as mild steel, the point is $^1/_{16}$ in. diam. and the load is 100 kg. (Rockwell B test); for harder metals a conical diamond point is used with a load of 150 kg. (Rockwell C test). The Rockwell hardness number is read directly from a numbered dial. One advantage of the Rockwell over the Brinell is that a smaller indentation is produced. The ultimate tensile strength of steel, $s_u$, is given in thousands of lb. per sq. in. by the equation $s_u = 3750/(130 - R_B)$, $R_B$ being the number read from the dial of the instrument (B test).

**THE VICKERS PYRAMID HARDNESS TESTING MACHINE** is a British instrument in which the principle employed is the same as that in the Brinell and Rockwell machines. The operation differs, in that the penetrator is a diamond in the form of a pyramid. By means of a microscope attachment, measurements are made of the distance across the diagonal of the indentation. One of the advantages of this machine is that it is suitable for any degree of hardness.

**THE SCLEROSCOPE,** invented by A. F. Shore, consists chiefly of a vertical glass tube in which slides freely a small cylinder of very hard steel, pointed on the lower end, called the hammer. This hammer is allowed to fall about 10 in. on the sample to be tested. The distance it rebounds is taken as a measure of the hardness of the sample. A scale on the tube is divided into 140 equal parts, and the hardness is expressed as the number on the scale to which the hammer rebounds. Measured in this way the hardness of different substances is as follows: Glass, 130; porcelain, 120; hardest steel, 110; tool steel, 1% C, may be as low as 31; mild steel, 0.5 C, 26 to 30; gray iron castings, 39; wrought iron, 18; babbitt metal, 4 to 10; soft brass, 12; zinc, 8; copper, 6; lead, 2. (*Cass. Mag.*, Sept., 1908.) The relationship between the Scleroscope number and the ultimate tensile strength, $s_u$ is given by Abbott (*Proc.* A.S.T.M., vol. xv, Pt. II, 1915) by the equation $s_u = 4.0S - 15$, where $s_u$ is in thousands of lb. per sq. in. and $S$ is the Scleroscope number.

**RELATIONSHIP BETWEEN DIFFERENT HARDNESS NUMBERS.**—A rational relation exists between the Brinell and Rockwell numbers, and also between these numbers and the Vicker's Pyramid numbers; an empirical relation exists also between the foregoing numbers and those of the Shore scleroscope. Petrenko (*Tech. Paper* No. 334, U. S. Bureau of Standards) gives the following equations:

$$B = 25{,}000/(100 - R_C) \text{ for } R_C > 40;$$

$$B = 1{,}420{,}000/(100 - R_C)^2 \text{ for } R_C \text{ from } -20 \text{ to } 40;$$

$$B = 7300/(130 - R_B) \text{ for } R_B \text{ from 35 to 100,}$$

where $B$ = Brinell number obtained with a 10 mm. ball and 3000-kg. load; $R_C$ = Rockwell number on the $C$ scale, using a 150-kg. load and a 120° diamond penetrator; $R_B$ = Rockwell number on the $B$ scale with a 100-kg. load and a $^1/_{16}$-in. ball.

Abbott (*Proc.* A.S.T.M., vol. xv, Pt, II, 1915) derives from tests the empirical relation $B = 5.5S - 28$, $B$ being the Brinell number and $S$ the scleroscope number. Table 9 gives these various hardness numbers in terms of each other. The values were obtained from the manufacturers of the instruments and from published experimental data. The scleroscope numbers, as tabulated, vary slightly from those given by the Sub-committee on Hardness Conversion of the A.S.S.T. in its 1930 handbook; the committee gives a scleroscope value of 81 for a Brinell number of 627 as compared with 84 in Table 9 (by interpolation), while the committee scleroscope number is 30 for a Brinell number of 197

**Table 9.—Conversion Table for Approximate Hardness Numbers**
(Compiled mainly from Manufacturers' Tables)

| Brinell 3000 kg. Load, 10-mm. Ball | | Rockwell Number | | Shore Scleroscope Number | Vicker's Pyramid Number | Brinell 3000 kg. Load, 10-mm. Ball | | Rockwell Number | | Shore Scleroscope Number | Vicker's Pyramid Number |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Diam. of Indentation, mm. | Hardness Number | *C* Scale, 150 kg. Load, 120° Diamond Cone | *B* Scale, 100 kg. Load, 1/16-in. Ball | | | Diam. of Indentation, mm. | Hardness Number | *C* Scale, 150 kg. Load, 120° Diamond Cone | *B* Scale, 100 kg. Load, 1/16-in. Ball | | |
| 2.40 | 653 | 62 | ...... | 86 | 783 | 4.60 | 170 | ...... | 87 | 24 | 170 |
| 2.60 | 555 | 55 | ...... | 75 | 622 | 4.70 | 163 | ...... | 85 | 23 | 163 |
| 2.80 | 477 | 49 | ...... | 66 | 513 | 4.80 | 156 | ...... | 82 | 23 | 156 |
| 3.00 | 415 | 44 | ...... | 58 | 439 | 4.90 | 149 | ...... | 80 | 22 | 149 |
| 3.10 | 388 | 41 | ...... | 54 | 404 | 5.00 | 143 | ...... | 78 | 21 | 143 |
| 3.20 | 363 | 39 | ...... | 51 | 374 | 5.10 | 137 | ...... | 75 | 20 | 137 |
| 3.30 | 341 | 37 | ...... | 48 | 352 | 5.20 | 131 | ...... | 73 | 19 | 131 |
| 3.40 | 321 | 35 | ...... | 45 | 329 | 5.30 | 126 | ...... | 70 | 18 | 126 |
| 3.50 | 302 | 32 | ...... | 42 | 303 | 5.40 | 121 | ...... | 68 | 17 | 121 |
| 3.60 | 285 | 30 | ...... | 40 | 285 | 5.50 | 116 | ...... | 65 | 16 | 116 |
| 3.70 | 269 | 27 | ...... | 37 | 269 | 5.60 | 111 | ...... | 62 | .... | 111 |
| 3.80 | 255 | 25 | ...... | 35 | 255 | 5.70 | 107 | ...... | 60 | .... | 107 |
| 3.90 | 241 | 23 | 99 | 33 | 241 | 5.80 | 103 | ...... | 57 | .... | 103 |
| 4.00 | 229 | 20 | 98 | 32 | 229 | 5.90 | 99.2 | ...... | 55 | .... | 99.2 |
| 4.10 | 217 | ...... | 96 | 30 | 217 | 6.00 | 95.5 | ...... | 52 | .... | 95.5 |
| 4.20 | 207 | ...... | 95 | 29 | 207 | 6.10 | 92.0 | ...... | 49 | .... | 92.0 |
| 4.30 | 197 | ...... | 93 | 28 | 197 | 6.20 | 88.7 | ...... | 47 | .... | 88.7 |
| 4.40 | 187 | ...... | 91 | 27 | 187 | 6.30 | 85.5 | ...... | 44 | .... | 85.5 |
| 4.50 | 179 | ...... | 89 | 25 | 179 | 6.40 | 82.5 | ...... | 42 | .... | 82.5 |

as compared with 28 in Table 9. Both lists are the same for scleroscope number 54 which equals Brinell number 387.

**MOH'S SCALE OF HARDNESS** is a scratch hardness scale. It consists of a series of minerals, each of which can be scratched by the mineral next above it in the scale. The differences in the original Moh scale are not uniform and the intervals between successive numbers increase rapidly as the hardness number increases. The step from 9 to 10 probably is as great as from 1 to 9. Carborundum frequently is given 9.6, being, therefore, much harder than corundum but not so hard as a diamond. In order to make a more uniform scale, a modification of the Moh scale was proposed by Ridgeway, Ballard and Bailey (*Trans.* Am. Electrochem. Soc. vol. lxiii, p. 392, 1933), wherein industrial minerals are included, and which subdivided the wide spaces between 8 and 9, and 9 and 10 in the Moh scale. The original Moh scale and the modified scale, are as follows:

| Original | Modified | Original | Modified | Original | Modified |
|---|---|---|---|---|---|
| 1. Talc | 1. Talc | 6. Orthoclase | 6. Orthoclase or Periclase | | 10. Garnet |
| 2. Gypsum | 2. Gypsum | | | | 11. Fused Zirconia |
| 3. Calc spar | 3. Calc spar | | 7. Vitreous Pure Silica | 9. Sapphire | 12. Fused Alumina |
| 4. Fluor spar | 4. Fluor spar | | | | 13. Silicon Carbide |
| 5. Apatite | 5. Apatite | 7. Quartz | 8. Quartz | | 14. Boron Carbide |
| | | 8. Topaz | 9. Topaz | 10. Diamond | 15. Diamond |

Certain metals have almost the same behavior in the scratch test as some of the minerals in the modified scale. These are, with their scale numbers: 8, Stellite; 11, Tantalum carbide; 12, Tungsten carbide.

Table 10 gives the relative hardness of various materials, according to the Moh scale. Table 11 gives the hardness of certain chemical elements according to the same scale.

**Table 10.—Hardness of Various Materials by Moh Scale**
(Smithsonian Tables)

| Material | Hardness | Material | Hardness | Material | Hardness | Material | Hardness |
|---|---|---|---|---|---|---|---|
| Agate | 7.0 | Bismuth | 2.5 | Gypsum | 1.6–2.0 | Plat-iridium | 6.5 |
| Alabaster | 1.7 | Boric acid | 3.0 | Hematite | 6.0 | Pyrite | 6.3 |
| Alum | 2.0–2.5 | Brass | 3.0–4.0 | Hornblende | 5.5 | Quartz | 7.0 |
| Aluminum | 2.0 | Calimine | 5.0 | Iridium | 6.0 | Rock-salt | 2.0 |
| Amber | 2.0–2.5 | Calcite | 3.0 | Iridosmium | 7.0 | Ross metal | 2.5–3.0 |
| Andalusite | 7.5 | Copper | 2.5–3.0 | Iron | 4.0–5.0 | Silver chloride | 1.3 |
| Anthracite | 2.2 | Corundum | 9.0 | Kaolin | 1.0 | Sulphur | 1.5–2.5 |
| Antimony | 3.3 | Diamond | 10.0 | Loess (0° C.) | 0.3 | Stibnite | 2.0 |
| Apatite | 5.0 | Dolomite | 3.5–4.0 | Magnetite | 6.0 | Serpentine | 3.0–4.0 |
| Aragonite | 3.5 | Feldspar | 6.0 | Marble | 3.0–4.0 | Silver | 2.5–3.0 |
| Arsenic | 3.5 | Flint | 7.0 | Meerschaum | 2.0–3.0 | Steel | 5.0–8.5 |
| Asbestos | 5.0 | Fluorite | 4.0 | Mica | 2.8 | Talc | 1.0 |
| Asphalt | 1.0–2.0 | Galena | 2.5 | Opal | 4.0–6.0 | Tin | 1.5 |
| Augite | 6.0 | Garnet | 7.0 | Orthoclase | 6.0 | Topaz | 8.0 |
| Barite | 3.3 | Glass | 4.5–6.5 | Palladium | 4.8 | Tourmaline | 7.3 |
| Beryl | 7.8 | Gold | 2.5–3.0 | Phosphor-bronze | 4.0 | Wax (0° C.) | 0.2 |
| Bell-metal | 4.0 | Graphite | 0.5–1.0 | Platinum | 4.3 | Wood's metal | 3.0 |

## 5. COMPRESSIBILITY

**COMPRESSIBILITY OF LIQUIDS.**—The coefficient of compression of a liquid is $\beta = (1/V_1) \times \{(V_1 - V_2)/(p_2 - p_1)\}$ where $V_1$ and $V_2$ are the volumes at the same temperature, deg. C., under pressures $p_1$ and $p_2$ (atmospheres) respectively. Values of $\beta \times 10^6$ are given in Table 11.

**Table 11.—Compressibility of Liquids**
(From Smithsonian Tables)

| Substance | $t$, deg. C. | Pressure, Atmospheres | $\beta \times 10^6$ | Substance | $t$, deg. C. | Pressure, Atmospheres | $\beta \times 10^6$ |
|---|---|---|---|---|---|---|---|
| Chloroform | 20 | ........ | 128 | Water | 20 | 1–25 | 49.1 |
| Alcohol, ethyl | 20 | 1–50 | 112 | “ | 20 | 25–50 | 47.6 |
| “ methyl | 18.1 | 8 | 120 | “ | 20 | 1–100 | 46.8 |
| Glycerine | 20.5 | ........ | 25 | “ | 20 | 100–200 | 44.2 |
| Mercury | 0 | ........ | 3.90 | “ | 20.4 | 1–500 | 43.4 |
| Nitric acid | 20.3 | 1–32 | 338 | “ | 0 | 500–1000 | 41.6 |
| Oil, olive | 20.5 | ........ | 63 | “ | 0 | 1000–1500 | 35.8 |
| “ petroleum | 16.5 | ........ | 70 | “ | 0 | 1500–2000 | 32.4 |
| “ rape | 20.3 | ........ | 60 | “ | 0 | 2000–2500 | 29.2 |
| “ turpentine | 19.7 | ........ | 79 | “ | 0 | 2500–3000 | 26.1 |

**COMPRESSIBILITY OF GASES.**—Table 12 gives the relative change in volume of several gases under various pressures and at various temperatures. For additional data on the change of volume of the following gases, see Kent's Mechanical Engineers' Handbook—Power, forming Vol. 2 of this series: Air, pp. 1–02 to 1–36; ammonia and sulphur dioxide, pp. 10–14 to 10–20; steam, pp. 503 to 515.

### Table 12.—Compressibility of Gases

Relative volume at various temperatures and pressures. Volume at 0° C. and 1 atmosphere pressure = 1,000,000
(Based on Smithsonian Tables)

| Pressure, Atmospheres | Oxygen | | | Air | | | Nitrogen | | |
|---|---|---|---|---|---|---|---|---|---|
| | 0° C. | 99.5° C. | 199.5° C. | 0° C. | 99.4° C. | 200.4° C. | 0° C. | 99.5° C. | 199.6° C. |
| 100 | 9265 | ...... | ...... | 9730 | ...... | ...... | 9910 | ...... | ...... |
| 200 | 4570 | 7000 | 9095 | 5050 | 7,360 | 9,430 | 5195 | 7,445 | 9,532 |
| 300 | 3208 | 4843 | 6283 | 3658 | 5,170 | 6,622 | 3786 | 5,301 | 6,715 |
| 500 | 2312 | 3244 | 4100 | 2680 | 3,565 | 4,422 | 2780 | 3,655 | 4,515 |
| 1000 | 1735 | 2151 | ...... | 1992 | 2,415 | 2,828 | 2068 | ...... | ...... |
| | Hydrogen | | | Ethylene | | | Carbon Dioxide | | |
| | 0° C. | 99.3° C. | 200.5° C. | 0° C. | 100° C. | 198.5° C. | 0° C. | 100° C. | 198° C. |
| 50 | ...... | ...... | ...... | 3520 | 23,840 | 33,040 | 2100 | 24,120 | ...... |
| 100 | ...... | ...... | ...... | 3100 | 10,050 | 15,800 | 2020 | 10,300 | 12,582 |
| 200 | 5690 | 7567 | 9420 | 2875 | 4,730 | 7,685 | ...... | ...... | ...... |
| 300 | 4030 | 5286 | 6520 | 2686 | 3,770 | 5,426 | 1863 | 2,966 | 4,976 |
| 500 | 2713 | 3462 | 4210 | 2512 | 3,156 | 3,970 | 1782 | 2,402 | 3,356 |
| 1000 | 1720 | 2093 | ...... | 2289 | 2,643 | ...... | 1656 | 1,999 | ...... |

**COMPRESSIBILITY OF SOLIDS.**—Table 13 gives the compressibility of the more important solid elements. The values are the mean elastic change in volume per megabar (0.097 atmosphere) between 100 and 500 megabars.

### Table 13.—Compressibility of the Solid Chemical Elements

Arranged in increasing order of atomic weights. (From Smithsonian Tables)

| Element | Change in volume per megabar pressure * | Element | Change in volume per megabar pressure * | Element | Change in volume per megabar pressure * |
|---|---|---|---|---|---|
| Lithium......... | $0.0_488$ | Iron............. | $0.0_540$ | Cadmium...... | $0.0_419$ |
| Carbon.......... | $.0_55$ | Nickel........... | $.0_527$ | Tin............ | $.0_416$ |
| Sodium......... | $.0_3154$ | Copper........... | $.0_554$ | Antimony...... | $.0_422$ |
| Magnesium...... | $.0_427$ | Zinc............. | $.0_415$ | Iodine......... | $.0_313$ |
| Aluminum....... | $.0_413$ | Arsenic.......... | $.0_443$ | Cesium......... | $.0_361$ |
| Silicon.......... | $.0_516$ | Selenium......... | $.0_3118$ | Platinum....... | $.0_521$ |
| Red phosphorus.. | $.0_490$ | Bromine.......... | $.0_3518$ | Gold........... | $.0_547$ |
| Sulphur......... | $.0_3125$ | Rubidium........ | $.0_340$ | Mercury....... | $.0_4371$ |
| Potassium....... | $.0_3315$ | Molybdenum..... | $.0_526$ | Thallium....... | $.0_426$ |
| Calcium........ | $.0_455$ | Palladium........ | $.0_538$ | Lead........... | $.0_422$ |
| Chromium....... | $.0_57$ | Silver............ | $.0_584$ | Bismuth........ | $.0_428$ |
| Manganese...... | $.0_57$ | | | | |

* The subscript indicates the number of zeros after the decimal point. Thus, $0.0_488 = 0.000088$.

# Section 2

## IRON AND STEEL

By Bradley Stoughton

## WELDING

By J. C. Hodge and R. E. Hiller

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# IRON AND STEEL

**By Bradley Stoughton**

Iron is a chemical element, which in its pure form has almost no commercial use. Iron which has a purity of 98 to 99% is used commercially for certain special purposes. See p. 2–06. The addition of other elements to iron, as carbon, silicon, manganese, nickel, etc., completely changes its character and renders it the most useful metal known to man.

## 1. PIG IRON

Iron is smelted from the ore in the blast furnace, the resulting product being known as pig iron. Analyses of various grades of pig iron are given in Table 1.

**Foundry Pig Iron** is pig iron intended for foundry uses. The silicon content is high, and the sulphur content usually is low.

**Charcoal Pig Iron** is pig iron made in blast furnaces using charcoal as fuel. Sulphur content does not exceed 0.05%, and silicon also is low. Charcoal iron is preferred for chilled castings, low in sulphur, such as chilled car wheels, chilled iron rolls, etc.

**Acid Pig Iron** is a trade name for pig iron suitable for use in the Bessemer converter and the acid open-hearth furnace, for steel making. Phosphorus content must be low.

**Basic Pig Iron** is a trade name for pig iron of phosphorus content higher than that of acid pig iron, and silicon content lower than that of either acid or foundry pig iron. It usually contains 1.75–2% manganese.

**Forge Pig Iron** is pig iron suitable for use in the puddling furnace.

**Electric Pig Iron** is pig iron made directly from ore in the electric furnace.

## 2. CAST IRON

Pig iron, or iron direct from the blast furnace, ordinarily is unsuitable for iron castings without the admixture of other elements. Castings may be made with metal direct from the blast furnace, provided sufficient metal, melted in a cupola, and containing the necessary alloying elements to correct the composition of the blast furnace metal, is added to the latter. The usual practice of making iron castings is to remelt the pig iron, together

**Table 1.—Analyses of Pig Irons or Cast Irons**

Figures in bold-faced type indicate essential constituents; figures in light-faced type are approximately correct, but are not the determining factor for the grade in question.

| Trade Name | Total C, percent | Si, percent | S, percent | P, percent | Mn, percent |
|---|---|---|---|---|---|
| No. 1 soft | 3.00 ± | **2.75 to 3.25** | **0.05 and under** | 0.30 to 1.50 | 0.10 to 1.00 |
| No. 1 foundry | 3.25 ± | **2.25 to 2.75** | **0.05 and under** | 0.30 to 1.50 | 0.10 to 1.00 |
| No. 2 foundry | 3.50 ± | **1.75 to 2.25** | **0.06 and under** | 0.30 to 1.50 | 0.10 to 1.00 |
| No. 3 foundry | 3.75 ± | **1.25 to 1.75** | **0.065 and under** | 0.30 to 1.50 | 0.10 to 1.00 |
| Gray forge | 3.50 ± | **0.75 to 1.75** | **0.07 and under** | Under 1.00 | 0.10 to 1.00 |
| Standard acid Bessemer pig | 3.50 to 4.00 | **1.00 to 1.50** | **Under 0.08** | **Under 0.09** | 0.20 to 1.00 |
| Basic pig | 3.50 to 4.00 | **Under 1.25** | Under 0.08 | 0.10 to 1.50 | **1.50 to 2.00** |
| Malleable Bessemer | 3.50 ± | 0.75 to 2.00 | Under 0.07 | Under 0.20 | 0.50 ± |
| Ferrosilicon* | 0.50 to 2.00 | **10.00 to 50.00** | Under 0.04 | Under 0.10 | 0.20 ± |
| Silicospiegel | 1.00 ± | **5.00 to 15.00** | Under 0.02 | Under 0.10 | **15.00 to 25.00** |
| Ferromanganese | 6.00 to 7.00 | 0.50 to 1.00 | Under 0.03 | .......... | **40.00 to 80.00** |
| Ferrophosphorus | 1.00 ± | 1.50 ± | Under 0.05 | **10.00 to 25.00** | .......... |
| Spiegeleisen | 4.50 to 6.00 | 1.00 ± | Under 0.04 | .......... | **15.00 to 35.00** |

* Ferrosilicon of 10 percent grade, ferromanganese, silicospiegel, and spiegeleisen are made in blast furnaces, by variations of the usual practice and by using special ores. Ferrosilicon of 30 to 80 percent grades, and several other so-called *ferroalloys*, are made in electric furnaces. These include ferrochrome, ferrotungsten, ferromolybdenum, ferrotitanium, ferrovanadium, etc. They are all practically pig irons with large amounts of the alloying metal. When these alloys are free from, or low in, carbon, they are made by the thermit and other processes.

with certain proportions of scrap, in the cupola. The various grades of pig iron and scrap are combined in the cupola charges to give the desired chemical composition of the iron in the castings. For methods of mixing and charging, see p. 20–10. The composition of the iron in the castings also may be varied by the addition of ferroalloys, as ferrosilicon, ferromanganese, etc., to the molten iron in the ladle. Soda ash and calcium silicide also may be added to the iron in the ladle to reduce the sulphur and oxygen in the metal.

**EFFECT OF VARIOUS ELEMENTS ON CAST IRON.**—**Carbon** depends on the presence of other elements. High manganese content may permit a carbon content of 7%, while high silicon may reduce carbon to 1%. Carbon may be present either in the combined form or as free graphitic carbon. Maximum tensile strength is attained when combined carbon is about 0.47%. Maximum transverse strength requires about 0.70% combined carbon, and maximum crushing strength requires over 1%. Hardness increases with the increase in combined carbon.

**Silicon** directly affects the combined carbon in cast iron. Carbon exists principally as graphite when the silicon content ranges from 0.75–2.50%. With silicon in excess of about 3.0%, its effect on the formation of graphite decreases, and when silicon increases to 4 or 5% the iron tends to become white.

**Sulphur** acts in the opposite manner to silicon, but with much greater effect. A sulphur content of 0.01% will counteract the effect of 1.15% silicon in producing graphite. Sulphur causes segregation and red shortness, and should be kept below 0.5%.

**Phosphorus** increases the fusibility, fluidity and brittleness of cast iron. It decreases shrinkage and density. Ornamental castings which must reproduce sharp impressions may contain as much as 1.5% phosphorus.

**Manganese** lessens the bad effect of sulphur, and should be present in quantities twice as great as the quantity of sulphur. It tends to make cast iron harder and more brittle. It also tends to strengthen the casting and fine the grain. It is introduced into the iron by additions of spiegeleisen or ferromanganese in the ladle.

**Nickel, Chromium, Molybdenum** are added to cast iron to increase the tensile strength. The proportions used have ranged up to 1.50% nickel, 0.50% chromium, or 0.50–1.50% molybdenum. Tensile strength of cast iron containing these elements may range as high as 70,000 lb. per sq. in.

**PHYSICAL PROPERTIES OF CAST IRON.**—Detailed specifications for gray iron castings and a discussion of the physical, electrical and thermal properties of cast iron are given in the section on Foundry Practice. See p. 20–03.

## 3. WROUGHT IRON

Wrought iron is commercial iron mechanically mixed with slag. It is the product of the reverberatory furnace or of the charcoal hearth. It melts at white heat, but is pasty at lower temperatures, and in this condition can be readily worked and welded. It is ductile when cold.

**GRADES.**—Wrought iron may be graded as: 1. Charcoal iron. 2. Puddle iron. 3. Busheled scrap. The first is the purest grade of wrought iron. The second is classified as staybolt iron (grade A) and merchant iron (grades B and C). The third grade is made from iron scrap, with which steel sometimes is mixed. It is irregular in quality.

**PROPERTIES OF WROUGHT IRON.**—Wrought iron is tough, ductile, and malleable. It can be welded readily. Its properties are only slightly changed by heat treatment. The following physical properties are given by different authorities. Specific gravity, 7.4–7.9; specific heat, 0.1138; melting point, 2732–2912° F.; coefficient of thermal expansion, 0.000 006 48; ultimate tensile strength, average 51,000 lb. per sq. in.; yield point, average, 31,000 lb. per sq. in.; elongation in 8 in., 25–40%; reduction in area, 40 55%; modulus of elasticity, 28,200,000; ultimate strength in single shear, 42,000 lb. per sq. in.; elastic limit in torsion, 20,500 lb. per sq. in.; modulus of elasticity in torsion, 12,800,000.

The ultimate compressive strength of good wrought iron is not well defined. The yield point in compression, which is about the same as the yield point in tension, should be taken as the ultimate compressive strength, for all practical purposes. The yield point ranges from 2000 to 4000 lb. per sq. in. higher than the elastic limit.

**SPECIFICATIONS FOR WROUGHT IRON.**—The Am. Soc. for Testing Materials has issued standard specifications for wrought-iron plate and bars. These are abstracted below from the 1936 Tentative* Standards.

**Wrought-iron Plates** (A.S.T.M. Designation A42–36T).—Plates shall be rolled from blooms, piles or slabs free from iron scrap or steel. Piles for plates are to be made of wrought iron. The chemical specification is Mn, max., 0.06%. Physical properties specified are: Tensile strength, min., 48,000 lb. per sq. in.; yield point, minimum, 27,000 lb. per sq. in.; elongation in 8 in., 14%. The tension test specimen should be made in the form shown in Fig. 2 (see p. 7–54), and be of full thickness of the plate as rolled, except that for plates 1 $^1/_2$ in. or more thick, specimen may be machined to a minimum thickness or diameter of $^3/_4$ in. for a distance of at least 9 in.

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* A Tentative Standard is subject to yearly revision. The latest issue of the A.S.T.M. Standards should be consulted.

The bend test consists in bending a specimen cold through 90° around a pin, without cracking on the outside of the bent portion. Diameter of pin = (40 $t$/specified longitudinal elongation in 8 in.), where $t$ = thickness of plate, in.

A nick test shall be made. The specimen when nicked on one side and broken shall show a wholly fibrous fracture.

In *Special Forming Plate* the longitudinal strength and ductility are related directly to the transverse properties and are scaled downward in ratio to the increase in transverse properties as follows:

$$\text{Tensile strength} = 51{,}000 - (1500\ \text{T.D.})$$

Longitudinal elongation in 8 in. = 16 − T.D., where T.D. = transverse ductility of plate expressed as a percentage of elongation in 8 in. Yield point = 27,000 lb. per sq. in. Representative requirements for special forming plates in accordance with the above formulas are

| | | | | | | |
|---|---|---|---|---|---|---|
| Transverse ductility, % | 3 | 4 | 5 | 6 | 7 | 8 |
| Minimum tensile strength (longitudinal), lb. per sq. in.* | 46,500 | 45,000 | 43,500 | 42,000 | 40,500 | 39,000 |
| Elongation in 8 in., %† | 13 | 12 | 11 | 10 | 9 | 8 |

Permissible variation in weight of plates ordered to weight per sq. ft. shall not exceed the percentages given in Table 7, p. 2–28. If ordered by thickness, the thickness shall not vary by more than 0.01 in. under that ordered. The overweight of each lot in each shipment shall not exceed the amount given in Table 7, p. 2–28.

**Wrought-iron Bars, Single and Double Refined** (A.S.T.M. Designation A189–36T).—Bars shall be made from pig-puddled or processed wrought iron, free from any admixture of steel, in two grades, as follows: Grade A, original muck bars twice piled and rerolled; Grade B, original muck bars piled and rerolled. Manganese shall not exceed 0.06%. Physical properties are given in Table 2.

The cold bend test comprises bending cold around a pin through 180 deg. without showing cracks on the outside of the bent portion. The diameter of the pin shall be equal to the thickness of the specimen. The hot bend test consists in bending the specimen flat on itself at a temperature of 1700–1800° F. without showing cracks on the outside of the bent portion. The nick test comprises nicking the bar 25% around its circumference with a tool having a 60-deg. edge, to a depth of not less than 16% of the diameter or thickness of the bar, and breaking the bar slowly. The fracture should be wholly fibrous in appearance.

Tension test specimens shall be the full section of the material as rolled, if possible, or

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* For plates over $^3/_4$ in. thick, a deduction of 1000 lb. per sq. in. from the indicated tensile strength is allowed for each increase of $^1/_8$ in. thickness over $^3/_4$ in., with a proportional reduction for fractional parts of $^1/_8$ in., to a minimum tensile strength of 39,000 lb. per sq. in.

† For plates under $^7/_{16}$ in. thick, deduct 1% from the percentage of elongation for each decrease of $^1/_{16}$ in. below $^7/_{16}$ in. thick, to a minimum of 8%.

**Table 2.—Physical Properties of Wrought-iron Bars**

A.S.T.M. Designation A189–36T

| Property | Grade A, Double-refined Bars | | | Grade B, Single-refined Bars | | |
|---|---|---|---|---|---|---|
| | Rounds, Squares or Hexagons Up to 1 $^5/_8$ in. diam.* or thick | Rounds, Squares or Hexagons 1 $^5/_8$–2 $^1/_2$ in. diam.* or thick | All Flats, Rounds, Squares or Hexagons 2 $^1/_2$ in. and over, diam.* or thick | Rounds, Squares or Hexagons Up to 1 $^5/_8$ in. diam.* or thick | Rounds, Squares or Hexagons 1 $^5/_8$–2 $^1/_2$ in. diam.* or thick | All Flats, Rounds, Squares or Hexagons 2 $^1/_2$ in. and over, diam.* or thick |
| Tensile strength, 1000 lb. per sq. in. | 48–54 | 47–54 | 46–54 | 48 | 47 | 46 |
| Yield point, min., 1000 lb. per sq. in. | 0.6× Ten. Str. | 0.55× Ten. Str. | 0.5× Ten. Str. | 0.6× Ten. Str. | 0.55× Ten. Str. | 0.50× Ten. Str. |
| Elongation in 8 in., min., percent | 28 | 25 | 22 | 25 | 22 | 20 |
| Reduction of area, min., percent | 45 | 40 | 35 | 40 | 35 | 30 |

* Short diameter of hexagons.

otherwise taken from the material as rolled. For bars 2 1/2 in. or less in diameter or thickness, the axis of the specimen shall coincide with the axis of the bar. For bars over 2 1/2 in. diameter or thickness, the axis shall be located at any point one-half the distance from the surface to the center of the bar, and parallel to the axis of the bar. It may be machined to a diameter of 1 in. for a distance of at least 9 in., with enlarged ends.

The bend test specimen for round, square or hexagon bars of not over 1 1/2 in. diameter or thickness, and flats not over 1 in. thick or 2 in. wide shall be the full section of the bar as rolled. For larger bars the specimen may be reduced to 1 1/2 in. diameter or thickness. For flats it may be reduced to a width not less than the thickness of the bar. Corners of machined edges may be rounded to a radius of not over 1/16 in.

The etch test comprises etching a ground or polished cross-section of the test specimen in a solution of 10% HCl (sp. gr. 1.19), 30% $H_2SO_4$ (sp. gr., 1.84) and 60% water, or in a solution of 25% $HNO_3$ (sp. gr. 1.42) and 75% water, for a period long enough to develop the structure and to show freedom from steel.

The step test, applicable to Grade A bars only, comprises reducing the specimen to 50% of its original diameter by smooth turned steps, each 2 in. long and 1/8 in. less in diameter than the preceding step. The specimen shall show freedom from injurious seams, laminations and other defects.

The number of tests required is as follows: Bend, tension, nick and step tests, two each, made on bars selected at random from each lot of 100 bars; etch test, one, made from the two bars selected for the other tests.

Permissible variations in size of rounds, squares and hexagons are as follows:

| Nominal size, in. . | up to 1/2 incl. | 1/2–1 incl. | 1–2 incl. | 2–3 incl. | 3–5 incl. | 5–8 incl. |
|---|---|---|---|---|---|---|
| Variation under. . . | 0.007 | 0.10 | 1/64 | 1/32 | 1/32 | 1/16 |
| Variation over. . . . | 0.007 | 0.10 | 1/32 | 3/64 | 3/32 | 1/8 |

## 4. ELECTROLYTIC AND INGOT IRON

Iron in which iron content is not less than 99.9% is tough, ductile, highly resistant to corrosion and has high magnetic and electrical properties. It is made by two processes, *viz.*, extensive refinement in the open-hearth furnace, the product being known as ingot iron; by electro-deposition, the product being known as electrolytic iron.

**ELECTROLYTIC IRON** is made from anodes of ingot iron, soft steel or pig iron. The cathodes are rotating mandrels or moving plates. The electrolyte usually is ferrous chloride. The current density ranges from 12 to 75 amperes per sq. ft. and the voltage drop is 3 to 4 volts per cell. Electrolytic iron has been used where high magnetic properties were required, but the development of alloys of high permeability has displaced it from this field.

**ARMCO INGOT IRON** * is a trade name for commercially pure iron produced by the basic open-hearth process. The operation of the furnace is similar to that for making mild steel, but the refining operations are carried farther. The reduction of metalloids to low values is accomplished by adding iron ore to the bath, which furnishes the oxygen necessary to oxidize the manganese, silicon, phosphorus, and carbon. The highly basic slag reacts with the products of oxidation to effect their removal from the bath. As the metalloids are removed during the refining period, the melting point of the metal is raised, necessitating furnace operation at a higher temperature than ordinarily used for the production of mild steel.

A typical chemical analysis of this material is: Carbon, 0.015%; manganese, 0.020; phosphorus, 0.005; sulphur, 0.025; silicon, trace.

The microstructure of Armco ingot iron consists of a simple network of grain boundaries, characteristic of all pure metals. No other phases are present than the ferrite which makes up the structure of the grains. The size and shape of the grains will differ with various mechanical and thermal treatments.

**Physical Properties.**—The average physical properties are: Specific gravity, 7.866; weight, lb. per cu. ft., 490; specific heat at 25° C., 0.108; heat of fusion, 65 calories per gram; thermal conductivity, 0.16 calorie per sec. per sq. cm. per deg. C. per cm. at 100° C.; thermal coefficient of expansion, 12.6 millionths per deg. C. (20–300° C.); electrical resistivity, 9.6 microhm-cm. at 0° C.; temperature coefficient of electrical resistivity, 0.0056 per deg. C. (0–100° C.).

Magnetic properties are best illustrated by the curves in Figs. 1 and 2. Permeability is high, and retentivity low, which make this material suitable for many special applications.

**Mechanical Properties.**—The average mechanical properties after various treatments are shown in Table 3. Young's modulus is 29,300,000 lb. per sq. in. in both tension and compression, and the shear modulus is 11,800,000 lb. per sq. in. Poisson's ratio is generally considered to lie between $^1/_4$ and $^1/_3$ for most metals, and is given for iron as 0.28 in various sources.

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* Contributed by R. L. Kenyon.

**Effect of Temperature on Mechanical Properties.**—The effect of temperature on the tensile properties as determined by short-time tests is shown in Table 4. The tensile properties at elevated temperatures are greatly influenced by the effect of creep, and the time factor must, therefore, be taken into consideration in designing structures exposed to elevated temperatures. Tapsell and Clenshaw have determined the limiting creep stress of Armco ingot iron as shown in Fig. 3.

The variation in torsional properties and the endurance limit, both at elevated temperatures, are shown in Table 5. Endurance limit, as shown, was determined on the Haigh alternating stress machine at 2400 cycles per minute with equal plus and minus stress on the basis of 10,000,000 reversals.

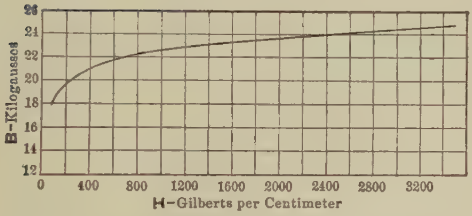

Fig. 1. Magnetic Properties of Annealed Bar

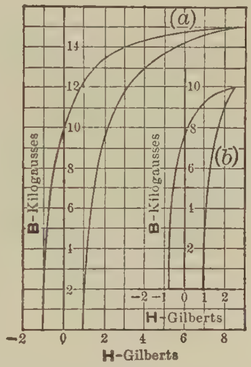

Fig. 2. Magnetic Properties of Annealed Ring. (*a*) Tip induction of 15,000 gausses. (*b*) Tip induction of 10,000 gausses

The plastic deformation of Armco ingot iron is well illustrated by the stress-strain curve determined in the tensile test. Fig. 4 shows a series of such curves made on annealed

**Table 3.—Physical Properties of Ingot Iron**

| Property | Hot Rolled Rods or Plates | Annealed Rods or Plates | Normalized | Dead Soft | Cold Worked (Approx. Max.) | Finished Cold | Finished Hot | Finished Hot Annealed | Finished Hot, Quenched, 1725° F. Water |
|---|---|---|---|---|---|---|---|---|---|
| Compression— | | | | | | | | | |
| Elastic Limit | ........ | 19,400 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Proportional Limit | ........ | 19,200 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Yield Strength | ........ | 20,600 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Tensile— | | | | | | | | | |
| Yield Strength | 26,000–32,000 | ........ | ........ | 19,000 | ........ | 26,900 | 19,100 | 18,300 | 30,300 |
| Tensile Strength | 42,000–48,000 | ........ | ........ | 38,500 | 100,000 | 43,800 | 42,200 | 41,000 | 47,000 |
| % Elongation | 22–28 | ........ | ........ | 43–48 | ........ | 41.8 | 44.7 | 47.0 | 36.2 |
| Gage Length, in. | 8 | ........ | ........ | 2 | ........ | 8 | 2 | 2 | 2 |
| % Reduction of Area | 65–78 | ........ | ........ | 70–77 | 65–70 | 75.6 | 77.3 | 70.6 | 70.0 |
| Torsion— | | | | | | | | | |
| Proportional Limit | 12,000–15,000 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Yield Strength | 14,000–23,000 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Hardness— | | | | | | | | | |
| Brinell Hardness No. | 82–100 | ........ | ........ | 67 | 220 | ........ | ........ | ........ | ........ |
| Rockwell B | 39–55 | ........ | ........ | ........ | ........ | 101 | 90 | 82 | 110 |
| Fatigue— | | | | | | | | | |
| Reversed Bend | ........ | 26,000 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Reversed Torsion | ........ | 12,800 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Axial Stress | ........ | 26,700 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| Impact— | | | | | | | | | |
| Charpy, kg-m | 0.9 | ........ | 2.66 | ........ | ........ | ........ | ........ | ........ | ........ |
| Izod, ft-lb. | ........ | ........ | 90.0 Long.<br>56.3 Trans. | ........ | ........ | ........ | ........ | ........ | ........ |

**Table 4.—Tensile Properties of Ingot Iron at Elevated Temperatures**

Short Time Tests

| Deg. F. | Temperature, deg. C. | Tensile Strength, lb. per sq. in. | Percent Elongation, Including Fracture $L = 4\sqrt{\text{Area}}$ | Percent Reduction of Area | Modulus $E$ |
|---|---|---|---|---|---|
| 60.8 | 16 | 49,500 | 45 | 69 | 29,800,000 |
| 212 | 100 | 57,800 | 27 | 61.5 | 29,250,000 |
| 302 | 150 | 59,800 | 26.5 | 60 | 28,400,000 |
| 392 | 200 | 65,600 | 26.5 | 54 | 27,800,000 |
| 482 | 250 | 64,700 | 30.0 | 54 | 27,200,000 |
| 572 | 300 | 62,500 | 38.5 | 56 | 26,500,000 |
| 662 | 350 | 56,700 | 44.5 | 61 | 25,800,000 |
| 752 | 400 | 44,800 | 51 | 67.5 | 25,000,000 |
| 842 | 450 | 37,000 | 51 | 67.5 | 24,000,000 |
| 932 | 500 | 27,100 | 49 | 63.5 | ......... |
| 1112 | 600 | 15,200 | 34 | 49.5 | ......... |
| 1292 | 700 | 9,200 | 53.5 | 55.5 | ......... |

**Table 5.—Torsional Properties and Endurance Limit of Ingot Iron at Elevated Temperatures**

| Torsional Properties | | | | | Estimated Limiting Range of Fatigue Stress, lb. per sq. in. |
|---|---|---|---|---|---|
| Deg. F. | Deg. C. | Modulus of Rupture | Modulus of Rigidity | Angle of Twist per in. at Fracture, deg. | |
| 60.8 | 16 | 63,200 | 11,600,000 | 778 | 26,700 |
| 212 | 100 | 66,100 | 11,500,000 | 632 | 24,000 |
| 302 | 150 | 70,300 | 11,400,000 | 628 | ...... |
| 392 | 200 | 68,300 | 11,200,000 | 479 | 25,500 |
| 482 | 250 | 68,300 | 11,100,000 | 447 | ...... |
| 572 | 300 | 69,000 | 11,000,000 | 406 | 33,400 |
| 662 | 350 | 65,900 | 10,700,000 | 297 | ...... |
| 752 | 400 | 51,700 | 10,500,000 | 335 | 28,900 |
| 842 | 450 | 41,000 | 10,300,000 | 326 | ...... |
| 932 | 500 | 25,100 | ......... | 249 | 12,300 |
| 1112 | 600 | 15,000 | ......... | 285 | ...... |

sheets that have been cold rolled various amounts. The effect of cold rolling on the tensile strength, yield strength, and elongation, as well as the change in the shape of these curves is clearly brought out.

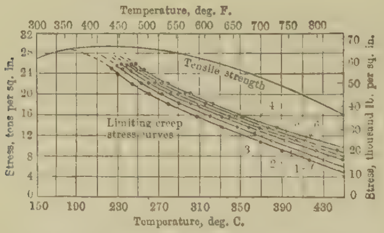

Fig. 3. Creep Strength of Ingot Iron

| Curve | Stretch, in. | Time, Days |
|---|---|---|
| 1 | Limiting creep-stress | |
| 2 | 0.001 | 300 |
| 3 | 0.005 | 60 |
| 4 | 0.010 | 25 |
| 5 | 0.025 | 8 |
| 6 | 0.075 | 2–3 |
| 7 | Gives the tensile strength | |

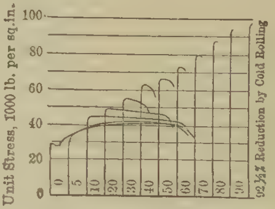

Fig. 4. Effect of Cold Rolling on Shape of Stress-Strain Curve of Armco Ingot Iron

**Uses.**—Armco ingot iron was originally developed as an especially durable metal highly resistant to corrosion, and in sheet form it has been galvanized for roofing, siding, tanks, and signs. In the corrugated form it has been used for culverts. Its high purity and freedom from gas-forming metalloids make it suitable for oxyacetylene and electric arc welding, both as material to be welded and as welding rods. It is also very suitable in the form of sheets for vitreous enameling, because of its freedom from enamel defects such as bubbles and blisters. It is very ductile, and is used for the most severe deep-drawing operations such as refrigerator parts, washing machine tubs, etc. Superior drawing properties and enameling quality are thus incorporated in a single material. Its relatively low electrical resistance, compared to steel, has led to its use as telegraph wire and bond wire for rail signal circuits for railroads. Its high magnetic permeability and low retentivity has stimulated its use for solenoid plungers, magnetic cores, and other parts of electrical equipment.

**FORMS.**—Armco ingot iron is available in the form of hot and cold rolled sheet and strip in all variations of surface and temper, in plate, wire, rods, bars, tubing, and spiral welded pipe.

**FABRICATION.—Cold Working.**—Because of its high ductility, Armco ingot iron can be bent and formed more easily than steel, thus making it possible, in some cases, to substitute cold-forming operations for hot forming with steel. Deep-drawing operations of a wide variety of forms and degrees of intricacy are carried out in cold stamping presses. It is very suitable for cold spinning operations.

**Hot Working.**—Armco ingot iron is not workable in the temperature range from 1552° to 1922° F. This is due to the very low manganese content. This temperature range is avoided in all hot-rolling and forging operations. When Armco ingot iron is to be forged or hammer welded, it should be heated to a white heat or above, when it can be worked until the temperature drops to 1922° F., or to a bright orange color. If more working is required, reheating is necessary. If worked in the critical range between the temperatures given, it will check or break. It is also possible to work the material below the lower limit of this range if hammer welding is not involved, and often this is more satisfactory than working above the critical range, because of the difficulty of preventing some portions of the piece from cooling into the critical range. When worked at temperatures either above or below the critical range satisfactory results will be obtained.

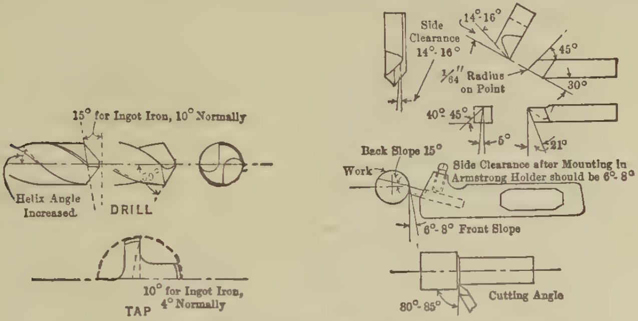
15° for Ingot Iron, 10° Normally
59°
Helix Angle Increased
DRILL
10° for Ingot Iron, 4° Normally
TAP
Side Clearance 14°-16°
14°-16°
45°
$\frac{1}{64}$" Radius on Point
30°
40°-45°
5°
21°
Back Slope 15°
Work
Side Clearance after Mounting in Armstrong Holder should be 6°-8°
6°-8° Front Slope
Cutting Angle
80°-85°

Fig. 5. Recommended Tool Shapes for Machining Ingot Iron

**Machining.**—The high ductility and extreme toughness of Armco ingot iron make it necessary to alter slightly the cutting angles of machine tools in order to get the best results. Shaper tools having a little more rake do away with the tendency to drag and will give surfaces comparable to mild steel. In the case of drilling, the clearance angle of the drill should be increased slightly, and the drill should be ground to cut continuous spirals rather than chips. Fig. 5 shows cutting angles for lathe and shaper tools and drills that have been found satisfactory.

**Welding.**—Both plates and welding wire of Armco ingot iron possess excellent welding characteristics because of purity, uniformity of grain structure, and freedom from gas-forming impurities, all of which result in a freedom from blowholes.

**Brazing and Soldering.**—The usual procedure for brazing and soldering may be carried out without difficulty on Armco ingot iron.

**Annealing.**—Annealing results in a change in the grain structure and physical properties of the material. At temperatures below the $A_3$ thermal critical point (1650° F. for Armco ingot iron), the recrystallization of the grain structure and the change in properties that is produced depends on the amount of previous cold work to which the material has been subjected. Under certain combinations of degree of cold working and annealing temperature, excessive grain growth and inferior physical properties may result. This is also true of low carbon steels, as well as Armco ingot iron, and should be borne in mind when parts are shop annealed in process of fabrication. For example, the best efforts of the manufacturer to produce a good microstructure and physical properties for deep-drawing operations will be entirely destroyed by ill-advised annealing which causes excessive grain growth after cold working. These difficulties can be largely overcome by normalizing or annealing above the $A_3$ temperature. In many cases the high degree of ductility of Armco ingot iron makes it possible to draw the finished part in one operation, thus obviating the necessity of any intermediate annealing treatment at all.

**Riveting.**—Due to the great ductility, rivets of Armco ingot iron up to $^3/_8$ in. can be driven cold. For larger sizes or whenever it is necessary to drive them hot, care should be taken not to heat them above the lower range of critical working temperature.

## 5. STEEL

Steel is iron containing up to 1.7% carbon and not containing admixed slag. The properties of steel are varied over a wide range by variations in the carbon content and in the percentage of other metals alloyed with it. These are discussed in great detail in the following pages.

**CLASSIFICATION OF STEEL.**—Steel may be classified, among other methods, by the process of manufacture, as follows:

**Bessemer Steel** is made in a Bessemer converter from melted pig iron. Only acid Bessemer steel, *i.e.*, steel made in a converter lined with silicious material, is made in the U. S. After all the constituents of the bath of iron in the converter, excepting the iron, have been burned out by means of air blown through the bath, spiegeleisen or ferromanganese is added to the bath to deoxidize the iron and give it the desired carbon content.

Bessemer steel usually is poured into ingots which are rolled into various shapes, as pipe, tin plate, etc., and drawn into wire. It also is used for the manufacture of steel castings, the converters being of about two tons capacity. In these converters air usually is blown on the top surface of the bath instead of through it, resulting in a much hotter metal, since the carbon is burned to $CO_2$ instead of to CO, as occurs when the air is blown through the bath.

**Open-hearth Steel** is steel made in the open-hearth furnace. The charge to the furnace may range from 100% steel scrap with no pig iron to 100% pig iron with no scrap, or any combination in between these limits. Liquid pig iron also may be charged to the furnace. The bath is kept liquid to the utmost limit of purification. The silicon, manganese, phosphorus and carbon are oxidized from the steel as desired, and the chemical composition is adjusted by means of suitable additions of carbon, manganese and silicon, and in the case of alloy steels by the addition of nickel, chromium, vanadium, tungsten, etc. Open-hearth steel may be either acid or basic steel, the distinction being in the character of the lining of the furnace, which, in turn, determines the character of the slag utilized in the process. In the acid open-hearth furnace, the slag is an acid slag, *i.e.*, rich in silica. Phosphorus will be repelled by such a slag and hence cannot be removed from the bath. Raw materials, low in phosphorus, must be used to obtain a good quality of steel in the acid furnace.

Basic open-hearth steel is made with a slag rich in bases, *e.g.*, lime, which will absorb the oxides of silicon, manganese and phosphorus and therefore keep them away from the steel as they are oxidized. Basic steel is free from phosphorus but may contain more oxide of iron than acid steel. Acid steel generally is preferred for service under fatigue conditions, but basic steel represents 85% of the steel used in the U. S. (1937).

**Duplex Steel** is steel that is made by the so-called duplex process in which melted metal from a Bessemer converter is used in the charge for the open-hearth furnace. The acid Bessemer process removes silicon, manganese and carbon rapidly from pig iron. If this purified metal then is charged to a basic open-hearth furnace, the phosphorus and the sulphur are removed. The duplex process produces steel in a minimum of time as compared with the other processes, except the Bessemer.

**Electric Furnace Steel** is steel made either in an arc or resistance type of furnace. The resistance type of furnace generates, by induction, electric current in the steel bath which is heated by virtue of the resistance of the steel to the passage of current through the bath. In the electric arc furnace, which is similar in operation and design to fuel-fired furnaces, the arc, instead of a flame, heats the charge. It is, therefore, capable of much higher heats than either the open-hearth or Bessemer furnace. Its distinguishing characteristic, however, is the absence of an oxidizing or carbonizing flame, enabling iron or steel scrap to be melted without either oxidizing or carbonizing. The electric furnace thus offers the advantage of a furnace atmosphere that may be neutral or reducing as desired, since the charge does not come in contact with any products of combustion. It also is possible to maintain a slag quite free from iron oxide. Such a slag will purify steel to any desired limit by eliminating sulphur, oxygen and oxide. Only the basic electric furnace can hold this purifying slag, but the acid furnace can remove all impurities except phosphorus and sulphur. The basic electric arc furnace hence is used largely for super-refining, that is, for the final stages of desulphurization and deoxidation of steel from the basic open-hearth furnace. The electric furnace has largely supplanted the crucible furnace for making tool steels and alloy steels for high-speed cutting tools.

**Crucible Steel** is steel made in pots or crucibles holding about 100 lb. of metal, and heated, usually, in a pit furnace. The raw material consisting of wrought iron, steel scrap or pure cast iron is packed in the crucible with charcoal or carbon, melted, and held at furnace heat for about one-half hour. Manganese is added to remove oxide from the iron, and silicon is absorbed from the crucible together with carbon, if the crucible is made of graphite. The quality of crucible steel depends on its freedom from objectionable elements as phosphorus, and on the complete removal of oxides, slag and blowholes. This is accomplished by dead melting or killing before pouring, and on the proper addition of alloying elements, as carbon, manganese, chromium, tungsten, etc. While the crucible process was the principal source of tool steel for nearly 150 years, it is now (1937) relatively unimportant.

**IDENTIFICATION OF DIFFERENT TYPES OF STEEL.—Acid and Basic Steel.***—Acid open-hearth or Bessemer steel may be distinguished from basic steel by being lower in manganese, sonims,† and oxygen, but higher in silicon, phosphorus and sulphur. It dissolves more slowly in dilute sulphuric and hydrochloric acids than does basic steel.

**Basic Open-hearth and Acid Bessemer Steel.**—Manganese, silicon, phosphorus and sulphur will be lower, and sonims† and oxygen higher in basic open-hearth than in acid Bessemer or acid open-hearth steel. Basic open-hearth steel dissolves more rapidly than acid Bessemer steel in dilute acids. For the same tensile strength, basic open-hearth steel will be more ductile than acid Bessemer steel with the possible exception of some steel castings.

**Acid Open-hearth and Acid or Basic Bessemer Steel.**—Acid open-hearth steel will have fewer sonims, less oxygen and manganese than acid or basic Bessemer steel, and less phosphorus than acid Bessemer steel.

**Crucible Steel** normally has less than 0.4% manganese, and more than 0.2% silicon, while open-hearth and Bessemer steels have more manganese and less silicon than these figures, except that silicon in steel castings may be higher. Crucible steel usually is high in carbon. If steel with not more than 0.30% carbon is sold as crucible steel, it should be tested for sonims which should be less than 0.03%. As yet (1937) there is no definite means for distinguishing crucible from electric steels except in the low carbon grades. In these grades, silicon usually is over 0.20% in crucible steel, and under 0.20% in electric steel.

**Electric Steel** usually runs under 0.40% manganese and under 0.50% silicon (except in steel of over 0.60% carbon). Even with high carbon steel, except steel castings, silicon generally is under 0.20%. Sonims practically are absent in best quality electric steel. Electric steel has higher elongation and reduction of area for the same tensile strength than any steel except crucible steel. It dissolves much more slowly in dilute sulphuric and hydrochloric acids than does basic open-hearth steel, which alone equals it in low phosphorus.

**EFFECT OF INGREDIENTS ON PROPERTIES OF STEEL.**—The effects of the various elements on the properties of steel are listed below.

**Carbon.**—The effect of carbon on steel is to some extent modified by the effect of other elements, and by the heat treatment that the steel may receive. In general, however, increase of carbon tends to decrease the specific gravity, melting point, elongation, reduction of area and modulus of elasticity, and to increase the tensile strength, yield point, and hardness. The hardness and cutting power of quenched and tempered steels varies almost directly with the carbon from 0 to 2%, the common commercial limits. Above the point of 100% pearlite, *i.e.*, 0.85% C, the cutting power increases more rapidly because of the cementite embedded in the mass. Softness, malleability and toughness decrease in proportion as hardness and brittleness increase with increasing carbon. See Tables 18 and 19 for the effect of various percentages of carbon and various heat treatments on different grades of steel.

**Manganese** increases strength and hardness, decreases oxygen content and mitigates the effect of oxygen and sulphur in the steel.

**Phosphorus** increases strength, brittleness, segregation and fluidity and decreases resistance to shock. With normal heating, it increases grain size.

**Sulphur** slightly increases strength but makes steel dirty, oxidized, segregated, brittle under shock, and red-short. The effect of sulphur can be lessened by the presence of twice the quantity of manganese.

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* This does not include acid and basic electric steel.

† Sonims are the solid non-metallic impurities in iron and steel, including slag particles, sulphides of iron or manganese, oxides of iron, silicon, aluminum, etc., or dirt from ladle or furnace linings.

**Copper** increases hardness of steel. In quantities of less than 1% it will decrease the tendency to corrosion. See Atmospheric Corrosion, p. 3–06. It appears to decrease segregation and grain size.

**Silicon** increases strength and hardness and decreases ductility. It increases the solvent power of steel for gases, especially oxygen and carbon monoxide. This influence decreases blowholes in soft steel, but increases the volume of pipe in ingots and castings. Silicon increases grain size. In castings, 0.20 to 0.40% of silicon often is added for the purpose of increasing strength. Silicon is used in cores of electro-magnets to increase electrical resistance and, therefore, diminish eddy currents.

**Oxygen** may occur in steel as $CO$, $FeO$ or $Fe_2O_3$, being harmful in any of the three forms. Its action is similar to that of sulphur. It produces brittleness in both hot and cold steel, and promotes the formation of blowholes. It probably is the most harmful of all constituents of steel. No satisfactory method has been found (1937) for rapidly determining small traces of oxygen.

**Nitrogen and Hydrogen** both occur in steel in very small quantities. Nitrogen is important in connection with arc welding, when a needle-like constituent appears, which is thought to be a crystal of iron nitride.

**Sonims** are the solid non-metallic impurities found in iron and steel. They include slag particles, sulphides of iron or manganese, oxides of iron, manganese, silicon, aluminum, etc., or dirt from ladle or furnace linings. Sonims remain suspended in some steel baths or ingots until caught in the solidifying mass. The crucible and electric processes of steel making give the best opportunity for the metal to clarify itself. Sonims in steel to a harmful extent are caused solely by bad steel-making practice, such as lack of care in adding oxidizing agents too near the end of the process, or adding deoxidizers in the ingots after the steel has become too cold to allow, for example, oxide of aluminum to rise to the surface.

## 6. METALLOGRAPHY OF IRON AND STEEL

**MICROSCOPIC CONSTITUENTS OF IRON AND STEEL.**—Iron and steel are not simple, homogeneous substances, like glass, but are composed of grains and crystals of different constituents. Thus, there may be, in slowly cooled iron castings, crystals of graphite, cementite, ferrite, and pearlite. The strongest simple steel made (excluding alloy steels, etc.) consists entirely of pearlite. Not only the kinds of constituents but also the size of the crystals in which they occur has an important effect on the qualities of the material which they compose. Thus, large crystals of graphite, etc., indicate weak and non-ductile metal. There are also other constituents which appear under the microscope, such as nitride of iron, and also minute particles of slag, oxides of iron, manganese, and silicon, sulphide of iron or manganese, carbide of manganese, etc. The following is a brief description of the principal constituents of iron and steel.

**Ferrite** is pure iron. It is soft, ductile, malleable; it can stand a great amount of cold work without annealing; it is not strong. It has a tendency to form in rather large crystals during cooling from solidification, or from a high temperature. It will not harden on rapid cooling, and can be formed of small-sized crystals if heated to a bright red heat and cooled very rapidly.

**Graphite**, which appears as free carbon in cast iron, occurs in flat plates which separate under strain. Its effect is to greatly weaken the iron. As the size of the crystals increase, the tensile strength of the iron decreases. Gray cast iron breaks along the planes of the graphite crystals. These, being the only substance visible, give the iron its characteristic color.

**Cementite** is carbide of iron, $Fe_3C$. Its hardness is 6 to 6.5 on Moh's scale, and U on the mineralogical scale. It will scratch glass, but not quartz. It is magnetic below 400° F. It decreases the tensile strength of iron and steel, but increases hardness and cutting properties.

**Pearlite** is an aggregate made of alternate crystals of ferrite and cementite. Its chief characteristic is its minute crystal size. Pearlite has about 7 parts by weight of ferrite to 1 part of cementite, and therefore contains about 0.85% carbon. Steel of this carbon is the strongest simple steel known.

**Austenite** is a solid solution of ferrite and cementite in each other. It forms when the metal solidifies, and remains a solution until it cools to a red heat. But 10% of manganese and over in the steel will preserve austenite to a temperature below 0 F., as will 12 to 25% nickel. Austenite is hard and non-magnetic; it resists wear, but is not brittle. Chromium assists rapid cooling to preserve some of the austenite. If austenite contains 1 part of cementite dissolved in 7 parts of ferrite, the solution will remain until the steel cools to a dark red heat. Precipitation then will occur and if the cooling is slow, the solid steel will change over into a pearlite structure. If the austenite contains more ferrite than the above proportions, ferrite crystals will begin to separate out progressively as the steel cools below a bright red heat, until a dark red heat and a proportion of ferrite to cementite of 7 to 1 is reached at one and the same time. Further cooling causes precipitation of this remaining austenite, and the final steel at atmospheric temperature contains ferrite crystals in pearlite. If the cementite be greater in proportion than 1 to 7, it will begin to separate crystals at bright red heat, with the same procedure, except that the cold steel will contain cementite crystals in pearlite.

**Martensite** is the chief constituent of rapidly-cooled steel. It is the imperfectly preserved austenite, which could not be cooled rapidly enough to prevent some decomposition, but was not so slowly cooled, as to precipitate into pearlite with or without some ferrite or cementite. It is a common constituent of hardened steels. It is not as tough as austenite, but is tough and brittle. It also differs from austenite in being magnetic, but is not as much so as is the same steel slowly cooled.

**Troostite** is a stage in the transformation of austenite one step beyond martensite; it also may be called a stage in the transformation of martensite. It may be formed in any one of three ways: 1. Cool rapidly to produce martensite; then temper to produce troostite. 2. Cool with moderate rapidity, such as quenching in oil. 3. Cool slowly until the transformation begins; then cool rapidly to prevent its completion. Troostite has somewhat less hardness and less brittleness than martensite.

**Sorbite** is the next step beyond martensite in the transformation. It is almost pearlite, and is desirable because of its strength. It has some of the durability of fully precipitated steel. It is formed by the cooling of steel, at a moderately rapid rate, from the temperature of solid solution to atmospheric temperature.

Fig. 6, p. 2–17, shows the relations between the various constituents of steel and iron at various temperatures and carbon contents.

For a discussion of the metallography of iron and steel, see Eshbach, Handbook of Engineering Fundamentals, pp. 11–26 to 11–29, forming Vol. 1 of this series.

## 7. HARDENING, ANNEALING AND TEMPERING

**HARDENING OF STEEL.**—The normal practice in hardening simple steel consists in heating it above a dark red heat and cooling it rapidly, as by plunging it in water, iced brine, or other liquid. The rationale of this is that the steel is heated above the line *PS* in Fig. 6, and cooled rapidly with the expectation of suppressing the change which would occur in slow cooling, that is, the change from austenite to pearlite, etc. What actually occurs is the transformation of the steel into martensite (see p. 2–17) which is hard and brittle. The rapid cooling along the line *PS* sets up strains in the metal, which sometimes are so great that the steel flies into pieces during hardening.

**Rationale of Hardening and Annealing.**—The *critical point curves* in Fig. 6 form the scientific basis of heat treatment. When iron or steel is heated above the line *PSK*, austenite forms; if it is heated within the area *AGOSE*, then all the steel is austenitic, that is, all the ferrite, or cementite (or carbon) is dissolved in the austenite. If now the austenite be formed and cooled rapidly, there is retained in the material some of the imperfectly transformed austenite, that is, martensite, or troostite. The hardness is due to these constituents. If, on the other hand, after forming austenite by heating above the line *PSK*, the steel be cooled slowly, the austenite will break up into pearlite. The softness and ductility of annealed steel is due to pearlite with ferrite.

The hardness of quenched steel usually is believed to be due chiefly to the hardness of the martensite. Martensite is believed to be composed of crystals of iron, body-centered tetragonal, or body-centered cubic, holding carbon in solid solution. Therefore, it must be a highly saturated solution in the high-carbon steels that usually are quenched. The hardness of martensite is believed to be due to the extremely small grain size. It also is believed that the austenite, during its rapid cooling, has broken down into at least incipient crystals of $Fe_3C$, which are distributed throughout the mass in a sub-microscopic size in a state of what is called *critical dispersion.* These extremely small hard crystals of $Fe_3C$ interfere with slip or movement within the crystals, and are responsible for the hardness of the mass.

**Temperature of Breaking Down of Austenite.**—When austenite is quenched, it is believed that it decomposes, not at 1300° F. as normally occurs according to the change phases shown in Fig. 6, but that the change point is depressed to about 500° F. at which temperature there is partial precipitation of $Fe_3C$ in extremely small size. Steel to be properly quenched must be cooled at a rate known as the *critical cooling velocity.* This is the speed of cooling which will prevent any change in the austenite until a temperature of about 500° F. is reached, and yet not so rapidly that some of the austenite will remain unchanged to martensite.

**Correct Temperature for Hardening.**—If steel is heated far above the line *PSK* its crystals grow to large size, which causes weakness and "shortness." (Shortness is that type of brittleness displayed by gray cast iron when it breaks, as distinguished from the violent, springy breaking of strong, hard tool steel.) The fracture shows to the eye bright crystal faces, and is described as "crystalline" or "fiery." Under the microscope the crystals are large, as compared with the minute size of grain of steel that is "silky" or "sappy." A test piece of overheated steel has a straight fracture as compared with the cup-and-cone fracture of "silky" steel, which is associated with high quality steel of low carbon and small crystal size. All heating of steel for quenching, whether of low or high carbon, should stop at the temperature where the steel loses its magnetism.

**Restoring Overheated Steel.**—Overheated steel may be restored to a fine size crystal, called refining, by allowing it to cool to below red heat and then reheating it to 1350° F. If the steel has not remained too long at the high temperature its structure may be wholly refined by one reheating. In this respect it differs from "burnt" steel which cannot be restored to good quality by any number of reheatings, unaccompanied by forging or rolling.

**Use of Pyrometers.**—In all careful heating, whether for hardening or annealing, the best practice controls the temperatures by means of pyrometers. Good quality will be promoted if each new kind of steel is studied experimentally to determine the best temperature, *i.e.*, the temperature that gives the best crystalline structure, irrespective of hardness. The hardness should be controlled by the chemical analysis and not by varying the temperature of hardening. This, of course, does not apply in tempering.

In automatically controlled heat-treating furnaces control is exercised both by pyrometers and magnetic indicators.

**RATIONALE OF GRAIN REFINING.**—When the austenite is formed, the recrystallization due to the dissolving of the constituents in one another has a tendency to destroy the previous crystals. A minute grain structure results at the temperature where the solid solution forms on heating. This structure increases in crystal size in proportion to the temperature above its formation point. If the steel be heated up near the line *AE* the structure becomes very coarse, and the crystals do *not* break up as the temperature falls. They either must be broken up by rolling or forging down to the temperature where they are normally small, or the metal must be allowed to cool below the line *PSK*, and then reheated to the point where the grain will be small. But if the excessive heating has been to a very high temperature, or maintained a very long time at the elevated temperature, then a mere reheating over the line *PSK* may not be effective in entirely curing the defect. The crystals either must be broken up by rolling or forging, or the steel twice reheated over the line *PSK*. It is most effective to first reheat to a moderately excessive temperature; this breaks up the very coarse crystals, but leaves them still coarser than they should be. A second reheating to exactly the right point restores to the steel the best structure obtainable by heat treatment alone. Rolling or forging below the line *PSK* makes finer crystals than does heat treatment alone, but slightly lowers the ductility. Cold rolling makes the smallest crystals of all; wire drawing is equally potential. By wire drawing to very small sizes, tensile strengths may be obtained in ordinary carbon steel (piano wire, for example) up to 400,000 lb. per sq. in., but with almost no ductility left.

**Effect of Heat on the Grain of Steel.**—A simple experiment will show the alteration produced in a high-carbon steel by different methods of hardening. If a bar of such steel be nicked at about 9 or 10 places, about $^1/_2$ in. apart, a suitable specimen is obtained for the experiment. Place one end of the bar in a good fire, so that the first nicked piece is heated to whiteness, while the rest of the bar, being out of the fire, is heated up less and less as the other end is approached. As soon as the first piece is at a good white heat, which burns a high-carbon steel, and the temperature of the rest of the bar gradually passes down to a very dull red, the metal should be taken out of the fire and suddenly plunged in cold water, where it should remain until quite cold. It should then be taken out and carefully dried. An examination with a file will show that the first piece has the greatest hardness, while the last piece is the softest, the intermediate pieces gradually passing from one condition to the other. On breaking off the pieces at each nick, it will be seen that very considerable and charactertistic changes have been produced in the appearance of the metal. The first burnt piece is open or crystalline in the fracture; the succeeding pieces become closer and closer in the grain until one piece is found to possess a perfectly even grain and velvet-like appearance. The first pieces also, which have been too much hardened, probably will be cracked; those at the other end will not be hardened through. Hence if it be desired to make the steel hard and strong, the temperature used must be high enough to harden the metal through, but not sufficient to open the grain.

**Effect of Heat Refining on Size of Crystals.**—Heat refining does not always make all the crystals small. For example, if a steel of 0.50% carbon be heated just above the line *PS K*, say to 1350° F., austenite begins to form with a very small grain size. There is, however, some ferrite which does not dissolve in the austenite, *i.e.*, the ferrite which separated from solution in cooling from the point *O* to the line *PS*. If, however, the heating is continued to higher temperatures, the ferrite progressively dissolves in the austenite, until, at the point *O*, the steel consists wholly of austenite. But in so doing, the austenite first formed has been overheated. This leads to double heat treatments. Thus, if the steel under discussion should be heated to the point *O* and cooled very rapidly, ferrite would be denied the time to separate, and the structure would be minute throughout, but the steel would be brittle from martensite. Most of the ductility then can be restored by reheating just above the line *PS* and cooling slowly.

**ANNEALING.**—The object of annealing steel is for the purpose of securing homogeneity of structure that is supposed to be impaired by unequal heating, or by the manipulation necessarily attendant on certain processes. The objects to be annealed should be heated throughout to a uniform temperature and uniformly cooled.

The physical effects of annealing, as indicated by tensile tests, depend on the grade of steel, or the amount of hardening elements associated with it; also on the temperature to which the steel is raised, and the method or rate of cooling the heated material.

The temperatures employed will vary from 1000° F. to 1500° F. In some cases the heated steel is withdrawn at full temperature from the furnace and allowed to cool in the atmosphere; in others the mass is removed from the furnace, but covered under a muffle, to lessen the free radiation; or, again, the charge is retained in the furnace, and the whole mass cooled with the furnace, and more slowly than by either of the other methods.

The best results from annealing probably will be obtained by the following procedure: 1. Introducing the material into a uniformly heated oven, in which the temperature is not so high as to cause a possibility of cracking by sudden and unequal changing of temperature. 2. Gradually raising the temperature of the material until it is uniformly about 1200° F. 3. Withdrawing the material after the temperature is somewhat reduced and cooling, under shelter of a muffle, sufficiently to prevent too free and unequal cooling on the one hand, or excessively slow cooling on the other.

**Fining the Grain by Annealing.**—Steel which is coarse-grained on account of leaving the rolls at too high a temperature may be fine-grained and have its ductility greatly increased without lowering its tensile strength by reheating to a cherry-red and cooling at once in air.

**Annealing to Relieve Strains.**—Forgings and castings of irregular shape are subject to internal strains, which may cause them to fail in service. These strains usually may be removed by heating the piece to the temperature of mobility, not necessarily to a red heat. Usually it is undesirable to heat forgings above 1300° F., as it lessens the fine structure that should have been given it by the mechanical treatment.

**Annealing of Steel Castings.**—The structure of steel castings is a feature of great importance. They do not get the beneficial kneading effect on the grain that rolled steel obtains, and the size of crystals must depend on the temperature of casting, rate of cooling, etc. When this crystal size is too large, or too open, both ductility and strength suffer. Correct annealing will improve all the qualities. The general rule is that steel castings must be reheated very slightly above the line *GOS* in Fig. 6, that is, above the temperature at which magnetism vanishes, and cooled slowly enough to acquire the requisite ductility. Research often is necessary to determine the best treatment. It must be borne in mind that it is the proper structure that is desired, and that tensile test, elongation, etc., are only indications of the structure. When the analysis is right and the mechanical qualities are not up to standard, it usually is evidence enough that the grain structure is too large; there even may be fine-sized crystals, but set in a network of large size, indicating a relict of bad crystallization when first cooled, called *ingotism*. In such case, it may be best to heat first to a high temperature; this breaks up the coarse network, but leaves the individual crystals large and "fiery." They may be refined by a second heating to immediately above the line *GOS*.

Damage to the structure of a steel casting most often is due either to too high casting temperature, or to too slow cooling just before the solidification temperature. For the very best structure, carbon steel castings of less than 0.25% carbon should be reheated to 1500° F., and then cooled rapidly to 1300° F., but for their best degree of softness and ductility, they should cool slowly from 1300° F. to 1100° F. This can best be accomplished by a double heating and cooling, when the cost thereof is justified. Specifications for annealing various classes of steel products are given in the notes accompanying Table 6.

**Annealing Tool Steel.**—Annealing or softening tool steel is accomplished by heating steel to a red heat and then cooling it very slowly, to prevent it from getting hard again.

There is nothing gained by heating a piece of steel hotter than a good, bright cherry-red; on the contrary a higher heat has several disadvantages: First, if carried too far it may leave the steel actually harder than a good red heat would leave it. Second, if a scale is raised on the steel, this scale will be harsh, granular oxide of iron, and will spoil the tools used to cut it. Third, a high scaling heat continued for a little time changes the structure of the steel, makes it brittle, liable to crack in hardening, and impossible to refine.

To anneal any piece of steel, heat it red-hot; heat it uniformly and heat it through, taking care not to let the ends and corners get too hot. As soon as it is hot, take it out of the fire, the sooner the better, and cool it as slowly as possible. A good rule for heating is to heat it at so low a red that when the piece is cold it will still show the blue gloss of the oxide that was put there by the hammer or the rolls. Steel annealed in this way will cut very soft; it will harden very hard, without cracking, and when tempered, it will be strong, nicely refined, and will hold a keen, strong edge.

**Annealing Alloy Steels.**—Special directions are necessary for annealing alloy steels used for cutting tools, etc. Such directions should be obtained from the maker. General directions will only be confusing. For nickel-steel, chromium-steel, etc., it may be assumed that slow cooling from a temperature above the critical points will soften the material. Chromium steels require specially slow cooling.

**TEMPERING.**—Tempering steel is the act of giving it, after it has been shaped, the hardness necessary for the work it has to do. This is done by first hardening the piece, generally a good deal harder than is necessary, and then toughening it by slow heating and gradual softening until it is just right for work. A piece of steel properly tempered always should be finer in grain than the bar from which it is made. If it is necessary, in order to make the piece as hard as is required, to heat it so hot that after being hardened the grain will be as coarse or coarser than the grain in the original bar, then the steel itself is of too low carbon for the desired work.

**Tempering by Oil Quenching.**—Tempering means literally "softening," and originally referred to drawing the hardness from quenched steel by reheating to temperatures of 425° F. to 625° F., as described above. Long usage, however, has caused the term among tool-makers to mean giving to steel the correct hardness for cutting purposes. Therefore, "oil tempering" relates to quenching steel in oil from a red heat. The trade now supplies oils of different degrees of cooling power, adjusted to give different degrees of hardness on quenching, and this hardness does not require any tempering. Quenching in melted lead gives less hardness than quenching in oil.

**Oil Tempering for Strength.**—The elastic limit of forgings may be raised by heating them to 1350° to 1550° F. as the case may be, and quenching them in heavy oil. Steel castings may be strengthened in this way if the shape is not such as to make them liable to crack while cooling. See notes accompanying Table 6, p. 2–22.

**DOUBLE HEAT TREATMENTS FOR STRENGTH.**—The theory outlined above is the basis of double heat treatments for increasing the strength of steels, especially steels very low in carbon, which are not much hardened by rapid cooling and whose structure is normally coarse on account of much ferrite.

## 8. ALLOY STEELS

**PROPERTIES OF CARBON STEELS WITHOUT THE SO-CALLED ALLOYING ELEMENTS.**—Normal carbon steel always contains small-sized crystals of Fe, known as ferrite, and of $Fe_3C$, known as cementite. A pure alloy containing 0.83% carbon, which corresponds to the point $S$ in the equilibrium diagram, Fig. 6, consists entirely of small-sized crystals of Fe and $Fe_3C$. With less than 0.83% carbon there are also large crystals of Fe; with more than 0.83% carbon there are also large crystals of $Fe_3C$. No normal carbon steel contains both large crystals of Fe and large crystals of $Fe_3C$.

Fe crystals are relatively soft, tough, ductile, not very strong, and yield easily along slip planes so as to be capable of considerable deformation before breaking. When relatively large areas of Fe crystals are present, as in steels of 0.22% C for example, the steel yields so easily that it breaks without exhibiting great strength, or even very great ductility. The $Fe_3C$ crystals occur in thin plates. When there are several crystals of $Fe_3C$ in excess of the small crystals of $Fe_3C$ and the small crystals of Fe, as, for example, in steel containing 1.25% C, the hard, brittle, cementite planes do not adhere strongly to the conglomerate. Steel containing 0.83% C is the strongest normal steel, because it has no large-sized crystals. This steel has a slight ductility, represented by about 5% elongation in 2 in., entirely due to the deformation which the crystals of Fe suffer under strain. This deformation is limited by the unyielding nature of the $Fe_3C$ crystals with which they are mingled in intimate contact. By *spheroidizing* the steel, the crystals of $Fe_3C$, both in the fine crystal matrix and in the excess, if any, become rounded rather than plate-like, and the toughness and resistance to impact is greater when steel is spheroidized by heat treatment than when in the unspheroidized condition.

Finally, the Fe and $Fe_3C$ are completely in solid solution above the lines *GS* and *SE* in Fig. 6. As the steel cools from these temperatures, any Fe in excess of 99.15% (for example, in steels containing 0.30% C) will begin to separate as the steel cools slowly past the line *GS*; if the carbon is in excess of 0.83 (which corresponds to $Fe_3C$ = 12.45%), any excess of $Fe_3C$ above 12.45% will begin to separate as the steel cools slowly below the line *SE*. Whatever the composition of the steel, small crystals of Fe and $Fe_3C$ completely separate from each other and eliminate any residual of solid solution left when the steel cools slowly below the line *PSK*. If, instead of cooling slowly, the rate of cooling is regulated, the breaking up of the solid solution which normally occurs in the areas *GPS* and *FCESK* in Fig. 6 can be prevented, to a greater or lesser degree.

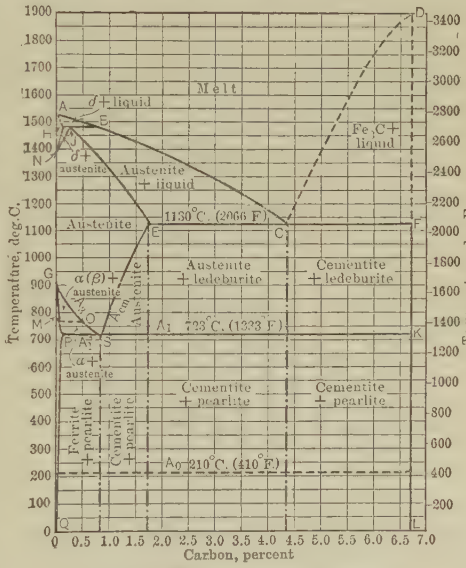

FIG. 6. Iron—Iron-carbide Diagram

**EFFECTS OF HEAT TREATMENT.**—Heat treatment of plain carbon steel will alter the size of the crystals of Fe and $Fe_3C$, or the shape of the crystals of $Fe_3C$, or the relation of these crystals to each other. Therefore, by varying the number of crystals of Fe and $Fe_3C$, which can be controlled by the amount of carbon in the steel, and by varying the heat treatment, steel may be made either hard or soft, or brittle or strong or not so strong, or high in magnetic retentivity or in permeability, or tough or ductile, etc.

**EFFECT OF ALLOYING ELEMENTS.**—Alloying elements such as nickel, chromium, vanadium, silicon, etc., will alter the phase changes, shown in Fig. 6, just as heat treatment will do, and will sometimes control them more effectively. For example, with more than 7% Mn or more than 20% Ni, or a suitable combination of two alloying elements, such as 18% Cr with 8% Ni, a solid solution can be retained down to atmospheric temperature; whereas, with heat treatment alone, the separation of the constituents can be only partially prevented, resulting often in the semi-separated, hard, brittle material which is familiar in tool steels. Alloying elements also will enter into the Fe or $Fe_3C$ crystals, as the case may be, and further alter their properties; for example, nickel will dissolve in the Fe crystals and make them harder, more rust resisting, and having greater resistance to impact stresses, especially at temperatures at and below freezing; chromium and vanadium will greatly intensify the hardness of $Fe_3C$, etc.

**NICKEL STEELS.**—Because nickel dissolves in ferrite, it is especially valuable in the low carbon steels. It causes the point *S* in Fig. 6 to move to the left, so that slowly-cooled nickeliferous steels of 0.35% C will have less excess Fe at atmospheric temperature than they would have if the nickel were not present. Furthermore, the presence of the nickel makes the Fe crystals which do come out in the area *GSP* much smaller than they otherwise would be. Therefore, a structural steel containing 0.35% C and 3.75% Ni will have almost twice the tensile strength and more than twice the elastic limit, with only about 20% less ductility, than the ordinary structural steel containing 0.22% C. The nickel steels also will have increased resistance to fatigue and to impact. Nickel steels also show greater improvement with heat treatment than do carbon steels as regards resistance to static stresses, fatigue stresses, and impact stresses. Because of the smaller size of Fe crystals, nickel steels containing 3%, 5%, and 7% Ni respectively, are used for carburizing, because the cores, which, in America, contain about 0.20% C, are tougher and stronger. In this connection a nickel-chromium steel, containing commonly 1.5% Ni and 0.65% Cr, gives very high strength after suitable heat treatment; the cores are not only tough, but the cases are extra hard because of the presence of the chromium and 0.90% carbon

(13.5% $Fe_3C$). A two-volume monograph on the alloys of iron and nickel issued by the Alloys of Iron Research Committee (McGraw-Hill Book Company, publishers) contains practically all the information extant in any language on the subject of nickel steels.

**Uses of Nickel Steels.**—Doubtless the largest use of nickel steels is for structural work, including buildings, bridges, automobiles, etc., and also rotating shafts, such as propeller shafts in vessels, which are now almost all hollow nickel steel shafts, axles in locomotives and other special purposes, airplanes, etc. Nickel also is used commonly with chromium, with molybdenum, with vanadium, or a combination of all three, and less often with manganese and other elements to form so-called *quaternary alloys*, meaning iron, carbon and two others. To a lesser extent nickel steels are used because of their hardness and because of their slightly increased resistance to corrosion. A special nickel steel containing 42% Ni and known as platinite has, at atmospheric temperature, the same expansion and contraction as glass. This alloy therefore is used for wires in the base of incandescent electric light bulbs, for armored glass, etc. An alloy containing 36% Ni is known as Invar. It has almost no expansion or contraction with atmospheric changes of temperature, and therefore is used for instruments of precision, pendulums, watch springs, surveyor's measuring tapes, etc.

**CHROMIUM STEEL.**—The best known chromium steels are the various types of stainless steel, which will be considered in a special paragraph below. Chromium makes the phase changes illustrated in Fig. 6 much more sluggish and slow to occur. It also hardens $Fe_3C$. For both of these reasons it intensifies the hardness produced by quenching steel. This is most important in increasing the depth of hardness of quenched articles and also in getting hardened materials by quenching in oil instead of water, which is less liable to produce distortion or cracking. For these purposes about 0.5% to 1.5% Cr ordinarily is used. Chromium moves the point $S$ in Fig. 6 to the left and, on quenching, produces a steel with extremely small crystals. For both these reasons it greatly increases the strength of the quenched material. Because of the very sluggish reaction which takes place in chromium steels, it is important to heat them to a higher temperature than other steels before quenching and also to " soak " for a longer time at temperature. A two-volume book on the Alloys of Iron and Chromium issued by the Alloys of Iron Research Committee contains practically all the available information on this subject to date (1937).

**Uses of Chromium Steels.**—Chromium is used in all steels where intensity or depth of hardness is to be increased, or equal hardness is to be obtained with less severe quenching. With chromium less than 1%, the steels are used largely for case hardening and structural steels for strength, especially when combined with nickel or vanadium. Steel containing between 1% and 2% Cr and from 0.75% to 1% C is used very largely for bearings, crushing machinery, burglar-proof safes, and tools. When heat treated, it has the highest combination of hardness and toughness of any plain or alloy steel. It is given a spheroidizing heat treatment before quenching. With 0.70% to 5% C (usually 2% to 4%) and 0.75% to 1.05% C, it is much used for permanent magnet steels. Heat-treated chrome-nickel steels for strength usually contain about 1.50% Ni and 0.65% Cr, or a like combination in which the nickel is approximately 2 1/2 times the chromium. The chrome-vanadium steels usually contain about 1% Cr and 0.18% V. They have a high combination of strength with toughness, and a high impact resistance.

**STAINLESS STEELS** (see also p. 3–19).—Chromium decreases the rusting of ordinary steel; when the amount of chromium is at least 11.5%, the steel retains a bright mirror-like surface in the atmosphere, whence the name stainless steel. These steels also resist the attack of many agents, such as nitric and acetic acids, fruit juices, dairy products, alkalis, oil refinery residua, etc. Especially when the chromium is above 17% or when there is a combination of 18% Cr with 8% Ni, they do not scale so readily at high temperatures as other steels. If they contain more than about 0.10% C and not more than 12% to 15% Cr, they must be heated until all the carbide is in solid solution and then quenched. Otherwise they will not resist corrosion and staining. The high strength combined with good ductility of the stainless steels also finds extensive use in the manufacture of airplanes, high-speed trains, and other purposes where strength and lightness of weight are important. The Book of Stainless Steels, edited by Ernest E. Thum (second edition, 1935), gives a wealth of information on this material and its uses.

**AUTOMOBILE ALLOY STEELS.**—The greatest tonnage of alloy steels comprises those containing nickel or chromium or both. With the exception of stainless steels, about 80% of all the alloy steels made are used in automobile manufacture, and many of these are the so-called quaternary steels, *i.e.* those containing three elements besides iron, for example, carbon, nickel, and chromium. See p. 2–37 for further details of the various automobile steels and their treatment.

**HEAT TREATMENTS FOR ALLOY STEELS.**—Even alloy steels are not so markedly increased in tensile strength when annealed in the usual manner, but the circumstance that they may be cooled with some degree of rapidity without great loss of ductility permits them to be quenched in heavy or medium oil. Thus their elastic limit and strength may be increased without destroying the toughness requisite in machinery parts. The treatment consists in raising the steel to a temperature above the line *GS* in Fig. 6, which line will be at somewhat different temperatures for different alloying elements; then soaking the steel until it is all in a solid solution, and quenching in oil; it then is drawn to temperatures of 800° to 1400° F., depending on the ductility required. Typical heat treatments are given in Table 20 and in the notes accompanying Table 6. However, every second or third year the A.S.M. publishes a new edition of the *Metals Handbook*, in which will be found the latest information as to the most desirable temperatures and treatments. The important point is that the first heating must bring the steel to a state of complete solid solution and the second heating must not be above the line *PSK* for the steel being treated.

**HIGH-SPEED STEEL.**—Tungsten and molybdenum have the apparently unique property of conferring on hardened steel the power of retaining its hardness even when heated for a few hours to 1150° F. (low red heat). Usually, hardened carbon steel will begin to soften at 200° to 300° F. Tungsten and molybdenum also give the steel some degree of self-hardening power; and, finally, they prevent the coarsening of the structure which occurs in ordinary steel when it is heated to bright red, or white, heats. Therefore a steel having 6% to 20% of tungsten, or 3% to 10% of molybdenum (or part of each), can be heated to near its melting point without damage. If then it be cooled with moderate rapidity, it will be harder than ordinary tool steel, and it can be kept at 1100° F. for a long time without losing much of the hardness. This enables such a steel to be used for a cutting tool at so rapid a speed that its point becomes red with the heat of friction without losing its hardness for several hours. The carbon in these high-speed steels usually is only 0.60%, because the hardness then is sufficient and the toughness is greater than in a tool steel of carbon of 1% or over. Frequently, some chromium is used to intensify the hardness, and some vanadium to increase the strength. A great variety of analyses and heat treatments is employed, with a general average of 0.60% C, 16% to 20% W, 2% to 4% Cr, and 1% V. 18% W, 4% Cr, 1% V is a common composition in the United States. Before use, the steel is heated to near the melting point—treatment that would ruin ordinary steel—and then is cooled with moderate rapidity, as by holding it in an air-blast, or quenching it in oil. The heating and forging of the steel requires great care, and the proper directions for each brand should be obtained from the manufacturer.

**Molybdenum in High-speed Steel.**—Sometimes molybdenum is used in high-speed steel. It takes only one-half as much molybdenum to form $Mo_2C$ as tungsten required to form $W_2C$. Molybdenum steels are more difficult to make and handle. Molybdenum is also sometimes used to replace a part of the tungsten.

**Heat Treatment for High-speed Steel.**—High-speed steel is heated to about 2300° F. and cooled with moderate rapidity—as quenching in heavy oil, or subjecting to a blast of cold air. This treatment, which would be ruinous to ordinary steel, suffices to confer the high-speed qualities. It has been suggested that the reason for this is the behavior of carbide of tungsten as distinguished from that of carbide of iron. Tungsten has an atom of large size, which, therefore, diffuses very slowly in a solid solution. When steel is heated to about 2300° F. and cooled with moderate rapidity, the effect is to form and retain a double carbide of tungsten and iron, probably only partially separated from the solid solution and in extremely small size—perhaps colloidal. The result is similar to the extremely fine dispersion of $Fe_3C$ to which is ascribed by many metallurgists the hardness of quenched carbon steel. But when quenched carbon steel is heated to 400° to 600° F., the $Fe_3C$ particles begin to grow by diffusion and accretion. Even before reaching these temperatures, the hardness begins to decrease. Some claim that other crystals in the quenched steel also grow. In any event, the hardness is decreased as the extremely fine state of dispersion is decreased. In high-speed steels, the iron-tungsten carbide is so stable and diffuses so sluggishly that the particle size does not grow until the steel reaches a red heat. The persistent critical dispersion of the double carbide is said also to prevent the growth of other crystals in the steel. This explanation has been offered for the retention of hardness in high-speed steel up to red heat.

**Other Tungsten Steels.**—The Alloys of Iron Research Committee published a book on the Alloys of Iron and Tungsten in 1934, which contains very complete information on this subject. Tungsten and molybdenum both have more powerful effects than any other element in increasing the strength of steel at temperatures of 500° F. to the melting point. Five percent to 7% tungsten is used in permanent magnet steels together with 0.70% to 0.75% carbon. Before magnetizing, these steels should be hardened by quenching in water at a temperature between 1470° and 1600° F. The addition of 2% chromium to tungsten magnet steel improves its properties.

**SILICON STRUCTURAL STEEL.**—Silicon has the effect of moving to the left the point $S$ in the diagram in Fig. 6. Thus steel with 1% to 1.50% Si and about 0.35% C will be almost free from large Fe crystals. It, therefore, will be much stronger than the same carbon steel without silicon and will have almost equal ductility, because it will have considerable Fe in the form of small-sized crystals. Silicon structural steel has come into considerable use for bridges, as it has almost equal physical properties in strength and ductility to nickel steel and is much less expensive. From 0.90% to 1.50% Si usually is used in structural steels. In 1933 the Alloys of Iron Research Committee published a comprehensive book on the alloys of iron and silicon.

**SILICON TRANSFORMER IRONS.**—The purest possible form of iron will make a very superior transformer steel if silicon is added to increase its electrical resistance, thereby reducing eddy currents, and to precipitate carbon from solid solution, thereby increasing magnetic permeability. As much as 0.015% C, and sometimes more, may be retained in steel in solid solution at atmospheric temperature. In this form carbon has a more damaging effect on magnetic permeability than the same amount of carbon in the form of $Fe_3C$ or graphite. Silicon precipitates the carbon from the solid solution and itself enters the steel in the form of a solid solution, probably as a compound FeSi. This solid solution increases the electrical resistivity of the iron. The best amount of silicon is 4.75%, but this increases the expense, because the steel is difficult to shear without cracking on the edges. Consequently, silicon transformer irons of 1.75% and 3.75% Si, respectively, often are used. Modern methods of heating, usually kept secret, have the effect of decreasing the brittleness and tendency to crack when sheared.

**CORROSION-RESISTANT SILICON CAST IRONS.**—Cast irons containing from 13% to 17% Si have considerable resistance to rusting and to attack by sulphuric, nitric, organic, and (to some extent) hydrochloric acids. They are much used in chemical manufacturing plants. Their most important drawback is their brittleness. They must be protected from shock and cannot be machined or forged. They must be cast in the form of bowls, because plates and bars will crack in cooling. They go under the name duriron, tantiron, corrosiron, thermisilid, elianite, etc. See p. 3–24.

**MEDIUM MANGANESE STEELS.**—Steel containing about 1.75% Mn was used tentatively for bridge construction, and has been used for railroad rails because of improved resistance to the formation of transverse fissures. It is stronger than carbon steel, but the full benefit of the properties can be obtained only by heat treatment of moderately light sections. Therefore, it is not entirely successful for structural work. This type of manganese steel, when made in low carbon, is used for carburizing parts. With carbon averaging about 0.30% it is used for machine parts, and it then is normalized and preferably further heat treated by quenching and drawing. In the automobile field, steel with slightly higher carbon, when heat treated, is used for bolts, yokes, flanges; and with carbon up to 0.60% is used for oil-hardened gears, axles, grinding balls, etc.

**WEAR-RESISTING MANGANESE STEEL.**—Steel containing about 13% Mn is in a state of solid solution when rapidly cooled from white heat to atmospheric temperature. This is the oldest alloy steel used in large quantities, and is especially advantageous because of its resistance to wear. For teeth of digging machinery, railroad rails on curves, and other parts requiring resistance to abrasion, it is unequaled. It has very high ductility and a moderately high tensile strength, but the yield point is only about $^1/_4$ of the tensile strength. In order to be ductile, it must be heated to about 1850° F. and quenched in water; and in order to retain its wear resistance, the surface must be constantly subjected to pressure or impact, as by digging broken rock, or grinding against hard surfaces.

**VANADIUM STEEL.**—Vanadium is a powerful deoxidizer, but if more than about 0.20% V is left in the steel, it becomes brittle. Vanadium inhibits the grain growth of steel above 1300° F. It is used in castings because it improves the grain size and is used in heavy forgings, especially locomotive forgings, when the carbon is about 0.50% and sometimes for hardened steels approaching 1% C. The most common use of vanadium is in combination with chromium for very strong and tough parts after quenching and drawing. It is used for carburizing parts, oil-hardened axles, shafts, gears, especially in automobiles. Chrome-vanadium steels of about 1% Cr and 0.15% V commonly are used for automobile coil and flat springs. Vanadium is also used largely for increasing the hardness of high-speed steel, tools, armor piercing projectiles, permanent magnets, etc.

**MOLYBDENUM STEEL.**—The Alloys of Iron Research Committee has published a book, The Alloys of Iron and Molybdenum, containing all available data on the alloys of these metals. Molybdenum increases the tensile strength of carbon steel with only a slight decrease in ductility. Like tungsten, it has the effect of increasing the strength of steel at high temperatures. It increases machinability, the range of temperatures used in heat treating, and is more important as an addition to other alloying elements, such as chrome-molybdenum, nickel-molybdenum, nickel-chrome-molybdenum, etc. Chrome-molybdenum steel is much used for the structural parts of airplanes because of the ease with which it can be fabricated and welded, and its high strength when heat treated. Molybdenum greatly improves cast iron in amounts from 0.5% to 1.5% Mo. In high-strength cast iron, it has about the same effect as the combination of 1.5% Ni and 0.5% Cr.

**SILICO-MANGANESE STEELS.**—Steel containing 1.75% to 2.25% Si and 0.60% to 0.90% Mn, after quenching and drawing, has high resilience and high strength with good ductility. It is used largely in the automotive and railroad field for coil and leaf springs.

**AUSTENITIC STEELS.**—Steels which are in the state of solid solution at atmospheric temperature are known as austenitic steels. They are characterized by great ductility and wear resistance. If the solid solution contains 12% or more of chromium, they also resist corrosion, by all agencies which do not corrode chromium. If they contain carbon, it may be necessary to cool them rapidly from about 1500° F. in order to retain all the carbon in solid solution. Without this treatment they do not resist corrosion or possess high ductility. The common austenitic steels are the 18% Cr 8% Ni steels, the high manganese steels, and steel containing more than 20% Ni.

**STEEL FOR SPRINGS.**—The preceding paragraphs may be summarized by saying that the steels chiefly used for springs are the chrome-vanadium and silico-manganese steels. Straight carbon-chromium or carbon-molybdenum steels are used; also chrome-molybdenum, chrome-silicon-manganese and silicon-vanadium. The manufacture of steel for springs is of prime importance, because the service of the material is seriously affected by imperfections on or near the surface of the coil or leaf spring. Great honest difference of opinion exists as to the best alloy steel for springs. Chrome-vanadium steel seems to have the best combination of strength, resilience and durability, but many claim that silico-manganese steel is less subject to surface defects. See also p. 10–03.

**STEEL FOR MAGNETS.**—Carbon steel containing about 1% C, and quenched from above the line *PSK* in Fig. 6, makes a good permanent magnet. If the carbon is reduced slightly and some alloying elements are added, such as from 1% to 6% Cr or from 5% to 6% W or, better yet, 2% to 10% Cr, 1% to 10% W, and about 1% Mo, with up to 36% Co, very much improved magnetic properties are obtained.

**STEEL FOR HEAT RESISTANCE.**—Steels containing tungsten or molybdenum retain strength better at high temperatures. Steels containing 17% to 20% Cr with 8% to 10% Ni resist scaling at high temperatures.

**NITRIDING.**—Since about 1925 steel has been hardened on the surface by a process known as nitriding instead of the familiar carburizing process. The great advantage in the nitriding process is that it involves no quenching. Therefore, members of a complicated shape or uneven cross-section can be treated with safety. The steel also has a greater hardness, ranging from 900 to 1200 Brinell, as compared with 650 for carburized surfaces. The members are stronger and are said to have slightly greater corrosion resistance. The disadvantage of the nitriding process is the expense, the expert ability required to perform it, and usually the long wait required for obtaining the finished article.

In the nitriding process, the steel articles are heated in an atmosphere of ammonia gas. It is common to nitride for 12 hours at 875° to 1000° F. and then follow with 12 hours at nearly 1200° F. The steel to be nitrided usually contains about 1.25% Al, with 1% to 1.50% Cr and 0.15% to 1% Mo. Before nitriding, the steel is given a double heat treatment for strength. The purpose of the aluminum is to make a stable nitride. An innovation which has met with some success has been the use of nitriding for hardening the surfaces of cast iron objects.

## 9. STANDARD SPECIFICATIONS FOR STEEL PRODUCTS

The American Society for Testing Materials in the triennial issue of its Standard Specifications gives specifications for the composition, physical characteristics, heat treatment, etc., of many iron and steel products. The more important of these, from the 1936 issue, have been condensed in the following pages. Chemical and physical specifications are given for most of these products in Table 6 and the notes which accompany it.

Specifications for carbon and alloy steel forgings and for hot rolled bars and cold-finished bars and shafting are given in Tables 12 to 17. In all specifications, the following provision or its equivalent should be included:

The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material.

Reference should be made to the latest issue of the Standard Specifications of the A.S.T.M. for complete specifications of any particular product.

Standard specifications for steels for use in the automotive industries are issued by the Society of Automotive Engineers. These are abstracted on pp. 2–37 to 2–48.

**Table 6.—Specifications**

Abstracted from A.S.T.M.

| A.S.T.M. Designation No. | Material | Process of Manufacture* | Chemical Composition | | | | |
|---|---|---|---|---|---|---|---|
| | | | C | Mn | P, max. | S, max. | Ni |
| | Carbon Steel Rails: | | | | | | |
| A1-36 | 50–69 lb. per yard | H | 0.50–0.63 | 0.60–0.90 | 0.04 | .......... | .......... |
| | 70–84 " | H | .53– .70 | .60– .90 | 0.04 | .......... | .......... |
| | 85–100 " | H | .64– .77 | .60– .90 | 0.04 | .......... | .......... |
| | 101–120 " | H | .67– .80 | .70–1.00 | 0.04 | .......... | .......... |
| | 121–140 " | H | .69– .82 | .70–1.00 | 0.04 | .......... | .......... |
| | Structural Steel: | | | | | | |
| A7-36 | Plates, shapes, bars | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.05 | .......... |
| A7-36 | Eyebar flats | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.05 | .......... |
| A141-36 | Rivet steel | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.05 | .......... |
| A9-36 | Steel for buildings | B, H, E | .......... | .......... | 0.10[5]; 0.06[6, 7] | .......... | .......... |
| A10-34 | Mild steel plate | H, E | .......... | .......... | 0.06[1]; 0.04[2] | .......... | .......... |
| A113-33 | Structural Steel for Locos. and Cars: | | | | | | |
| | For cars | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.06 | .......... |
| | For locomotives | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.05 | .......... |
| | Plates for cold pressing | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.06 | .......... |
| A131-36 | Structural Steel for Ships | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.05 | .......... |
| | Rivet steel and steel for cold bending | H, E | .......... | .......... | 0.06[1]; 0.04[2] | 0.05 | .......... |
| A94-36 | Structural Silicon Steel | H, E | 0.40[8, 9]<br>0.44[8, 10] | ..........<br>.......... | 0.06[1, 9]; 0.04[2, 9]<br>0.75[1, 10]; 0.05[2, 10] | 0.05<br>0 063 | ..........<br>.......... |
| A195-36T† | High Strength Structural Rivet Steel (Suitable for use with Structural Silicon Steel) | H, E | 0.30[8] | 1.65[8] | 0.06[1]; 0.04[2] | 0.05 | .......... |
| A8-36T† | Structural Nickel Steel: | | | | | | |
| | Plates, shapes, bars | H, E | 0.43[8] | 0.80[8] | 0.04 | 0.05 | 3.00–4.00 |
| | Eyebar flats | H, E | 0.43[8] | 0.80[8] | 0.04 | 0.05 | 3.00–4.00 |
| A78-33 | Steel Plates, structural quality for forge welding: | | | | | | |
| | Grade A | H, E | 0.15[8, 12]<br>0.17[8, 13] | 0.35–0.60 | 0.06 | 0.05 | (16) |
| | Grade B | H, E | 0.20[8, 12]<br>0.22[8, 13] | 0.35–0.60 | 0.06 | 0.05 | (16) |
| | Concrete Reinforcement: | | | | | | |
| A15-35 | Billet Steel Reinforcement Bars | | | | | | |
| | Plain bars, structural steel grade | B, H, E | .......... | .......... | | .......... | .......... |
| | Plain bars, intermediate grade | B, H, E | .......... | .......... | 0.10[1, 5]<br>0.05[2, 6, 7]<br>0.08[1, 6, 7] | .......... | .......... |
| | Plain bars, hard grade | B, H, E | .......... | .......... | | .......... | .......... |
| | Deformed bars, structural steel grade | B, H, E | .......... | .......... | | .......... | .......... |
| | Deformed bars, intermediate grade | B, H, E | .......... | .......... | | .......... | .......... |
| | Deformed bars, hard grade | B, H, E | .......... | .......... | | .......... | .......... |
| | Cold-twisted bars[17] | B, H, E | .......... | .......... | | .......... | .......... |
| A16-35 | Rail Steel Reinforcement ment Bars[20] | | | | | | |
| | Plain bars | ...... | .......... | .......... | .......... | .......... | .......... |
| | Deformed and hot-twisted[17] bars | ...... | .......... | .......... | .......... | .......... | .......... |
| | Spring Steel: | | | | | | |
| A14-27 | Carbon Steel Bars for Railway Springs | | | | | | |
| | Grade A[22] | H, E, C | 0.90–1.10 | 0.50 | 0.05 | 0.05 | .......... |
| | Grade B[23] | H, E, C | 0.95–1.15 | 0.50 | 0.05 | 0.05 | .......... |
| A68-36 | With Special Silicon Requirements | H, E, C | 0.95–1.05 | 0.25–0.50 | 0.05 | 0.05 | .......... |
| A60-27 | Chrome-Vanadium-Steel Bars for Railway Springs | H, E, C | 0.50–0.60 | 0.60–0.90 | 0.045 | 0.045 | .......... |
| A59-27 | Silico-Manganese-Steel Bars for Railway Springs | H, E, C | 0.55–0.65 | 0.60–0.90 | 0.045 | 0.045 | .......... |
| A58-27 | Carbon Steel Bars for Vehicle and General Purpose Springs | | | | | | |
| | Grade A | H, E, C | 0.85–1.05 | 0.25–0.50 | 0.05[1, 2] | 0.05 | .......... |
| | Grade B | H, E, C | 0 90–1 05 | 0.25–0.50 | 0.05[1]; 0.04[2] | 0.05 | .......... |

*See footnotes at end of table.*

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| Chemical Composition | | Physical Properties | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Si, | Other Elements. See Footnotes | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | | Elongation, min., percent | | Red. of Area, percent | Bend Test No. See Table 9, p. 2–28 | Other Requirements. See Note No. |
| | | | Min. | But not less than | In 8 in. | In 2 in. | | | |
| 0.10–0.23 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | See paragraph on Drop Test for Carbon Steel Rails, on p. 2–37 |
| 0.10–0.23 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| 0.10–0.23 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| 0.10–0.23 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| 0.10–0 23 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| ...... | (3) | 60–72 | 0.5 T.S.[4] | 33 | ...... | 22 | ...... | 5, 17, 29, 36, 43 | 1–7 |
| ...... | (3) | 67–82 | 0.5 T.S.[4] | 36 | ...... | 20 | ...... | 7, 20, 32, 38, 46 | 1–7 |
| ...... | (3) | 52–62 | 0.5 T.S.[4] | 28 | Note 8 | ...... | ...... | 1 | 9–10 |
| ...... | (3) | 60–72 | 0.5 T.S.[4] | 28 | Note 11 | 22 | ...... | 6, 18, 30, 36, 43 | 1–7, 12, 13 |
| ...... | (3) | 55–65 | 0.5 T.S.[4] | 30 | Note 8 | 24 | ...... | 5a, 17, 28, 35, 42 | 1–7 |
| ...... | (3) | 50–65 | 0.5 T.S.[4] | ...... | Note 8 | 22 | ...... | 5a, 23, 25 | 1–7, 9, 13–15 |
| ...... | (3) | 55–65 | 0.5 T.S.[4] | ...... | Note 8 | 22 | ...... | 5a, 23, 25 | |
| ...... | (3) | 48–58 | 0.5 T.S.[4] | ...... | Note 8 | ...... | ...... | 5a, 23, 25 | |
| ...... | ...... | 60–70 | 0.5 T.S.[4] | 33 | Note 8 | 22 | ...... | 6, 24, 27 | 1–4, 7, 14–16 |
| ...... | ...... | 55–65 | 0.5 T.S.[4] | 30 | Note 8 | ...... | ...... | 5a, 23, 25 | 1–4, 7, 16 |
| 0.20[9, 15]<br>0.18[10, 15] | ......<br>...... | 80–95 | ...... | 45 | Note 8 | Note 17 | 30[11] | 6, 18, 27a, 26 | 1–6, 18 |
| 0.25[8] | (3) | 70–80 | 0.45 T.S. | 38 | Note 55 | 20 | ...... | ...... | 55, 56 |
| ...... | (3) | 90–115 | 0.5 T.S. | 55 | Note 55 | Note 57 | 30[28] | 7, 24, 26 | 1–4, 6, 7 |
| ...... | (3) | 85–100 | 48 | ...... | 10[29] | ...... | 30 | 7, 24, 36 | 1–4[30], 6, 7 |
| 0.05[8] | (16) | 45 | 0.5 T.S. | 24 | 28[14] | ...... | ...... | 1 | 1–4, 7, 9, 19 |
| 0.05[8] | (16) | 50 | 0.5 T.S. | 27 | 25[14] | ...... | ...... | 1 | 1–4, 7, 9, 19 |
| ...... | ...... | 55–70 | ...... | 33 | Note 20 | ...... | ...... | 6, 11 | 9, 26, 28 |
| ...... | ...... | 70–90 | ...... | 40 | Note 21 | ...... | ...... | 8, 12[21] | 9, 26, 28 |
| ...... | ...... | 80[15] | ...... | 50 | Note 22 | ...... | ...... | 9, 13[21] | 9, 26, 28 |
| ...... | ...... | 55–70 | ...... | 33 | Note 23 | ...... | ...... | 6, 11 | 9, 26, 28 |
| ...... | ...... | 70–90 | ...... | 40 | Note 24 | ...... | ...... | 9, 13[21] | 9, 26, 28 |
| ...... | ...... | 80[15] | ...... | 50 | Note 25[19] | ...... | ...... | 10, 14[21] | 9, 26, 28 |
| ...... | ...... | (18) | ...... | 55 | 5 | ...... | ...... | 8, 13 | 9, 27, 28 |
| ...... | ...... | 80 | ...... | 50 | Note 22 | ...... | ...... | 9, 13[21] | 26, 28, 29 |
| ...... | ...... | 80 | ...... | 50 | Note 25 | ...... | ...... | 10, 14[21] | 26, 27, 28, 29 |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 30 |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| 0.15[15] | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| ...... | (24) | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| 1.80–2.20 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

*Table continued on following page.*

## Table 6.—Specifications

Abstracted from A.S.T.M.

| A.S.T.M. Designation No. | Material | Process of Manufacture* | Chemical Composition | | | | |
|---|---|---|---|---|---|---|---|
| | | | C | Mn | P, max. | S, max. | Ni |
| A63–36 | Quenched-and-tempered Alloy-Steel Axles, Shaft and Forgings for Locos. and Cars.... | H, E | | | | | |
| Class K See Note 31 | Up to 2 in. diam. or thick, max. wall, 1 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 2–4 in. diam. or thick, max. wall, 2 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 4–7 in. diam. or thick, max. wall, 3 1/2 in.............. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 7–10 in. diam. or thick, max. wall 5 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 20 in. max. diam. or thick, 5–8 in. wall.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| Class L See Note 31 | Up to 2 in. diam. or thick, max. wall 1 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 2–4 in. diam. or thick, max. wall 2 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 4–7 in. diam. or thick, max. wall 3 1/2 in.............. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 7–10 in. diam. or thick, max. wall 5 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| | 20 in. diam. or thick, max. wall 5–8 in.................. | | .......... | .......... | 0.05[1]; 0.04[2] | 0.05 | .......... |
| A19–36 | Quenched-and-tempered Carbon-steel Axles, Shafts and Other Forgings for Locos. and Cars.......................... | H, E | | | | | |
| | Up to 4 in. diam. or thick, 2 in. max. wall.................. | | 0.25–0.60 | 0.40–0.70 | 0.05 | 0.05 | .......... |
| | 4–7 in. diam. or thick, 3 1/2 in. max. wall.................. | | 0.35–0.60 | 0.40–0.70 | 0.05 | 0.05 | .......... |
| | 7–10 in. diam. or thick, 5 in. max. wall.................. | | 0.35–0.65 | 0.40–0.70 | 0.05 | 0.05 | .......... |
| | 20 in. diam. or thick, max. wall 5–8 in.................. | | 0.35–0.70 | 0.40–0.70 | 0.05 | 0.05 | .......... |
| A17–29 | Carbon-steel and Alloy-steel Blooms, Billets and Slabs for Forgings.............. | H, E | | | | | |
| | Carbon Steel, Type A........ | | See Note 44 | 0.50–0.80 | 0.045 | 0.05 | .......... |
| | Alloy Steel, Type B.......... | | " | 0.50–0.80 | 0.04 | 0.045 | 3.00[15] |
| | " " Type C.......... | | " | 0.50–0.80 | 0.04 | 0.045 | 1.00–1.50 |
| | " " Type D.......... | | " | 0.30–0.60 | 0.04 | 0.045 | 1.50–2.00 |
| | " " Type E.......... | | " | 0.45–0.75 | 0.04 | 0.045 | 2.75–3.75 |
| | " " Type F.......... | | " | 0.30–0.60 | 0.04 | 0.045 | 3.00[15] |
| | " " Type G.......... | | " | 0.30–0.60 | 0.04 | 0.045 | .......... |
| | " " Type H.......... | | " | 0.50–0.80 | 0.04 | 0.045 | .......... |
| A26–16 | Steel Tires.................... | H, E | | | | | |
| | See Note 45 Class A......... | | 0.50–0.70 | 0.75[8] | 0.05 | 0.05 | .......... |
| | Class B......... | | 0.60–0.80 | 0.75[8] | 0.05 | 0.05 | .......... |
| | Class C......... | | 0.70–0.85 | 0.75[8] | 0.05 | 0.05 | .......... |
| A148–36 | Alloy Steel Castings for Structural Purposes............ | B, H<br>E, C | | | | | |
| | Class A, full annealed | | | | | | |
| | Grade 1.................... | | See Note | 47 | 0.05 | 0.06 | .......... |
| | " 2.................... | | | | 0.05 | 0.06 | .......... |
| | Class B, normalized, or normalized and tempered and drawn[37].............. | | | | | | |
| | Grade 1.................... | | See Note | 47 | 0.05 | 0.06 | .......... |
| | " 2.................... | | | | 0.05 | 0.06 | .......... |
| | " 3.................... | | | | 0.05 | 0.06 | .......... |
| | Class C, liquid quenched and tempered or drawn | | | | | | |
| | Grade 1.................... | | See Note | 47 | 0.05 | 0.06 | .......... |
| | " 2.................... | | | | 0.05 | 0.06 | .......... |
| | " 3.................... | | | | 0.05 | 0.06 | .......... |
| A128–33 | Austenitic Manganese-Steel Castings................... | B, H<br>C, E | 1.00–1.40 | 10.0[15] | 0.10 | 0.05 | .......... |

*See footnotes at end of table.*

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| Chemical Composition | | Physical Properties | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Si, | Other Elements. See Footnotes | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | | Elongation, min., percent | | Red. of Area, percent | Bend Test No. See Table 9, p. 2–28 | Other Requirements. See Note No. |
| | | | Min. | But not less than | In 8 in. | In 2 in. | | | |
| ...... | ...... | 95–115 | $70^{25}$ | ...... | ...... | 20 | 50 | 2 | 32, 33, 42 |
| ...... | ...... | 90–110 | $65^{25}$ | ...... | ...... | 20 | 50 | 2 | |
| ...... | ...... | 90–110 | $65^{25}$ | ...... | ...... | 20 | 50 | 2 | |
| ...... | ...... | 90–110 | $65^{25}$ | ...... | ...... | 20 | 50 | 3 | |
| ...... | ...... | 85–105 | $60^{25}$ | ...... | ...... | 20 | 50 | 3 | |
| ...... | ...... | 105–125 | $80^{25}$ | ...... | ...... | 20 | 50 | 2 | |
| ...... | ...... | 100–120 | $75^{25}$ | ...... | ...... | 20 | 50 | 2 | |
| ...... | ...... | 100–120 | $75^{25}$ | ...... | ...... | 20 | 50 | 2 | |
| ...... | ...... | 100–120 | $75^{25}$ | ...... | ...... | 18 | 45 | 3 | |
| ...... | ...... | 95–115 | $70^{25}$ | ...... | ...... | 18 | 45 | 3 | |
| ...... | ...... | 90 | $55^{25}$ | ...... | ...... | Note 34 | Note 35 | 2 | 32, 33, 42 |
| ...... | ...... | 85 | $50^{25}$ | ...... | ...... | Note 36 | Note 37 | 2 | |
| ...... | ...... | 85 | $50^{25}$ | ...... | ...... | Note 38 | Note 39 | 3 | |
| ...... | ...... | 82.5 | $48^{25}$ | ...... | ...... | Note 40 | Note 41 | 3 | |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | Cr=0.45–0.75 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | Cr=0.90–1.25 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | Cr=0.60–0 95 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | Cr= 1 $00^{15}$ | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | Cr=0.60–0.90 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | V=0.15; Cr=0.80–1.10 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| Si<br>0.15–0.35 | ...... | 105 | ...... | ...... | ...... | 12 | 16 | ...... | Note 46 |
| 0.15–0.35 | ...... | 115 | ...... | ...... | ...... | 10 | 14 | ...... | |
| 0.15–0.35 | ...... | 125 | ...... | ...... | ...... | 8 | 12 | ...... | |
| ...... | ...... | 75 | 40 | ...... | ...... | 24 | 35 | $2^{26}$ | 48, 51, 52 |
| ...... | ...... | 85 | 53 | ...... | ...... | 22 | 35 | $2^{26}$ | |
| ...... | ...... | 85 | 55 | ...... | ...... | 22 | 40 | ...... | 49, 51, 52 |
| ...... | ...... | 90 | 60 | ...... | ...... | 22 | 45 | ...... | |
| ...... | ...... | 100 | 65 | ...... | ...... | 18 | 30 | ...... | |
| ...... | ...... | 90 | 65 | ...... | ...... | 20 | 40 | ...... | 50, 51, 52 |
| ...... | ...... | 120 | 100 | ...... | ...... | 14 | 35 | ...... | |
| ...... | ...... | 150 | 125 | ...... | ...... | 10 | 25 | ...... | |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | $2^{27}$ | 53, 54 |

*Table continued on following page.*

**Table 6.—Specifications**

Abstracted from A.S.T.M.

| A.S.T.M. Designation No. | Material | Process of Manufacture* | Chemical Composition | | | | |
|---|---|---|---|---|---|---|---|
| | | | C | Mn | P, max. | S, max. | Ni |
| A27–36T † | Carbon Steel Castings....... | H, E<br>B, C | | | | | |
| | Grade A1, unannealed........ | | ......... | 1.00 | 0.05 | 0.06 | ........... |
| | " A2, normalized........ | | | | | | |
| | " A3, full annealed...... | | | | | | |
| | " B, normalized......... | | | | | | |
| | " B1, full annealed...... | | | | | | |
| | " B2, " " ...... | | | | | | |
| | " H, normalized......... | | | | | | |
| | " H1, full annealed...... | | | | | | |
| | Nickel-Steel and Alloy-Steel Steel Castings: | | | | | | |
| A168–35T † | 12% Nickel-Steel Castings.... | E | 0.15$^{8,31}$ | 0.25–0.75 | 0.05 | 0.05 | 0.80$^{8}$ |
| A169–35T † | 19% Chromium-Steel Castings | E | 0.30$^{8}$ | 0.25–0.75 | .05 | .05 | 1.00$^{8}$ |
| A170–35T † | 28% Chromium-Steel Castings | E | 0.25–0.50 | 0.25–0.75 | .05 | .05 | 3.00$^{8}$ |
| A198–36T † | 20% Cr-9% Ni-Steel Castings Grade A.................. | E | 0.07$^{8}$ | 1.00$^{8}$ | .05 | .05 | 8.00$^{15}$ |
| | 20% Cr-9% Ni-Steel Castings, Grade B.................. | E | 0.20$^{8}$ | 1.00$^{8}$ | .05 | .05 | 8.00$^{15}$ |
| A171–35T † | 24% Cr-12% Ni-Steel Castings | E | 0.25$^{8,32}$ | 0.25–1.00 | .05 | .05 | 11.0–13.0 |
| A172–35T † | 25% Cr-20% Ni-Steel Castings | E | 0.30$^{8,32}$ | 0.25–1.00 | .05 | .05 | 19.0–22.0 |
| A173–35T † | 28% Cr-9% Ni-Steel Castings | E | 0.30$^{8,32}$ | 0.25–1.00 | .05 | .05 | 8.0–10.0$^{34}$ |
| A174–35T † | 20% Ni-9% Cr-Steel Castings | E | 0.20$^{8,33}$ | 0.25–1.00 | .05 | .05 | 19.0–22.0 |
| A175–35T † | 35% Ni-15% Cr-Steel Castings | E | 0.60$^{8}$ | 0.25–1.00 | .05 | .05 | 34.0–37.0 |
| A87–36 | Carbon-Steel and Alloy-Steel Castings for Railroads: | B, H<br>E, C | | | | | |
| | Grade A–1, Carbon Steel..... | | ........... | 0.85$^{8}$ | 0.05 | 0.06$^{1}$; 0.05$^{2}$ | ........... |
| | " A–2, " " ..... | | ........... | 0.85$^{8}$ | 0.05 | .06$^{1}$; .05$^{2}$ | ........... |
| | " B " " ..... | | ........... | 0.85$^{8}$ | 0.05 | .06$^{1}$; .05$^{2}$ | ........... |
| | " C " " ..... | | ........... | ......... | 0.05 | .05 | ........... |
| A107–36 | Commercial Hot-rolled Bar Steel | | See p. 2–32 | | | | |
| A108–36 | Commercial Cold-finished Bar Steel and Cold-finished Shafting........................ | | See p. 2–31 | | | | |
| A18–30 | Carbon-Steel and Alloy Steel Forgings.................. | | See p. 2–29 | | | | |

* B = Bessemer; H = open-hearth; E = electric furnace; C = crucible.

† These are tentative specifications and subject to yearly revision. The latest issue of the A.S.T.M. standards should be consulted.

[1] Acid steel

[2] Basic steel

[3] Cu, if specified, 0.20

[4] T.S. = tensile strength

[5] Bessemer steel

[6] Open-hearth steel

[7] Electric furnace steel

[8] Maximum

[9] Ladle analysis

[10] Check analysis

[11] For material over 3/4 in. thick, deduct 0.5% for each increase of 1/16 in. of specified thickness or diam. above 3/4 in. to a minimum of 24%

[12] Plates 3/4 in. thick or less

[13] Plates over 3/4 in. thick

[14] For material over 3/4 in. thick, deduct 0.25% for each increase of 1/32 in. over the specified thickness above 3/4 in. to a minimum of 20%

[15] Minimum

[16] Ni and Cr, 0.05 max., each. Preferably should be absent

[17] To be twisted with 1 twist in a length not more than 12 times thickness of bar

## Notes for Table 6

**Note 1.**—Test specimens shall be prepared for testing from material in the forged or rolled condition, except that specimens for annealed material, except eyebars, shall be from material as annealed for use, or from a short length of a full section from the same melt similarly treated. They shall be taken longitudinally and except as specified in notes 3, 4, and 5, shall be of full thickness of material as rolled.

**Note 2.**—Test specimens for plates, shapes and flats may be machined to form and dimensions of Fig. 2 (p. 7–54) or with both edges parallel.

**Note 3.**—Tension test specimens for material over 1 1/2 in. thickness or diameter, except pins and rollers, may be machined to a thickness or diameter of at least 3/4 in. for a length of at least 9 in., or may conform to dimensions of Fig. 2 (p. 7–54). For pins and rollers specimens shall conform to Fig. 2 (p. 7–54).

**Note 4.**—Bend test specimens: For material over 1 1/2 in. thickness (1 1/4 in. thick for A94–36) or diameter, except pins and rollers, may be machined to thickness or diameter of at least 3/4 in., or to a 1 × 1/2 in. section. For pins and rollers, specimens shall be

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| Chemical Composition | | Physical Properties | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | Yield Point, 1000 lb. per sq. in. | | Elongation, min., percent | | | | |
| Si, | Other Elements. See Footnotes | Tensile Strength, 1000 lb. per sq. in. | Min. | But not less than | In 8 in. | In 2 in. | Red. of Area, percent | Bend Test No. See Table 9, p. 2–28 | Other Requirements. See Note No. |
| ........... | ......... | 60 | 30 | ..... | ......... | 22 | 30 | ............... | 51, 52, 58 |
| | | 60 | 30 | ..... | ......... | 26 | 38 | ............... | 49, 51, 52, 58 |
| | | 60 | 30 | ..... | ......... | .24 | 35 | ............... | 48, 51, 52, 58 |
| | | 70 | 38 | ..... | ......... | 24 | 36 | ............... | 49, 51, 52, 59 |
| | | 66 | 33 | ..... | ......... | 22 | 33 | ............... | 48, 51, 52, 59 |
| | | 70 | 35 | ..... | ......... | 20 | 30 | ............... | 48, 51, 52, 59 |
| | | 80 | 43 | ..... | ......... | 17 | 25 | ............... | 49, 51, 52, 60 |
| | | 80 | 40 | ..... | ......... | 17 | 25 | ............... | 48, 51, 52, 60 |
| | Cr. | | | | | | | Brinell Hardness No., max. | |
| 1.00[8] | 11.5–14.0 | 80[15] | 50[15] | ..... | ......... | 20[15] | 30[15] | 215 | 53, 61, 62, 65, 66 |
| 1.00[8] | 17.0–21.0[35] | 90[15] | 55[15] | ..... | ......... | 5[15] | 5[15] | 230 | 53, 61, 63, 65, 67 |
| 0.80[8] | 26.0–30.0 | 55[15] | 40[15] | ..... | ......... | ......... | ....... | 230 | 53, 61, 63, 65, 68 |
| 2.00[8] | 18.0[15, 35] | 70[15] | 30[15] | ..... | ......... | 35[15] | 40[15] | 200 | 53, 61, 64, 65, 69 |
| 2.00[8] | 18.0[15, 35] | 70[15] | ........... | ..... | ......... | ......... | ....... | ............... | 53, 61, 64, 65, 69 |
| 0.25–1.50 | 22.0–26.0 | 70[15] | 0.45 T.S.[15] | ..... | ......... | 30[15] | 30[15] | 200 | 53, 61, 64, 65, 70 |
| 0.25–1.50 | 23.0–27.0 | 72.5[15] | 0.45 T.S.[15] | ..... | ......... | 25[15] | 25[15] | 225 | 53, 61, 62, 65, 70 |
| 0.25–1.50 | 26.0–30.0 | 85[15] | 0.45 T.S.[15] | ..... | ......... | 20[15] | 20[15] | 225 | 53,61,62,65,71,73 |
| 0.25–2.00 | 8.0–10.0 | 65[15] | 30[15] | ..... | ......... | 30[15] | 30[15] | ............... | 53, 61, 64, 65, 72 |
| 0.50–1.75 | 14.0–17.0 | 60[15] | 0.45 T.S.[15] | ..... | ......... | ......... | ....... | 185 | 53, 61, 63, 65, 71 |
| ........... | ........... | 60[15] | 30[15] | ..... | ......... | 22[15] | 30[15] | ............... | 53, 74, 75 |
| ........... | ........... | 60[15] | 30[15] | ..... | ......... | 26[15] | 38[15] | ............... | |
| ........... | ........... | 70[15] | 38[15] | ..... | ......... | 24[15] | 36[15] | ............... | |
| ........... | ........... | 90[15] | 55[15] | ..... | ......... | 22[15] | 40[15] | ............... | |

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[18] Recorded only

[19] But not less than 14%

[20] Bars rolled from standard section tee rails

[21] 90-deg. bend

[22] For elliptical and helical springs

[23] For helical springs

[24] Cr = 0.90–1.10; V = 0.15 min.

[25] Elastic limit; determined by extensometer. See Note 43

[26] Through 120 deg.

[27] Through 150 deg.

[28] For material over 3/4 in. thickness or diameter, deduct 0.50% for each increase of 1/16 in. over the specified thickness or diameter

[29] In 18 ft.

[30] Bend test specimen for eyebar flats over 1 1/2 in. thick may be 1/2 in. sq. section. Corners may be rounded to a maximum radius of 1/16 in.

[31] If C exceeds 0.12%, Cr should be not less than 12.50%.

[32] For heat-resistant applications, C may be 0.40 max.

[33] For heat-resistant applications, C may be 0.35 max.

[34] For certain applications, Ni = 10.0–12.0% is desirable

[35] For particular and severe conditions, Mo = 2.00–4.00% sometimes is used as a constituent of this alloy. For free-machining or better non-seizing qualities, suitable combinations of Mo and S, or of Se and P, may be added as follows: Mo, 0.40–0.80; S, 0.20–0.40; Se, 0.20–0.35; P, max., 0.17.

[36] Cu = 0.90–1.20% required for heat-resistant castings.

[37] Also may be full annealed. See Note 48, p. 2–31.

1 × 1/2 in. section. Machined sides of rectangular specimens may have corners rounded to a radius of not over 1/16 in.

**Note 5.**—Test specimens for pins and rollers shall be so taken that axis is 1 in. from the surface of the material.

**Note 6.**—Number of tests: Two tension and two bend tests to be made from each melt; except that if material from one melt differs 3/8 in. or more in thickness, one tension and one bend test shall be made from both the thickest and the thinnest material rolled.

**Note 7.**—Variation of weight of plates: Cross-section or weight of each size structural shape, and each universal mill or sheared plate up to 36 in. wide and 2 in. thick, not to vary more than 2.5% from that specified. Sheared and universal mill plates may vary as follows, 1 cu. in. of rolled steel being assumed to weigh 0.2833 lb.: *a.* If ordered to weight per sq. ft., the weight of each lot in each shipment shall not vary from the weight ordered by more than the amount given in Table 7. *b.* If ordered to thickness, the thickness shall not vary more than 0.01 in. under that ordered. The overweight of each lot in each shipment shall not exceed the amount given in Table 8.

### Table 7.—Permissible Variations of Plates Ordered to Weight*

| Ordered Weight, lb. per sq. ft. | Permissible variations in average weights per sq. ft. of plates for widths given, expressed in percentages of ordered weights. | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Under 48 in. | | 48 to 60 in., excl. | | 60 to 72 in., excl. | | 72 to 84 in., excl. | | 84 to 96 in., excl. | | 96 to 108 in., excl. | | 108 to 120 in., excl. | | 120 to 132 in., excl. | | 132 in. or over. | |
| | Over. | Under. | Over. | Under. | Over. | Under. | Over. | Under. | Over. | Under. | Over. | Under. | Over. | Under. | Over. | Under. | Over. | Under. |
| Under 5 | 5 | 3 | 5.5 | 3 | 6 | 3 | 7 | 3 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 5 to 7.5 excl. | 4.5 | 3 | 5 | 3 | 5.5 | 3 | 6 | 3 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 7.5 " 10 " | 4 | 3 | 4.5 | 3 | 5 | 3 | 5.5 | 3 | 6 | 3 | 7 | 3 | 8 | 3 | ... | ... | ... | ... |
| 10 " 12.5 " | 3.5 | 2.5 | 4 | 3 | 4.5 | 3 | 5 | 3 | 5.5 | 3 | 6 | 3 | 7 | 3 | 8 | 3 | 9 | 3 |
| 12.5 " 15 " | 3 | 2.5 | 3.5 | 2.5 | 4 | 3 | 4.5 | 3 | 5 | 3 | 5.5 | 3 | 6 | 3 | 7 | 3 | 8 | 3 |
| 15 " 17.5 " | 2.5 | 2.5 | 3 | 2.5 | 3.5 | 2.5 | 4 | 3 | 4.5 | 3 | 5 | 3 | 5.5 | 3 | 6 | 3 | 7 | 3 |
| 17.5 " 20 " | 2.5 | 2 | 2.5 | 2.5 | 3 | 2.5 | 3.5 | 2.5 | 4 | 3 | 4.5 | 3 | 5 | 3 | 5.5 | 3 | 6 | 3 |
| 20 " 25 " | 2 | 2 | 2.5 | 2 | 2.5 | 2.5 | 3 | 2.5 | 3.5 | 2.5 | 4 | 3 | 4.5 | 3 | 5 | 3 | 5.5 | 3 |
| 25 " 30 " | 2 | 2 | 2 | 2 | 2.5 | 2 | 2.5 | 2.5 | 3 | 2.5 | 3.5 | 3 | 4 | 3 | 4.5 | 3 | 5 | 3 |
| 30 " 40 " | 2 | 2 | 2 | 2 | 2 | 2 | 2.5 | 2 | 2.5 | 2.5 | 3 | 2.5 | 3.5 | 3 | 4 | 3 | 4.5 | 3 |
| 40 or over | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2.5 | 2 | 2.5 | 2.5 | 3 | 2.5 | 3.5 | 3 | 4 | 3 |

* The weight per square foot of individual plates shall not vary from the ordered weight by more than 1 1/3 times the amount given in this table.

### Table 8.—Permissible Overweights of Plates Ordered to Thickness*

| Ordered Thickness, in. | Permissible excess in average weights per square foot of plates for widths given, expressed in percentages of nominal weights. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Under 48 in. | 48 to 60 in., excl. | 60 to 72 in., excl. | 72 to 84 in., excl. | 84 to 96 in., excl. | 96 to 108 in., excl. | 108 to 120 in., excl. | 120 to 132 in., excl. | 132 in. or over. |
| Under 1/8 | 9 | 10 | 12 | 14 | ........ | ........ | ........ | ........ | ........ |
| 1/8 to 3/16 excl. | 8 | 9 | 10 | 12 | ........ | ........ | ........ | ........ | ........ |
| 3/16 " 1/4 " | 7 | 8 | 9 | 10 | 12 | ........ | ........ | ........ | ........ |
| 1/4 " 5/16 " | 6 | 7 | 8 | 9 | 10 | 12 | 14 | 16 | 19 |
| 5/16 " 3/8 " | 5 | 6 | 7 | 8 | 9 | 10 | 12 | 14 | 17 |
| 3/8 " 7/16 " | 4.5 | 5 | 6 | 7 | 8 | 9 | 10 | 12 | 15 |
| 7/16 " 1/2 " | 4 | 4.5 | 5 | 6 | 7 | 8 | 9 | 10 | 13 |
| 1/2 " 5/8 " | 3.5 | 4 | 4.5 | 5 | 6 | 7 | 8 | 9 | 11 |
| 5/8 " 3/4 " | 3 | 3.5 | 4 | 4.5 | 5 | 6 | 7 | 8 | 9 |
| 3/4 " 1 " | 2.5 | 3 | 3.5 | 4 | 4.5 | 5 | 6 | 7 | 8 |
| 1 or over | 2.5 | 2.5 | 3 | 3.5 | 4 | 4.5 | 5 | 6 | 7 |

* The weight of individual plates ordered to thickness shall not exceed the nominal weight by more than 1 1/3 times the amount given in this table.

### Table 9.—Bend Tests for Specifications Given in Table 6

Test specimen to bend cold 180 deg., unless otherwise noted in table, around a pin of diameter given, without cracking on outside of bend.

| Bend Test No. | Material Thickness, *t* | *d** | Bend Test No. | Material Thickness, *t* | *d*/*t** | Bend Test No. | Material Thickness, *t* | *d*/*t** | Bend Test No. | Material Thickness, *t* | *d*/*t** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | ........ | 0† | 11 | 3/4 in. and over | 1 | 23 | 3/4-1 1/4 in. incl. | 1 | 33 | 1 1/2–2 in. | 1 |
| 2 | ........ | 1 in. | 12 | " | 2 | 24 | " | 2 | 34 | " | 1 1/2 |
| 3 | ........ | 1 1/2 in. | 13 | " | 3 | 25 | Over 1 1/4 in. | 2 | 35 | " | 2 |
| 4 | ........ | 2 in. | 14 | " | 4 | 26 | " | 2 1/2 | 36 | " | 2 1/2 |
| | | *d*/*t* | 15 | 3/4–1 in. incl. | 0† | 27 | " | 3 | 37 | " | 3 |
| 5a | up to 3/4 in. inc. | 0† | 16 | " | 0.5 | 27a | 1–1 1/4 in. | 2 | 38 | " | 4 |
| 5 | " | 0.5 | 17 | " | 1 | 28 | 1–1 1/2 in. | 1 | 39 | Over 2 in. | 1 |
| 6 | " | 1. | 18 | " | 1 1/2 | 29 | " | 1 1/2 | 40 | " | 1 1/2 |
| 7 | " | 1 1/2 | 19 | " | 2 | 30 | " | 2 | 41 | " | 2 |
| 8 | " | 2 | 20 | " | 2 1/2 | 31 | " | 2 1/2 | 42 | " | 2 1/2 |
| 9 | " | 3 | 21 | " | 3 | 32 | " | 3 | 43 | " | 3 |
| 10 | " | 4 | 22 | " | 4 | | | | 44 | " | 3 1/2 |
| | | | | | | | | | 45 | " | 4 |
| | | | | | | | | | 46 | " | 4 1/2 |

* *d* = diameter of pin. † Specimen to bend flat on itself without cracking on outside of bend.

**Table 10.—Permissible Variation Over Ordered Thickness of Plates Over 2 in. Thick**

| Ordered Thickness, in. | Width, in. | | | | | |
|---|---|---|---|---|---|---|
| | Up to 36 excl. | 36–60 excl. | 60–84 excl. | 84–120 excl. | 120–132 excl. | 132–144 excl. |
| | Permissible excess thickness, in. | | | | | |
| Over 2 to 3 excl. | 0.062 (1/16) | 0.094 (3/32) | 0.109 (7/64) | 0.125 (1/8) | 0.125 (1/8) | 0.141 (9/64) |
| 3 " 4 " | .078 (5/64) | .094 (3/32) | .109 (7/64) | .125 (1/8) | .125 (1/8) | .141 (9/64) |
| 4 " 6 " | .094 (3/32) | .125 (1/8) | .141 (9/64) | .141 (9/64) | .156 (5/32) | .172 (11/64) |
| 6 " 8 " | .109 (7/64) | .125 (1/8) | .156 (5/32) | .156 (5/32) | .172 (11/64) | .......... |
| 8 " 10 " | .......... | .172 (11/64) | .187 (3/16) | .187 (3/16) | .......... | .......... |
| 10 " 12 " | .......... | .187 (3/16) | .234 (15/64) | .234 (15/64) | .......... | .......... |
| 12 " 15 " | .......... | .219 (7/32) | .250 (1/4) | .......... | .......... | .......... |

**Note 8.***—Elongation in 8 in., % min., = 1,500,000/tensile strength.

**Note 9.**—One tension and one bend test to be made from each melt; if two bars from one melt differ 3/8 in. or more in thickness or diameter, one tension and one bend test shall be made from both greatest and least diameters rolled. Test specimens to be full diameter of bars as rolled. Cold-drawn bars to be normalized before testing.

**Note 10.**—Permissible variations in diameter of rivet bars is as follows:

| Diameter...... | | Up to 1/2 in. incl. | Over 1/2 up to 1 in. incl. | Over 1 up to 2 in. incl. |
|---|---|---|---|---|
| Permissible... Variation.... | over | 0.007 in. | 0.010 in. | 1/32 in. |
| | under | 0.007 in. | 0.010 in. | 1/64 in. |

**Note 11.***—Elongation in 8 in., %, min., = 1,400,000/tensile strength.

**Note 12.**—Steel for plates or shapes over 7/16 in. thick which are to be punched, are to be made only by open-hearth or electric furnace processes.

**Note 13.**—Plates over 2 in. thick shall conform to permissible variations over ordered thickness as given in Table 10.

**Note 14.**—Permissible variation in weight, expressed as percentage of nominal weight, of bar size angles, tees, zees and channels not to exceed the following: Thickness over 3/16 in., any dimension over 1 1/2 in., variation, ± 4%; all dimensions 1 1/2 in. and less, variation, ± 5%. Thickness 3/16 in. and less, any dimension over 1 1/2 in., variation, ± 6%; all dimensions 1 1/2 in. and less, variation, ± 7%.

**Note 15.**—Permissible variation in cross-sectional dimensions of rounds, squares, and hexagons, not to exceed the following:

| Dimensions, in............ | up to 1/2 | 1/2–1 | 1–2 | 2–3 | 3–5 | 5–8 |
|---|---|---|---|---|---|---|
| Variation over, in.......... | 0.007 | 0.010 | 1/32 | 3/64 | 3/32 | 1/8 |
| Variation under, in......... | 0.007 | 0.010 | 1/64 | 1/32 | 1/32 | 1/16 |

Permissible variation in cross-sectional dimensions of flats not to exceed the following:

| Dimensions, in............... | Up to 1* | 1†–2* | 2†–4* | 4†–6* |
|---|---|---|---|---|
| Variation in width, in......... | + 1/32, − 1/64 | + 3/64, − 1/32 | + 1/16, − 3/64 | + 3/32, − 1/16 |
| Variation in thickness, in. | | | | |
| 3/16 in. thick and under..... | ±0.006 | ±0.008 | ±0.010 | ±0.010 |
| 3/16–1/2 in.*†.............. | ±0.008 | ±0.012 | ±0.015 | ±0.015 |
| 1/2–1 in.*†................ | ±0.010 | ±0.015 | ±0.020 | ±0.020 |
| 1–2 in.*.†................. | ....... | ± 1/32 | ± 1/32 | ± 1/32 |

**Note 16.**—Two tension tests and two cold bend tests shall be made from each melt of structural steel and steel for cold bending, except that one tension and one cold bend test is sufficient for melts of less than 25 tons. Two tension and two cold bend tests shall be made from each melt of rivet steel. If material from one melt varies 0.0150 in. or more in thickness, the thickest and thinnest material shall each be represented by one tension and one bend test, regardless of the weight of the melt.

**Note 17.**—Minimum elongation in 2 in. = 1,600,000/tensile strength, when specimen shown in Fig. 3, p. 7–54, is used.

**Note 18.**—Cross-section or weight of each piece of steel shall not vary more than 2.5% from that specified, except that sheared plates, ordered by weight per sq. ft., shall not vary more than the amount given by Table 7. When ordered to thickness, no plate shall vary more than 0.01 in. under thickness specified. Overweight in each shipment shall not exceed the values given in Table 8.

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* For structural steel over 3/4 in. thick, 0.25% is to be deducted from the percentage specified by the formula, for each increase of 1/32 in. of the specified thickness above 3/4 in., to a minimum of 18%. For steel under 5/16 in. thick, 1.25% shall be deducted for each decrease of 1/32 in. of the specified thickness below 5/16 in.

† For plain and deformed bars over 3/4 in. diameter or thickness, a deduction from the percentage of elongation of 0.25% shall be made for each increase of 1/32 in. of specified thickness or diameter above 3/4 in. For bars under 7/16 in. diameter or thickness, a deduction of 0.5% shall be made for each 1/32 in. decrease of thickness or diameter below 3/16 in.

**Note 19.**—Permissible variation in thickness of plates ordered to weight is not more than 0.01 in. under thickness specified. Overweight in each shipment shall not exceed values given in Table 8.

**Note 20.***—Elongation in 8 in., %, min., = 1,400,000/tensile strength.

**Note 21.***—Elongation in 8 in., %, min., = 1,300,000/tensile strength, but not less than 16%.

**Note 22.***—Elongation in 8 in., %, min., = 1,200,000/tensile strength.

**Note 23.***—Elongation in 8 in., %, min., = 1,250,000/tensile strength.

**Note 24.***—Elongation in 8 in., %, min., = 1,125,000/tensile strength.

**Note 25.***—Elongation in 8 in., %, min., = 1,000,000/tensile strength.

**Note 26.**—Tension and bend test specimens from plain or deformed bars shall be of full section of bars rolled. For tension tests of deformed bars, sectional area used for unit stress determination to be calculated from weight of test piece.

**Note 27.**—Tension and bend test pieces for cold twisted bars to be taken from finished bar without further treatment.

**Note 28.**—Permissible variation in weight: Bars $^3/_8$ in. diameter or over, ± 3.5%; bars under $^3/_8$ in. diameter, ± 5%.

**Note 29.**—One tension and one bend test to be made from each lot of 10 tons or less of each size of bar rolled from rails varying not more than 10 lb. per yd. in nominal weight.

**Note 30.**—Permissible variation in diameter or thickness of rounds and squares is as follows: Up to $^1/_2$ in. incl., ± 0.007 in.; over $^1/_2$ to 1 in. incl., ± 0.010 in.; over 1 to 2 in. incl., ± 0.016 in. Permissible variation in dimension of flats: Variation in width, flats 1 to 2 in. incl., + 0.030 in., − 0.015 in.; flats over 2 to 4 in. incl., + 0.047 in., − 0.015 in.; flats over 4 to 6 in. incl., + 0.062 in., − 0.032 in. Variation in thickness, all widths, up to $^3/_8$ in. thick incl., + 0.015, − 0.005; over $^3/_8$ to $^5/_8$ in. thick, incl., + 0.015 in., − 0.010.

**Note 31.**—Class K for forgings for main- and side-rods, straps, piston rods and all other forgings which are to be machined with milling cutters or complicated forming tools.

**Note 32.**—Tension and bend test specimens to be taken from full size prolongation of any forging; may be taken from forging itself with a hollow drill, if approved by purchaser. Axis of specimen shall be located at any point midway between center and surface of solid forgings, midway between inner and outer surfaces of bored forgings, and parallel to axis of forging in direction in which metal is most drawn out. Tension test specimens to conform to dimensions of Fig. 2 (p. 7–54). Bend test specimens to be $^1/_2$ in. square section, with corners rounded to radius of not over $^1/_{16}$ in.; need not be over 6 in. long.

**Note 33.**—One tension test and one bend test (if specified) to be made from each tempering charge, or from each quenching charge if more than one is represented in a tempering charge. If more than one melt is represented, tests shall be made from each melt. If more than one class of forgings (by size) is represented, tests shall be made from each class.

**Note 34.**—Elongation in 2 in. = (2,100,000/tensile strength), but not under 20.5%.

**Note 35.**—Reduction of area = (4,100,000/tensile strength), but not under 39%.

**Note 36.**—Elongation in 2 in. = (2,000,000/tensile strength), but not under 20.5%.

**Note 37.**—Reduction of area = (3,800,000/tensile strength), but not under 39%.

**Note 38.**—Elongation in 2 in. = (1,900,000/tensile strength), but not under 19.5%.

**Note 39.**—Reduction of area = (3,600,000/tensile strength), but not over 37%.

**Note 40.**—Elongation in 2 in. = (1,800,000/tensile strength), but not under 19%.

**Note 41.**—Reduction of area = (3,400,000/tensile strength), but not under 36%.

**Note 42.**—Impact proof test required. Three methods of proof testing for quenched and tempered forgings are given in Table 11, which represent practices of four companies. Practice A requires axles, shafts and similar forgings to undergo an impact proof test on a A.R.A. drop test machine, with supports spaced 3 ft. center to center. Two blows are struck with a tup weighing 1640 or 2240 lb. The forging is to have one of the points of support as near as practicable to one end for the first blow and as near as practicable to the other end for the second blow, the forging being turned 90 deg. on its longitudinal axis between blows. Practices B and C require forgings to have an impact proof test of one blow at the center of the forging by a tup delivering the number of ft.-lb. of energy shown in Table 11. Practice B carries the forging on rigid supports placed 5 ft. center to center. Practice C carries the forging on rigid supports at the end of the forging.

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* For plain and deformed bars over $^3/_4$ in. diameter or thickness, a deduction from the percentage of elongation of 0.25% shall be made for each increase of $^1/_{32}$ in. of specified thickness or diameter above $^3/_4$ in. For bars under $^7/_{16}$ in. diameter or thickness, a deduction of 0.5% shall be made for each $^1/_{32}$ in. decrease of thickness or diameter below $^3/_{16}$ in.

**Table 11.—Proof Tests for Quenched-and-tempered Forgings**

| Diameter, in. | Practice A | | | | Practice B<br>Practice C | Diameter, in. | Practice A | | | | Practice B<br>Practice C |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Height of Drop | | | | Energy of Blow, ft.-lb. | | Height of Drop | | | | Energy of Blow, ft.-lb. |
| | 1640-lb. Tup. | | 2240-lb. Tup. | | | | 1640-lb. Tup. | | 2240-lb. Tup. | | |
| | Ft. | In. | Ft. | In. | | | Ft. | In. | Ft. | In. | |
| 4 1/2 | 0 | 11 | .. | .. | 700 | 8 1/2 | 6 | 2 | 4 | 6 | 7,500 |
| 5 | 1 | 3 | .. | .. | 900 | 9 | 7 | 3 | 5 | 3 | 9,000 |
| 5 1/2 | 1 | 8 | 1 | 2 | 1,200 | 9 1/2 | 8 | 7 | 6 | 3 | 10,000 |
| 6 | 2 | 2 | 1 | 7 | 1,700 | 10 | 10 | 0 | 7 | 4 | 11,000 |
| 6 1/2 | 2 | 9 | 2 | 0 | 2,500 | 10 1/2 | 11 | 7 | 8 | 6 | 12,000 |
| 7 | 3 | 5 | 2 | 6 | 3,500 | 11 | 13 | 5 | 9 | 10 | 13,000 |
| 7 1/2 | 4 | 3 | 3 | 1 | 4,700 | 11 1/2 | 15 | 3 | 11 | 1 | 14,000 |
| 8 | 5 | 1 | 3 | 8 | 6,000 | 12 | 17 | 4 | 12 | 8 | 15,000 |

**Note 43.**—The elastic limit shall be determined by an extensometer, reading to 0.0002 in., attached to the specimen at the gage markings. The testing machine shall be operated to increase the load on the specimen at a uniform rate. The load at which the rate of elongation shows a sudden increase determines the elastic limit.

**Note 44.**—Each type of steel is subdivided, according to the carbon content into seven grades of carbon steel, and eight grades of each type of alloy steel. Carbon ranges are as follows:

Carbon Steel. Type A

| Grade........ | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| Carbon, %.... | 0.05–.15 | .15–.25 | .20–.30 | .25–.40 | .30–.45 | .35–.50 | .40–.55 | .45–.60 |

Alloy Steels, Grades B–H

| Grade........ | 11 | 12 | 13 | 14 | 15 | 16 | 17 |
|---|---|---|---|---|---|---|---|
| Carbon, %.... | 0.10–.20 | .15–.25 | .20–.30 | .25–.38 | .30–.43 | .35–.50 | .45–.60 |

When steel is to be used for case hardening purposes, manganese should not exceed 0.50%; when minimum carbon specified is 0.35% or over, manganese range may be specified as 0.30–0.60%.

**Note 45.**—Steel Tires: Class A, for driving tires for passenger locomotives; Class B, for driving tires for freight locomotives; tires for locomotive-truck, tender-truck, trailer and car wheels; Class C, driving tires for switching locomotives.

**Note 46.**—Tension test specimens representing each melt to be taken from test ingot taken during pouring of melt; each specimen shall have received approximately the same amount of work as tire which it represents. Test specimens to be as in Fig. 3,* the ends to be of a form to fit the holders of the testing machine in such a way as to cause the load to be axial. Tires to be grouped as to outside diameters and shipped in sets.

**Note 47.**—Unless otherwise specified, the content of carbon, silicon and manganese and other alloying elements shall be selected by the manufacturer to meet specified physical properties. The prescription of definite composition types of steel to meet various physical requirements is not to be recommended except by mutual agreement between purchaser and manufacturer. There are at present (1936) approximately 70 available alloy cast steels, many of which can be made to conform to the tensile requirements of a grade in any class listed by proper heat treatment. See Automotive Steels, pp. 2–37 to 2–48.

**Note 48.**—Full annealing consists allowing the casting to cool after pouring, to a temperature below the critical range, reheating to the proper temperature and cooling thereafter slowly in the furnace. Castings shall not be removed from the furnace until the furnace charge has fallen uniformly to at least 500° F.

**Note 49.**—Normalizing consists of allowing the castings to cool after pouring, reheating to the proper temperature, above the critical range, holding at that temperature for the proper time,† and cooling in still air at room temperature. Normalizing may be followed by a tempering or drawing treatment at a temperature below the critical range, but not less than 400° F.

**Note 50.**—Class C castings shall be heat treated by full annealing, or by normalizing, or by heating above the critical temperature, followed by rapid cooling. These treatments may be followed by a supplementary heating and rapid cooling. Rapid cooling may be by quenching, by spraying with a liquid, or by subjecting castings to a mechanical air blast. These operations may be performed one or more times. After rapid cooling, castings shall be tempered or drawn at a temperature of 400° F. or more to relieve cooling stresses. Liquid-quenched castings, excepting simple light section designs, shall be removed from the quenching bath before they attain room temperature, and immediately placed in a tempering or drawing furnace.

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* For shape of tension test specimens, see Fig. 3, p. 7–54, under Standard Test Pieces—Methods of Testing.

† A common rule is 1 hr. per 1 in. thickness.

**Note 51.**—Tension test specimens and bend test specimens shall be prepared from test bars attached to castings. If this is impracticable, test bars shall be cast attached to special blocks. Test bars shall remain attached to the castings or blocks until submitted for inspection and shall be heat treated with the castings. Tension test specimens shall conform to dimensions of Fig. 3, p. 7–54. The end shall be of a form to fit the holders of the testing machine in such a way that the load shall be axial.

**Note 52.**—One tension test, and when specified, one bend test shall be made from each melt in each heat treatment charge.

**Note 53.**—An analysis of each melt of steel shall be made from drillings taken at least $^1/_4$ in. below the surface of a test ingot obtained during the pouring of the melt.

Purchaser may make analyses on samples obtained from a casting (A128–33), which shall be taken not less than $^1/_4$ in. below the surface of rough castings, or at the surface of castings finished by grinding or machining, provided that at least $^1/_4$ in. of metal has been removed from the casting at the point selected for sampling. Manganese steel may be drilled best after it has been annealed for several hours at from 900 to 1100° F.

**Note 54.**—Bend tests shall be required only when specified. For castings weighing not over 50 lb., test to destruction by pressure or blows, may be made by agreement, as an alternative to bend tests. One casting from each lot shall be so tested and shall show the material to be tough and ductile, and suitable for the purpose intended. Bend test specimens shall be $^1/_2 \times {}^3/_4$ in. and 12 in. long, poured in separate molds, from the same heat or melt of steel, and heat treated with the castings they represent, without being machined or ground. Test bars should be protected from oxidation by enclosing them in pipes or containers containing a small amount of carbonaceous material and suitably sealed to prevent contact of furnace gases with the test bars. When bend tests are specified, one test shall be made from each melt or heat of steel in each heat treatment charge.

**Note 55.**—Elongation in 8 in. = 1,600,000/tensile strength.*

**Note 56.**—Bars to be annealed by heating to 1450° F., holding at that temperature for 30 min., and cooling slowly in the furnace, or in still air. Tension test specimens to be full diameter of bar as rolled, and to be annealed. Two tension tests to be made for each melt. If melt is less than 30 tons, but one tension and one upset test are required. If thickness of material from one melt varies $^1/_4$ in. or more, one tension test is required from the thinnest and thickest sections. One upset test is required for each size of bar rolled from each melt. Test consists in hammering cold in a longitudinal direction, a specimen of length = ($1\ ^1/_4 \times$ diam. of bar), until its length is ($^3/_4 \times$ diam. of bar), without developing in the specimen seams or other injurious defects. Permissible variation in diameter or out of round of bars is as follows:

| Diam. of Bar, in. | Up to $^5/_{16}$ | $^5/_{16}$–$^7/_{16}$ | $^7/_{16}$–$^5/_8$ | $^5/_8$–$^7/_8$ | $^7/_8$–1 | 1–1 $^1/_8$ | 1 $^1/_8$–1 $^1/_4$ | 1 $^1/_4$–1 $^3/_8$ | 1 $^3/_8$–1 $^1/_2$ | 1 $^1/_2$–2 |
|---|---|---|---|---|---|---|---|---|---|---|
| Variation in Diam., in., ± | 0.007 | .007 | .008 | .009 | .010 | 0.012 | .014 | .016 | .018 | .022 |
| Out of round, in. | 0.010 | .011 | .012 | .014 | .015 | 0.016 | .018 | .022 | .026 | .030 |

**Note 57.**—Minimum elongation in 2 in. = 1,700,000/tensile strength, when 2 in. test specimen is used. (See Fig. 3, p. 7–54.)

**Note 58.**—Grade A castings are very soft and machinable in the as-cast, normal or full-annealed condition, due to low carbon. They are adapted to fusion fabrication with wrought parts.

**Note 59.**—Grade B castings, normalized, are intended for miscellaneous industrial applications. Grade B-1, full-annealed and Grade B-2, full-annealed, are used for castings whose member thickness or intended use render a full anneal desirable to develop the required physical properties. All three grades are easily machined with ordinary machine tools.

**Note 60.**—Grade H normalized and Grade H-1, full-annealed, are used for purposes calling for higher strength and lower ductility than are typical of other grades. They can be machined without difficulty by ordinary machine tools.

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* For structural nickel-steel plates, shapes, and bars, over $^3/_4$ in. thickness or diameter, 0.25% is to be deducted from the percentage specified by the formula for each increase of $^1/_{16}$ in. of thickness or diameter above $^3/_4$ in., to a minimum of 14%. For material under $^5/_{16}$ in. thickness or diameter, deduct 1.25% for each decrease in thickness or diameter of $^1/_{32}$ in. below $^5/_{16}$ in.

**Note 61.**—Tension and bend test specimens are taken from test bars attached to separately-cast blocks, and conform to the dimensions of a 2-in. test specimen. See Fig. 3, p. 7–54. Bend test specimens are to be machined to 1 × $^1/_2$ in. cross-section with the corners of the machined edges rounded to a radius of not over $^1/_{16}$ in.

**Note 62.**—Bend test required only when specified; it consists of bending specimen through 90 deg. around a 1-in. pin.

**Note 63.**—Bend test not required.

**Note 64.**—Bend test required only when specified; it consists of bending specimen through 120 deg. around a 1-in. pin.

**Note 65.**—Since corrosion resistance and heat resistance are of prime importance, tension tests and bend tests shall be made only when specified, in which case one tension test and one bend test shall be made from each melt. An average of three Brinell readings shall be taken as the Brinell hardness number.

**Note 66.**—Heat treatment consists of normalizing at not less than 1875° F., drawing at 1100–1350° F., and cooling in air.

**Note 67.**—Heat treatment consists in annealing at 1450–1550° F., cooling in furnace to 1100° F., and removing to cool in air. The steel should be cooled rapidly between 1050–900° F. to avoid inter-crystalline precipitations.

**Note 68.**—Heat treatment consists of annealing at 1450–1550° F., cooling in furnace to 1100° F., and cooling in air from 1100° F. Heavy castings may be cooled in water from 1100° F. to obtain maximum impact toughness. Castings of small dimensions may be quenched from 1650° F. to obtain best machinability.

**Note 69.**—Heat treatment consists in heating to a sufficient temperature for a long enough time to insure solution of carbides, and cooling at a rate to insure their retention in solution. If relief of quenching strains is desired, castings may be drawn at a temperature of 800° F., maximum, for not over 2 hours.

**Note 70.**—Heat treatment consists of heating to 1875° F., minimum, and quenching in air or water. Intricate castings may be drawn for strain relief at 800° F. maximum.

**Note 71.**—Heat treatment not required.

**Note 72.**—Heat treatment consists in heating to 1900° F. and quenching in water or in air. Oil may be used if water quenching is too drastic. Intricate castings may be drawn for strain relief at 800° F. maximum.

**Note 73.**—Values given for physical properties apply to heat-resistant alloy only when carbon does not exceed 0.30%. With higher carbon, ductility is lowered.

**Note 74.**—Tension test specimens shall conform to the dimensions of Fig. 3 (see p. 7–54.) Test coupons shall be cast attached near the end of each locomotive frame, to each locomotive cylinder, to each wheel center and to each other casting weighing over 100 lb., except as otherwise provided. Test castings shall remain attached to the castings throughout annealing and until the castings are presented for inspection. If the design is such that the test coupon cannot satisfactorily be attached to the casting, it shall be attached to a runner connected with the casting.

One tension test shall be made from a location near each end of every locomotive frame and one tension test may be made from each wheel center and each locomotive cylinder casting, and at least one tension test shall be made from one wheel center and one locomotive cylinder from each melt. One tension test shall be made for each melt represented by the castings covered by the specifications, except locomotive frames, wheel centers, and locomotive cylinders, as provided for above. For castings weighing not more than 150 lb. each, a test to destruction may be substituted for the tension test.

**Note 75.**—Castings shall not vary more than 5% above nor more than 3% below the weight agreed on as normal for the casting.

**CARBON STEEL AND ALLOY STEEL FORGINGS** (A.S.T.M. Designation A18–30).—Specifications cover the following classes of forgings: Class A which may be welded or case hardened. Class B, mild steel forgings for structural purposes, minor ship fittings, etc. Class C, mild steel forgings for structural purposes, ships, etc. Classes D, E, F, G, H, and I, various machinery forgings depending on design and stresses and services to be imposed. Classes K, L, and M, various machinery forgings, depending on design, stresses and services to be imposed, and on character of machining operations to be done. Steel shall be made either by the open-hearth or electric furnace, and shall conform to the chemical compositions of Table 12. The percentages of the elements specified in Table 12 shall be determined by an analysis of each melt, made from a test ingot taken during the pouring of the melt. Heat treatment, if required, shall consist of either annealing, or quenching and tempering, as specified. Annealing procedure consists in allowing the work, immediately after forging, to cool below the critical range under conditions that will prevent injuries by too rapid cooling, followed by uniform reheating to the proper temperature to refine the grain, after which the work is allowed to cool uniformly.

Quenching and tempering procedure consists in cooling and reheating the work as in annealing practice, and then quenching in some medium under substantially uniform conditions for each quenching charge. Finally, the work shall be uniformly reheated to the proper temperature for tempering or drawing back and allowed to cool uniformly.

Tensile properties for the various classes are given in Table 13. The classifications by size of the forging shall be determined by the specified diameter or thickness which

**Table 12.—Required Chemical Composition for Steel Forgings**
A.S.T.M. Designation A18–30

| Element | Class | | | | |
|---|---|---|---|---|---|
| | A | B,C,D,E,F,G | H, I | K, L | M |
| Manganese, percent.... | 0.30–0.60 | 0.40–0.80 | 0.40–0.80 | ......... | ......... |
| Phosphorus, percent | | | | | |
| Acid.............. | 0.05 max. | 0.05 max. | 0.04 max. | 0.05 max. | 0.04 max. |
| Basic.............. | 0.05 max. | 0.05 max. | 0.04 max. | 0.04 max. | 0.04 max. |
| Sulphur, percent....... | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. |
| Nickel, percent......... | ......... | ......... | 3.00 min. | ......... | ......... |

**Table 13.—Tensile Requirements for Steel Forgings**
A.S.T.M. Designation A18–30

| Class | Condition † | Outside Diam. or Overall Thickness, in. | Max. Wall Thickness, in. | Tensile Strength, 1000 lb. per sq. in., min. | Elastic Limit, 1000 lb. per sq. in., min. | Yield Point, 1000 lb. per sq. in., min. | Elongation in 2 in. | | Reduction of Area | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | percent, min. | but not under | percent, min. | but not under |
| A* | U | All sizes | ...... | 47–60 | ......... | 0.5 T.S. | 1,500,000/T.S. | ...... | 2,500,000/T.S. | ...... |
| B* | U | Not over 12 | ...... | 60 | ......... | 0.5 T.S. | 1,550,000/T.S. | 22 | 2,400,000/T.S. | 35 |
| | | 12–20 | ...... | 60 | ......... | 0.5 T.S. | 1,480,000/T.S. | 21 | 2,220,000/T.S. | 32 |
| C* | An | Not over 12 | ...... | 60 | ......... | 0.5 T.S. | 1,700,000/T.S. | 25 | 2,700,000/T.S. | 38 |
| | | 12–20 | ...... | 60 | ......... | 0.5 T.S. | 1,600,000/T.S. | 24 | 2,520,000/T.S. | 36 |
| D* | U | Not over 8 | ...... | 75 | ......... | 0.5 T.S. | 1,600,000/T.S. | 18 | 2,200,000/T.S. | 24 |
| | | 8–12 | ...... | 75 | ......... | 0.5 T.S. | 1,500,000/T.S. | 17 | 2,000,000/T.S. | 22 |
| | | 12–20 | ...... | 75 | ......... | 0.5 T.S. | 1,400,000/T.S. | 16 | 1,800,000/T.S. | 20 |
| E* | An | Not over 8 | ...... | 75 | ......... | 0.5 T.S. | 1,800,000/T.S. | 20 | 2,800,000/T.S. | 33 |
| | | 8–12 | ...... | 75 | ......... | 0.5 T.S. | 1,725,000/T.S. | 19 | 2,640,000/T.S. | 31 |
| | | 12–20 | ...... | 75 | ......... | 0.5 T.S. | 1,650,000/T.S. | 18 | 2,400,000/T.S. | 29 |
| F* | An | Not over 8 | ...... | 80 | ......... | 0.5 T.S. | 1,800,000/T.S. | 20 | 2,800,000/T.S. | 32 |
| | | 8–12 | ...... | 80 | ......... | 0.5 T.S. | 1,725,000/T.S. | 19 | 2,640,000/T.S. | 30 |
| | | 12–20 | ...... | 80 | ......... | 0.5 T.S. | 1,650,000/T.S. | 18 | 2,400,000/T.S. | 28 |
| G‡ | QT | Up to 4 | 2 | 90 | 55 | ......... | 2,100,000/T.S. | 20.5 | 4,000,000/T.S. | 39 |
| | | 4–7 | 3 1/2 | 85 | 50 | ......... | 2,000,000/T.S. | 20.5 | 3,800,000/T.S. | 39 |
| | | 7–10 | 5 | 85 | 50 | ......... | 1,900,000/T.S. | 19.5 | 3,600,000/T.S. | 37 |
| | | Not over 20 | 5–8 | 82.5 | 48 | ......... | 1,800,000/T.S. | 19 | 3,400,000/T.S. | 36 |
| H* Ni-steel | An | Not over 12 | ...... | 80 | 50 | ......... | 2,000,000/T.S. | 22 | 3,600,000/T.S. | 40 |
| | | 12–20 | ...... | 80 | 50 | ......... | 1,900,000/T.S. | 21 | 3,400,000/T.S. | 38 |
| I‡ Ni-steel | QT | Up to 4 | 2 | 100 | 70 | ......... | 2,200,000/T.S. | 20 | 4,500,000/T.S. | 41 |
| | | 4–7 | 3 1/2 | 100 | 65 | ......... | 2,100,000/T.S. | 20 | 4,300,000/T.S. | 41 |
| | | 7–10 | 5 | 90 | 60 | ......... | 2,000,000/T.S. | 20 | 4,100,000/T.S. | 41 |
| | | Not over 20 | 5–8 | 85 | 55 | ......... | 1,900,000/T.S. | 20 | 3,900,000/T.S. | 41 |
| K‡ Alloy Steel | QT | Up to 2 | 1 | 90–115 | 70 | ......... | 20 | ...... | 50 | ...... |
| | | 2–4 | 2 | 90–110 | 65 | ......... | 20 | ...... | 50 | ...... |
| | | 4–7 | 3 1/2 | 90–110 | 65 | ......... | 20 | ...... | 50 | ...... |
| | | 7–10 | 5 | 90–110 | 65 | ......... | 20 | ...... | 50 | ...... |
| | | Not over 20 | 5–8 | 85–105 | 60 | ......... | 20 | ...... | 50 | ...... |
| L‡ Alloy Steel | QT | Up to 2 | 1 | 105–125 | 80 | ......... | 20 | ...... | 50 | ...... |
| | | 2–4 | 2 | 100–120 | 75 | ......... | 20 | ...... | 50 | ...... |
| | | 4–7 | 3 1/2 | 100–120 | 75 | ......... | 20 | ...... | 50 | ...... |
| | | 7–10 | 5 | 100–120 | 75 | ......... | 18 | ...... | 45 | ...... |
| | | Not over 20 | 5–8 | 95–115 | 70 | ......... | 18 | ...... | 45 | ...... |
| M‡ Alloy Steel | QT | Up to 2 | 1 | 125 | 105 | ......... | 16 | ...... | 50 | ...... |
| | | 2–4 | 2 | 115 | 95 | ......... | 16 | ...... | 45 | ...... |
| | | 4–7 | 3 1/2 | 110 | 85 | ......... | 16 | ...... | 45 | ...... |
| | | 7–10 | 5 | 100 | 75 | ......... | 18 | ...... | 45 | ...... |
| | | Not over 20 | 5–8 | 100 | 70 | ......... | 18 | ...... | 45 | ...... |

* Specification covers forgings of a maximum outside diameter or overall thickness of ⋜ 20 in.
† U = untreated; A = annealed; QT = quenched and tempered.
‡ Specification covers forgings having a maximum outside diameter or thickness of not over 10 in. when solid, or not over 20 in. when bored.

governs the size of the prolongation from which the test specimen is taken. Yield point shall be determined by drop of the beam or halt in the gage of the testing machine. The elastic limit shall be determined by an extensometer, reading to 0.0002 in., the load at which the rate of elongation shows a sudden increase being taken as the yield point.

Tests shall be made as follows: Untreated forgings, one tension test from each melt; annealed forgings, one tension test from each annealing charge, but if more than one melt is represented in each charge, one tension test shall be made from each melt; quenched and tempered forgings, one tension test from each tempering charge, but if more than one quenching charge is represented in a tempering charge, or more than one melt is represented in a quenching charge, one tension test shall be made from each quenching charge and melt, respectively; if more than one class of forgings by size is represented in any lot, one test from a forging of each class by size shall be made. Tests shall be made only after final treatment of the forgings.

Tension test specimens shall be taken from a full size prolongation of any forging. On forgings with large ends or collars, the prolongation may be of the same cross-section as that of the forging back of the end or collar. Specimens may be taken from the forging itself with a hollow drill, if approved by the purchaser. The axis of the specimen shall be located at any point midway between the center and surface of solid forgings, and at any point midway between the inner and outer surfaces of bored forgings, and shall be parallel to the axis of the forging in the direction in which the metal is most drawn out. Specimens shall conform to the dimensions of Fig. 3 (see p. 7–54). The ends shall be of a form to fit the holders of the testing machine in such a way that the load shall be axial.

**COMMERCIAL COLD-FINISHED BAR STEEL AND COLD-FINISHED SHAFTING** (A.S.T.M. Designation A108–36).—Specifications cover rounds, square and hexagons of all sizes, and flats up to 6 in. wide, and shafting rounds up to 8 in. inclusive. Steel shall be made by the open-hearth or acid Bessemer processes. Open-hearth steel is understood to mean steel made in the open-hearth or electric furnaces. Bar steel shall conform to the chemical compositions specified in Table 14. Check analyses may be made by the purchaser from finished bars representing each melt of open-hearth and each melt or lot of bars of Bessemer steel. Maximum permissible variation from the specified analyses, as shown by the check analysis, is given in Table 14. The permissible variation from specified diameters of cold finished round bars or shaftings and distances between parallel faces of hexagon squares or flats is given in Table 15.

**COMMERCIAL QUALITY HOT-ROLLED STEEL BARS** (A.S.T.M. Designation A107–36).—Specifications cover round, squares, and hexagons of all sizes, and flats up to 6 in. wide, of steel made by the open-hearth or acid Bessemer process. Open-hearth steel

**Table 14.—Composition of Cold-finished Bar Steels and Cold-finished Shafting**
A.S.T.M. Designation A108-36

| Grade No. | Specified Carbon, percent | Check Analysis Permissible Variations | | Specified Manganese, percent | Check Analysis Permissible Variations | | Specified Phosphorus, percent | Check Analysis Permissible Variations | | Specified Sulphur, percent | Check Analysis Permissible Variations | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Over | Under | | Over | Under | | Over | Under | | Over | Under |
| Open-hearth Grades | | | | | | | | | | | | |
| 1 | 0.05–0.15 | 0.03 | 0.02 | 0.30–0.60 | 0.05 | 0.05 | 0.045* | 0.005 | ...... | 0.055* | 0.005 | ...... |
| 2 | .10– .20 | .03 | .02 | .30– .60 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 3 | .15– .25 | .03 | .02 | .30– .60 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 4 | .15– .25 | .03 | .02 | .70–1.00 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 5 | .20– .30 | .04 | .03 | .30–0.60 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 6 | .25– .35 | .04 | .03 | .60– .90 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 7 | .30– .40 | .04 | .03 | .60– .90 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 8 | .35– .45 | .04 | .03 | .60– .90 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 9 | .40– .50 | .04 | .03 | .60– .90 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| 10 | .45– .55 | .04 | .03 | .60– .90 | .05 | .05 | .045* | .005 | ...... | .055* | .005 | ...... |
| Bessemer Grades | | | | | | | | | | | | |
| 11 | 0.15* | 0.03 | 0.02 | 0.60* | 0.05 | 0.05 | 0.11* | 0.005 | ...... | ....... | 0.005 | ...... |
| Bessemer Free-cutting Steels | | | | | | | | | | | | |
| 12 | 0.08–0.16 | 0.03 | 0.02 | 0.60–0.90 | 0.05 | 0.05 | 0.09–0.13 | ....... | ...... | 0.10–0.20 | ...... | 0.01 |
| 13 | .08– .16 | .03 | .02 | .60– .90 | .05 | .05 | .09– .13 | ....... | ...... | .20– .30 | ...... | .01 |
| Open-hearth Free-cutting Steels | | | | | | | | | | | | |
| 14 | 0.15–0.25 | 0.03 | 0.02 | 0.60–0.90 | 0.05 | 0.05 | 0.06* | 0.005 | ...... | 0.075–0.15 | ...... | 0.01 |
| 15 | .10– .20 | .03 | .02 | 1.00–1.30 | .10 | .05 | .045* | .005 | ...... | .075– .15 | ...... | .01 |
| 16 | .10– .20 | .03 | .02 | 1.30–1.60 | .10 | .05 | .045* | .005 | ...... | .075– .15 | ...... | .01 |
| 17 | .30– .40 | .04 | .03 | 1.35–1.65 | .10 | .05 | .045* | .005 | ...... | .075– .15 | ...... | .01 |

* Maximum.

### Table 15.—Permissible Variation in Sizes of Cold-finished Steel Bars and Shafting

A.S.T.M. Designation A108-36

| Specified Side of Square, Short Diam. of Hexagon, or Thickness of Flat, in. | | Permissible Variations, in., Under Specified Size. (Permissible Variation Over = 0.) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Squares | | Hexagons | | Flats, Variation in Width or Thickness. | | | | | |
| | | | | | | Widths up to 1 1/2 in. | | Widths 1 1/2 in. to 4 in. incl. | | Widths over 4 in. to 6 in. incl. | |
| | | Carbon Content, percent | | | | | | | | | |
| From | To and Including | 0.30 & less | 0.31–.50 | 0.30 & less | 0.31–.50 | 0.30 & less | 0.31–.50 | 0.30 & less | 0.31–.50 | 0.30 & less | 0.31–.50 |
| ........ | 0.3 | 0.003 | 0.004 | 0.002 | 0.003 | 0.003 | 0.004 | 0.005 | 0.006 | 0.008 | 0.010 |
| 0.3 | 1. | .004 | .005 | .003 | .004 | .004 | .005 | .005 | .006 | .009 | .012 |
| 1 | 2 1/2 | .005 | .006 | .004 | .005 | .005 | .006 | .006 | .008 | .010 | .012 |
| 2 1/2 | 4. | .006 | .008 | .005 | .006 | ........ | ........ | ........ | ........ | ........ | ........ |

Round Bars, Shafting

| | Carbon Content 0.30% and less | | | | | Carbon Content 0.31–0.50% | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Specified Diam., in...... | Up to 1 in. incl. | 1–2 in. incl. | 2–4 in. incl. | 4–6 in. incl. | 6–8 in. incl. | Up to 1 in. incl. | 1–2 in. incl. | 2–4 in. incl. | 4–6 in. incl. | 6–8 in. incl. |
| Permissible Variation, under.. | 0.002 | 0.003 | 0.004 | 0.005 | 0.006 | 0.003 | 0.004 | 0.005 | 0.006 | 0.008 |

### Table 16.—Chemical Composition of Hot-rolled Bar Steel

A.S.T.M. Designation A107-36

| Grade No. | Specified Carbon, percent | Check Analysis Permissible Variations | | Specified Manganese, percent | Check Analysis Permissible Variations | | Specified Phosphorus, percent | Check Analysis Permissible Variations | | Specified Sulphur, percent | Check Analysis Permissible Variations | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Over | Under | | Over | Under | | Over | Under | | Over | Under |
| Open-hearth Grades | | | | | | | | | | | | |
| 1 | 0.05–0.10 | 0.03 | 0.02 | 0.30–0.60 | 0.03 | 0.02 | 0.04* | 0.005 | ...... | 0.05* | 0.005 | ...... |
| 2 | .10– .15 | .03 | .02 | .30– .60 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 3 | .15– .20 | .03 | .02 | .30– .60 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 4 | .15– .20 | .03 | .02 | .70–1.00 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 5 | .20– .25 | .03 | .02 | .30–0.60 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 6 | .25– .35 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 7 | .30– .40 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 8 | .35– .45 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 9 | .40– .50 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 10 | .45– .55 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 11 | .50– .60 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 12 | .60– .75 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 13 | .75– .90 | .03 | .02 | .60– .90 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| 14 | .90–1.05 | .04 | .03 | .25– .50 | .03 | .02 | .04* | .005 | ...... | .05* | .005 | ...... |
| Bessemer Grades | | | | | | | | | | | | |
| 15 | 0.12* | 0.03 | 0.02 | 0.60* | 0.03 | 0.02 | 0.11* | 0.005 | ...... | 0.08* | 0.005 | ...... |
| 16 | .15* | .03 | .02 | .70* | .03 | .02 | .11* | .005 | ...... | ....... | ...... | ...... |
| 17 | 0.15–0.25 | .03 | .02 | .90* | .03 | .02 | .11* | .005 | ...... | ....... | ...... | ...... |
| 18 | .25– .35 | .03 | .02 | .90* | .03 | .02 | .11* | .005 | ...... | ....... | ...... | ...... |
| 19 | .35– .50 | .03 | .02 | .90* | .03 | .02 | .11* | .005 | ...... | ....... | ...... | ...... |
| Bessemer Free-cutting Steels | | | | | | | | | | | | |
| 20 | 0.08–0.16 | 0.03 | 0.02 | 0.60–0.90 | 0.03 | 0.02 | 0.09–0.13 | ........ | ...... | 0.10–0.20 | ...... | 0.01 |
| 21 | .08– .16 | .03 | .02 | .60– .90 | .03 | .02 | .09– .13 | ........ | ...... | .20– .30 | ...... | .01 |
| Open-hearth Free-cutting Steels | | | | | | | | | | | | |
| 22 | 0.15–0.25 | 0.03 | 0.02 | 0.60–0.90 | 0.03 | 0.02 | 0.06* | 0.005 | ...... | 0.075–0.15 | ...... | 0.01 |
| 23 | .10– .20 | .03 | .02 | 1.00–1.30 | .05 | .03 | .04* | .005 | ...... | .075– .15 | ...... | .01 |
| 24 | .10– .20 | .03 | .02 | 1.30–1.60 | .05 | .03 | .04* | .005 | ...... | .075– .15 | ...... | .01 |
| 25 | .30– .40 | .03 | .02 | 1.35–1.65 | .05 | .03 | .04* | .005 | ...... | .075– .15 | ...... | .01 |
| Bessemer Resulphurized Nut Stock | | | | | | | | | | | | |
| 26 | 0.08–0.16 | 0.03 | 0.02 | 0.50–1.00 | 0.03 | 0.02 | 0.11* | 0.005 | ...... | 0.075–0.15 | ...... | 0.01 |
| Open-hearth Resulphurized Nut Stock | | | | | | | | | | | | |
| 27 | 0.15–0.25 | 0.03 | 0.02 | 0.50–1.00 | 0.03 | 0.02 | 0.06* | 0.005 | ...... | 0.075–0.15 | ...... | 0.01 |

* Maximum.

is understood to be steel made by either the open-hearth or electric furnace. Steel shall conform to the chemical compositions shown in Table 16.

For Bessemer steel an analysis for each melt shall be made for carbon and manganese, and for Bessemer Grade 15 (welding steel), free cutting steels and resulphurized nut stock, sulphur shall be determined for each melt. Phosphorus determinations shall represent

**Table 17.—Permissible Variation in Size of Hot-rolled Bars**

A.S.T.M. Designation A106-36

| Rounds and Squares | | | | |
|---|---|---|---|---|
| Size, in. | | Permissible Variation, in. | | |
| From | To and Including | Under | Over | Out of Round or Square |
| ........ | 5/16 | 0.007 | 0.007 | 0.010 |
| 5/16 | 7/16 | .007 | .007 | .011 |
| 7/16 | 5/8 | .008 | .008 | .012 |
| 5/8 | 7/8 | .009 | .009 | .014 |
| 7/8 | 1 | .010 | .010 | .015 |
| 1 | 1 1/8 | .012 | .012 | .016 |
| 1 1/8 | 1 1/4 | .014 | .014 | .018 |
| 1 1/4 | 1 3/8 | .016 | .016 | .022 |
| 1 3/8 | 1 1/2 | 018 | .018 | .026 |
| 1 1/2 | 2 | .022 | .022 | .030 |
| 2 | 2 1/2 | 0 | 1/16 | 1/32 |
| 2 1/2 | 3 1/2 | 0 | 5/64 | 3/64 |
| 3 1/2 | 4 1/2 | 0 | 3/32 | 1/16 |
| 4 1/2 | 5 1/2 | 0 | 7/64 | 5/64 |
| 5 1/2 | 8 | 0 | 1/8 | 3/32 |

| Hexagons | | | | |
|---|---|---|---|---|
| Short Diameter, in. | | Permissible Variation, in. | | |
| From | To and Including | Under | Over | Difference, 3 Measurements |
| .......... | 1/2 | 0.007 | 0.007 | 0.011 |
| 1/2 | 1 | .010 | .010 | .015 |
| 1 | 1 1/2 | .013 | .021 | .025 |
| 1 1/2 | 2 | 1/64 | 1/32 | 1/32 |
| 2 | 2 1/2 | 1/64 | 3/64 | 3/64 |
| 2 1/2 | 3 1/2 | 1/64 | 1/16 | 1/16 |

| Flats, Square- and Round-edge | | | | | | |
|---|---|---|---|---|---|---|
| Width, in. | | | | Variation in Thickness, in. | | |
| From | To and Including | Variation | | Flats 1/4 in. to 1/2 in. Thick | Flats Over 1/2 in. to 1 in. Thick | Flats Over 1 in. to 2 in. Thick |
| | | Under | Over | | | |
| ...... | 1 | 1/64 | 1/64 | ±0.008 | ±0.010 | ....... |
| 1 | 2 | 1/32 | 1/32 | ± .012 | ± .015 | ± 1/32 |
| 2 | 4 | 1/32 | 1/16 | ± .015 | ± .020 | ± 1/32 |
| 4 | 6 | 1/16 | 3/32 | ± .015 | ± .020 | ± 1/32 |

the average of melts made during each 8-hr. turn. Open-hearth steel shall be analzyed for carbon, manganese, phosphorus and sulphur for each melt. Check analyses may be made by the purchaser from finished bars representing each melt of open-hearth steel and each melt or lot of bars of Bessemer steel. Maximum permissible variation from the specified analysis, as shown by this check analysis, is given in Table 16.

Variation from specified diameter or distance between parallel faces shall not exceed the limits given in Table 17.

**DROP TEST FOR CARBON STEEL RAILS** (A.S.T.M. designation A1–36).—Physical qualities to be determined by drop test, with a 2000-lb. tup, with a striking face of 5 in. radius, and 20,000-lb. anvil block supported on springs. Test pieces to be placed head up, on supports which are part of or firmly secured to anvil, and which have bearing surface of 5 in. radius. Supports to be spaced 3 ft. between centers for rails up to and including 106 lb. per yard, and 4 ft. between centers for rails over 106 lb. per yard. Height of free drop of tup to be as follows:

| | | | | | | |
|---|---|---|---|---|---|---|
| Weight of rail, lb. per yd..... | 50–60 | 61–80 | 81–90 | 91–100 | 101–120 | 121–140 |
| Height of drop, ft........... | 16 | 17 | 18 | 19 | 20 | 22 |

Three specimens from each melt must endure, without fracture, one blow each of the tup. One specimen shall be given a sufficient number of blows to determine the exhausted ductility, reported inch by inch, over the entire 6-in. gage length. Permanent set measured by middle ordinate in inches in length of 3 ft. shall be recorded after the first blow on all specimens. Brinell hardness shall be determined on one specimen, by means of a 19 mm. ball, with a pressure of 100,000 lb. Elongation, permanent set and hardness are determined for information only. The fracture of nicked and broken sections of No. 1 rails shall be free from seams, laminations, cavities, interposed foreign matter or a distinctly bright or fine-grained structure.

## Automotive Steels

The Society of Automotive Engineers has issued specifications (see S.A.E. Handbook) for steels that cover practically all requirements for ferrous metal parts in the construction of automotive equipment. These specifications have been widely adopted by industry in general, and it is common practice to specify steel by its S.A.E. number.

**S.A.E. Steel Numbering System** comprises, in general, a series of four-digit numbers, the first digit of which indicates the type of steel as follows:

| | |
|---|---|
| Carbon steels........................ | 1 |
| Nickel steels........................ | 2 |
| Nickel-chromium steels............. | 3 |
| Molybdenum steels.................. | 4 |
| Chromium steels.................... | 5 |
| Chromium-vanadium steels.......... | 6 |
| Tungsten steels..................... | 7 |
| Silico-manganese steels.............. | 9 |

For simple alloy steels, the second digit, generally, indicates the approximate percentage of the predominant alloying element. The last two digits, usually, indicate the average

(*Continued on p.* 2–41)

**Table 18.—Applications of S.A.E. Steels**

| Quality or Use | S.A.E. No. |
|---|---|
| Agricultural equipment | 1055 |
| " " | 1070 |
| " " | 1075 |
| " " | 1080 |
| " " | 1085 |
| " " | 1090 |
| " " | 1095 |
| " " | 2330 |
| " " | 2335 |
| Aircraft forgings | 4140 |
| " sheet and strip | X4130 |
| " " " " | 4130 |
| Anti-fatigue | 3335 |
| " | 3340 |
| " | 4340 |
| " | 4345 |
| " | 4615 |
| " | 4620 |
| " | 4640 |
| " | 6145 |
| " | 6150 |
| Automatic machine products | 1112 |
| Machinability | X1112 |
| Automobile trim | 51510 |
| Automotive parts | 2330 |
| " " | 2335 |
| Aviation parts | 2330 |
| " " | 2335 |
| Axle, axles | 4130 |
| " " | X4130 |
| " front | 1040 |
| " " tubing | 4140 |
| " rear | 1040 |
| " " | 4140 |
| " shafts | 1045 |
| " " | 2340 |
| " " | 2345 |
| " " | 3135 |
| " " | 3140 |
| " " | X3140 |
| " " | 3245 |
| " " | 3250 |
| " " | 4340 |
| " " | 4345 |
| " " | 6135 |
| " " | 6140 |
| " tubular | 1040 |
| Ball bearing races | 6195 |
| " " " | 52100 |
| Ball bearings | 6195 |
| " " | 52100 |
| Balls, steel | 1090 |
| " " | 1095 |
| Bars, bumper | 1085 |
| " " | 1090 |
| " " | 1095 |
| Beams, plow | 1070 |
| Bearings, ball | 52100 |
| " " corrosion-resistant | 51335 |
| Bearings, roller | 4815 |
| " " | 4820 |
| Bending | 1115 |
| " | 1120 |
| Binder parts | 1090 |
| " " | 1095 |
| Bolts, anchor | 1040 |
| " connecting-rod | 3125 |
| " " | 3130 |
| Bolts, heat-treated | 2330 |
| " " | 2335 |
| " high strength | 4815 |
| " " " | 4820 |
| " hub | X1025 |
| " rim | X1025 |
| " steering arm | 3125 |
| " " " | 3130 |
| Brackets | 1030 |
| Bright surfaces, exposed to oxidation | 51510 |
| Bumper bars | 1085 |
| " " | 1090 |
| " " | 1095 |
| Bundle carrier teeth for binders | 1090 |
| do. | 1095 |
| Bushings, corrosion-resistant | 51335 |
| Cam shafts | 1020 |
| Cams, free wheeling | 4615 |
| " " " | 4620 |
| Carburized parts | 1010 |
| " " | 1020 |
| " " | X1020 |
| " " | 1112 |
| " " | X1112 |
| " " | 2015 |
| " " | 2315 |
| " " | 2512 |
| " " | 2515 |
| " " | 3115 |
| " " | 3120 |
| " " | 3215 |
| " " | 3220 |
| " " | 3312 |
| " " | 3415 |
| " " | 4615 |
| " " | 4620 |
| " " | 4815 |
| " " | 4820 |
| " " | 6115 |
| " " | 6120 |
| Case-hardened parts | 1025 |
| " " | 1030 |
| " " | 1115 |
| " " | 1120 |
| " " machinability | X1314 |
| do. | X1315 |
| Chain pins | 4815 |
| " " | 4820 |
| Clutch discs | 1060 |
| " " | 1070 |
| " " | 1085 |
| " fingers | 1020 |
| Coil springs, *see* Springs | |
| Cold working | 1030 |
| " " | 1035 |
| " " | 30905 |
| " " | 30915 |
| " " corrosion-resistant | 51510 |
| Connecting-rod | 1040 |
| " | X3140 |
| " | 6135 |
| " | 6140 |
| " bolts | 3125 |
| " " | 3130 |
| Core hardness | 1025 |
| Core hardness (*Cont.*): | 1030 |
| " " | 4815 |
| " " | 4820 |
| " toughness | X1314 |
| " " | X1315 |
| " " | 2512 |
| " " | 2515 |
| " " | 3215 |
| " " | 3220 |
| " " | 3312 |
| " " | 3415 |
| Corrosion resistance | 51210 |
| " " | 51335 |
| " " | X51410 |
| " " | 51510 |
| Coulter blades | 1075 |
| Countershafts | 3115 |
| " | 3120 |
| Crank-shafts | 1040 |
| " | 1045 |
| " | 3135 |
| " | 3140 |
| " Diesel | 4340 |
| Cultivator discs | 1075 |
| Cushion springs, *see* Springs | |
| Cutlery | 51335 |
| Cyanided parts | 1112 |
| " " | X1112 |
| " " | 5140 |
| Discs, clutch | 1060 |
| " " | 1070 |
| " " | 1085 |
| " cultivator | 1075 |
| " harrow | 1080 |
| " " | 1090 |
| " " | 1095 |
| " knotter | 1080 |
| " plow | 1080 |
| " seeder | 1090 |
| " " | 1095 |
| Diesel engine crank-shaft | 4340 |
| Drag links | 1020 |
| Drawing, deep | 30905 |
| " " | 30915 |
| Drive chain links | 3135 |
| " " " | 3140 |
| Dynamic stresses, resistance to | 3325 |
| do. | 3335 |
| do. | 3340 |
| do. | 3450 |
| Fan blades | 1020 |
| Fatigue, resistance to, *see* Resistance | |
| Forgings | 1020 |
| " | 1030 |
| " | 1035 |
| " | 1040 |
| " | 1045 |
| " | 3450 |
| " | 30905 |
| " | 30915 |
| " aircraft | 4140 |
| " anti-fatigue | 6145 |
| " " | 6150 |
| " corrosion-resistant | 51210 |
| do. | X51410 |

**Table 18.—Applications of S.A.E. Steels—*Continued***

| Quality or Use | S.A.E. No. | Quality or Use | S.A.E. No. | Quality or Use | S.A.E. No. |
|---|---|---|---|---|---|
| Forgings, corrosion-re- | | Harrow discs (*Cont.*).. | 1095 | Mower knives........ | 1085 |
| sistance (*Cont.*)..... | 51510 | Hay rake teeth........ | 1090 | " sections....... | 1070 |
| Forgings, heat-treated. | 3240 | " " " ........ | 1095 | Music wire.......... | 1085 |
| " " . | 5140 | Heat resistance........ | 30905 | Nuts................ | 2330 |
| " " , | | " " ........ | 30915 | " ................ | 2335 |
| " high strength.. | 6145 | Heat-treated forgings.. | 6145 | " ................ | 3125 |
| do. | 6150 | " " .. | 6150 | " ................ | 3130 |
| " heavy......... | 6145 | " parts..... | 3450 | " corrosion-resistant | 51210 |
| " " ......... | 6150 | " " | | " " | X51410 |
| " high physical | | " large-section.. | 2350 | " " | 51510 |
| properties...... | 3245 | Heat-treated springs... | X1065 | Oil hardened parts..... | 2115 |
| do. | 3250 | Heavy duty........... | 3245 | " " " ..... | 2320 |
| do. | 3450 | " " .......... | 3250 | " " " ..... | 3245 |
| Forgings, resistance to | | " sections........ | 3145 | " " " ..... | 3250 |
| dynamic stress...... | 3450 | " " ........ | 3150 | " " " ..... | 3325 |
| Forgings, severe duty.. | 6145 | " " ........ | 4150 | " " " ..... | 5120 |
| " " " | 6150 | " " ........ | 4345 | " quenched parts..... | 6125 |
| Forming............. | 1115 | High fatigue resistance, | | " " " ..... | 6130 |
| " ............. | 1120 | *see* Resistance | | " " " ..... | T1335 |
| Free cutting.......... | X1330 | High physical properties | X1335 | " " " ..... | T1340 |
| " " ......... | X1335 | " " " | X1340 | " tempered wire...... | X1065 |
| " " ......... | X1340 | " " " | 3230 | Ornamental purposes... | 51710 |
| Free wheeling cams.... | 4615 | " " " | 3240 | Physical properties, | |
| " " " .... | 4620 | " " " | 3245 | good............... | 3125 |
| " " shells... | 4815 | " " " | 3250 | do. | 3130 |
| " " " .... | 4820 | " " " | 3325 | do. | 3135 |
| Front axle tubing...... | 4140 | " " " | 3335 | do. | 3140 |
| Gearing, transmission.. | 3115 | " " " | 3340 | Physical properties, | |
| " " .. | 3120 | " " " | 3450 | high, *see* High Physi- | |
| " " .. | 3312 | " " " | 4640 | cal Properties | |
| " " .. | 6115 | High strength......... | 3415 | Pins................. | 1090 |
| " " .. | 6120 | " " ......... | 4340 | " ................. | 1095 |
| Gears............... | 2315 | " " ......... | 4345 | " chain ........... | 4815 |
| " ............... | 2340 | " " ......... | 4615 | " " .......... | 4820 |
| " ............... | 2345 | " " ......... | 4620 | " lock............. | 1085 |
| " ............... | 3245 | " " ......... | 6145 | " steering knuckle... | 4815 |
| " ............... | 3250 | " " ......... | 6150 | " " " ... | 4820 |
| " ............... | 5140 | Keys................ | 1030 | Pinions.............. | 3115 |
| " ............... | 5150 | " ................ | 1090 | " .............. | 3120 |
| " ............... | 6145 | " ................ | 1095 | Plow beams........... | 1070 |
| " ............... | 6150 | " ................ | 2330 | " discs........... | 1080 |
| " anti-fatigue..... | 4345 | " ................ | 2335 | Plows................ | 1075 |
| " " ..... | 4615 | " ................ | 3125 | Plowshares........... | 1080 |
| " " ..... | 4620 | " ................ | 3130 | Propeller shafts........ | 2340 |
| " " ..... | 4640 | Knives, mower........ | 1085 | " " ........ | 2345 |
| " carburized ...... | 4615 | Knotter discs......... | 1080 | " " ....... | 4140 |
| " " ...... | 4620 | Leaf springs, *see* Springs | | " " ....... | 6135 |
| " " ...... | 4815 | Levers.............. | 2330 | " " ....... | 6140 |
| " " ...... | 4820 | " .............. | 2335 | Races, ball and roller... | 52100 |
| " cyanided ....... | 6135 | " brake.......... | 1030 | " " " " ... | 6195 |
| " " ....... | 6140 | " " ......... | 1040 | Rear axle gearing...... | 3312 |
| " high strength ... | 4345 | " shift.......... | 1030 | " " " ...... | 3415 |
| " " " ... | 4620 | Links, drag.......... | 1020 | Resistance to corrosion. | 51210 |
| " large.......... | 4345 | " drive chain..... | 3135 | " " " | 51335 |
| " oil hardened.... | 2350 | " " " ..... | 3140 | " " " | X51410 |
| " " " .... | 3145 | Lock pins........... | 1085 | " " " | 51510 |
| " " " .... | 3150 | " washers......... | X1055 | " " dynamic | |
| " " " .... | 4340 | " " ......... | 1060 | stresses........... | 3325 |
| " rear axle....... | 3312 | Machined parts...... | 6145 | do. | 3335 |
| " " " ....... | 3415 | " " high | | do. | 3340 |
| " ring.......... | 1045 | strength and anti- | | do. | 3450 |
| " " .......... | 3115 | fatigue............. | 6150 | Resistance to fatigue... | 4340 |
| " " .......... | 3120 | Machining.......... | 1025 | " " " ... | 4345 |
| " transmission..... | 3312 | " ............ | 1030 | " " " ... | 4615 |
| Hard-drawn wires..... | X1055 | " ............ | 1040 | " " " ... | 4620 |
| " " ..... | 1065 | " ............ | 1112 | " " " ... | 4640 |
| Hardness, surface...... | 4815 | " ............ | X1112 | " " " ... | 6145 |
| " " ...... | 4820 | " ............ | X1314 | " " " ... | 6150 |
| Harrow discs.......... | 1080 | " ............ | X1315 | " " oxidation. | 51510 |
| " " .......... | 1090 | Molds.............. | 1080 | " " wear..... | 4815 |

(*Table continued on following page*)

Table 18.—Applications of S.A.E. Steels—*Continued*

| Quality or Use | S.A.E. No. | Quality or Use | S.A.E. No. | Quality or Use | S.A.E. No. |
|---|---|---|---|---|---|
| Resistance to wear | | Shafts, rear axle | 4345 | Steering arm bolts | 3125 |
| (*Cont.*) | 4820 | " spline | 2340 | " " " | 3130 |
| Ring gears | 1045 | " " | 2345 | " knuckles | X3140 |
| " " | 3115 | " " | 3115 | " " | 4130 |
| " " | 3120 | " " | 3120 | " " | X4130 |
| Rings, snap | X1055 | " " | 3135 | " " arms | 4130 |
| " " | 1060 | " " | 3140 | " " " | X4130 |
| Riveting | 1115 | " transmission | 4140 | " " pins | 4815 |
| " | 1120 | Sheet | 1020 | " " " | 4820 |
| " | 30905 | " aircraft | 4130 | Strength and toughness | 2330 |
| " | 30915 | " " | X4130 | " " " | 2335 |
| Rivets, corrosion-resis- | | " corrosion-resis- | | " " " | 2340 |
| tant | 51210 | tant | 51710 | " " " | 2345 |
| do. | X51410 | Shells, free wheeling | 4815 | " " " | 3125 |
| do. | 51510 | " " " | 4820 | " " " | 3130 |
| Rod, corrosion-resistant | 51210 | Shift forks | 1030 | " high, *see* | |
| " " | X51410 | Shovels | 1080 | High strength | |
| " " | 51510 | Snap rings | X1055 | Stress, *see* Resistance to | |
| Roller bearing races | 6195 | " " | 1060 | dynamic stress | |
| " " " | 52100 | Spline shafts, *see* Shafts, | | Strip | 1020 |
| Roller bearings | 4815 | spline | | " aircraft | 4130 |
| " " | 4820 | Spring-clips | 1040 | " " | X4130 |
| Rollers | 6195 | " | 1060 | " corrosion-resistant | 51210 |
| Screening, corrosion-re- | | Springs | 1080 | " " | X51410 |
| sistant | 51210 | " | 5150 | " " | 51510 |
| do. | X51410 | " coil | 1045 | " " | 51710 |
| do. | 51510 | " " | 1055 | Structural parts | 2515 |
| do. | 52100 | " " | 1065 | " " | 3115 |
| Screws | 1112 | " " | X1065 | " " | 3120 |
| " | X1112 | " " | 1090 | " " good | |
| " | 2330 | " " | 1095 | physical qualities | 3125 |
| " | 2335 | " " | 6145 | do. | 3130 |
| " | 3135 | " " | 6150 | do. | 3135 |
| " | 3140 | " clutch | 1060 | do. | 3140 |
| " | 6135 | " cushion | X1055 | do. | X3140 |
| " | 6140 | " " | 1060 | Structural parts, | |
| Screw stock | 1112 | " hard-drawn | X1065 | for heavy sections | 3145 |
| " " | X1112 | " heat-treated | X1065 | do. | 3150 |
| " " | 1115 | " leaf | 1085 | do. | 3215 |
| " " | 1120 | " " | 1090 | do. | 3220 |
| Seeder discs | 1090 | " " | 1095 | do. | 4150 |
| " " | 1095 | " " | 6145 | Structural parts, | |
| Severe duty | 3435 | " " | 6150 | high physical proper- | |
| " " | 6145 | " " | 9255 | ties | 3335 |
| " " | 6150 | " " | 9260 | Structural parts, resis- | |
| Shafting | 4345 | " oil-tempered | 1055 | tance to dynamic | |
| " corrosion-re- | | " " | X1065 | stresses | 3340 |
| sistant | 51210 | " seat | 1090 | Structural parts, | |
| do. | X51410 | " " | 1095 | strength and tough- | |
| do. | 51510 | " valve | 1060 | ness | 2340 |
| " heavy duty | 4340 | " " | 1065 | do. | 2345 |
| Shafts | 4615 | " wire | 1065 | Studs | 1040 |
| " | 4620 | " " hard- | | " | 1112 |
| " | 5150 | drawn | X1055 | " | X1112 |
| " | 6115 | Springs, wire, oil tem- | | " | 2330 |
| " | 6120 | pered | X1065 | " | 2335 |
| " | 6145 | Steel balls | 1090 | " | 3135 |
| " | 6150 | " " | 1095 | " | 3140 |
| " anti-fatigue | 4640 | " cold-rolled | 1070 | " | 4815 |
| " axle, *see* Axle, | | " killed | 1010 | " | 4820 |
| shafts | | " " | 1015 | " | 6135 |
| Shafts, cam | 1020 | " oil-hardening | X1335 | " | 6140 |
| " corrosion-resistant | 51335 | " " | X1340 | " cylinder | 3125 |
| Shafts, counter- | 3115 | " " | T1335 | " " | 3130 |
| " " | 3120 | " " | T1340 | Surface hardness | 4815 |
| " crank- | X3140 | " " | T1345 | " " | 4820 |
| " " | 4340 | " | T1350 | Swaging | 1115 |
| " cyanided | 5140 | " planished | 1070 | " | 1120 |
| " propeller, *see* | | " rimmed | 1010 | Thrust washers, *see* | |
| Propeller shafts | | " " | 1015 | Washers | |

**Table 18.—Applications of S.A.E. Steels**—*Continued*

| Quality or Use | S.A.E. No. | Quality or Use | S.A.E. No. | Quality or Use | S.A.E. No. |
|---|---|---|---|---|---|
| Toughness, *see* Strength and toughness | | Tubing, front axle | 4140 | Welded tubing | 1020 |
| Transmission gearing | 3112 | " seamless | 1030 | Welding, corrosion-resistant | 30905 |
| " " | 6115 | " welded | 1020 | do. | 30915 |
| " " | 6120 | Twine holder, agricultural | 1080 | do. | 51510 |
| " shafts | 4140 | Upsetting, cold | 1035 | Wire, corrosion-resistant | 51210 |
| Trim, corrosion-resistant | 51510 | Valves, corrosion-resistant | 51335 | do. | X51410 |
| Tubing | 1040 | Washers, lock | X1055 | Wire, music | 1085 |
| " corrosion-resistant | 51210 | " " | 1060 | " spring, hard-drawn | X1055 |
| do. | X51410 | " thrust | X1055 | Wire, spring, oil tempered | X1065 |
| do. | 51510 | " " | 1060 | | |
| | | " " | 5150 | | |

carbon content in points, *i.e.*, hundredths of 1%. Thus, 2340 indicates a nickel steel of approximately 3% Ni (3.25–3.75%) and 0.40% C (0.35–0.45%); 71360 indicates a tungsten steel containing about 13% W (12–15%) and 0.60% C (0.50–70%). The prefix X is used to designate variations in the range of manganese, sulphur and chromium. The prefix T is used to differentiate the 1300 series of manganese steels of different manganese range, identified by the same number without the prefix. Table 18 shows the uses of the various S.A.E. steels. Table 19 gives the chemical composition and identifying numbers of various S.A.E. standard steels. The various steels are specified by chemical composition only, since the physical properties can be varied by heat treatment. Table 20, compiled from curves in the S.A.E. Handbook, 1936 edition, gives the effect of heat treatment on various steels.

## Notes to Table 19

**CHARACTERISTICS AND USES OF THE VARIOUS S.A.E. STEELS.**—The following notes are abstracted from the S.A.E. Handbook, 1936 Edition.

**CARBON STEELS.—S.A.E. 1010 and S.A.E. 1015.**—Made by two methods of manufacture, known as rimmed and killed steel, respectively. Rimmed steel is used for sheet, strip, rod, and wire where high surface finish is desired. Suitable for deep drawing operations, and for cold heading wire. Killed steel is preferable to rimmed steel for carburized parts, especially where both core and rim of rimmed steel are involved in heat treatment.

These steels are of low tensile values and should not be used where much strength is desired. Heat treatment by direct quenching does not increase strength or hardness materially, or improve machinability, but cold drawing or cold rolling increases strength and hardness about 20% as compared with the hot-rolled condition, which improvement is lost if the steel is heated to 1000° F. or more.

**S.A.E. 1020.**—Standard carburizing grade of carbon steel. Used for wrist-pins, cam shafts, drag links, welded tubing, and forged and automatic parts where high strength is not necessary. May be brazed, butt-welded and drawn, but is not as desirable for deep drawing as S.A.E. 1010. Is not recommended for smooth threading, turning or broaching. Does not respond materially to heat treatment unless carburized or cyanided.

**S.A.E. X1020.**—A high-manganese variant of S.A.E. 1020, with better machining and carburizing properties.

**S.A.E. 1025.**—Has fair machining properties. Forgings usually machine better without annealing, or by simple normalizing. Is not considered a carburizing steel, but is suitable for case hardening where core hardness is desirable.

**S.A.E. X1025.**—A high-manganese variant of S.A.E. 1025, with better machining and physical properties; used for hub and rim bolts, studs, etc.

**S.A.E. 1030.**—Responds well to heat treatment, and is used for forged, machined or cold worked parts requiring higher physical properties than are obtainable with the lower carbon steels. Machines satisfactorily either without annealing or by a simple normalizing. Used for seamless tubing, key stock, levers, rail brackets, etc. Suitable for case hardening where core hardness is desired or for large sections.

**S.A.E. 1035.**—Fair machining properties, suitable for small and medium size forgings of moderate physical properties. Used for wire and rod for cold upsetting. Bolts and screws made from it, less than 3/8 in. should not be quenched in water, because of danger of quenching fractures.

**S.A.E. 1040.**—A medium carbon steel of fair machining properties and deep drawing characteristics. Suitable for small and medium size forgings. Used for tubing, crankshafts, connecting-rods, tubular and solid axles, spring clips, levers, bolts, studs, etc. Quenching of thin sections or small diameters in water should be done with caution.

**Table 19.—Specifications of S.A.E. Steels**

| S.A.E. No. | C | Mn | P | S | Ni | Cr | Mo | W | V† | Si |
|---|---|---|---|---|---|---|---|---|---|---|
| Carbon Steels | | | | | | | | | | |
| 1010 | 0.05–0.15 | 0.30–0.60 | 0.045* | 0.055* | | | | | | |
| 1015 | 0.10–0.20 | 0.30–0.60 | .045* | .055* | | | | | | |
| X1015 | 0.10–0.20 | 0.70–1.00 | .045* | .055* | | | | | | |
| 1020 | 0.15–0.25 | 0.30–0.60 | .045* | .055* | | | | | | |
| X1020 | 0.15–0.25 | 0.70–1.00 | .045* | .055* | | | | | | |
| 1025 | 0.20–0.30 | 0.30–0.60 | .045* | .055* | | | | | | |
| X1025 | 0.20–0.30 | 0.70–1.00 | .045* | .055* | | | | | | |
| 1030 | 0.25–0.35 | 0.60–0.90 | .045* | .055* | | | | | | |
| 1035 | 0.30–0.40 | 0.60–0.90 | .045* | .055* | | | | | | |
| 1040 | 0.35–0.45 | 0.60–0.90 | .045* | .055* | | | | | | |
| X1040 | 0.35–0.45 | 0.40–0.70 | .045* | .055* | | | | | | |
| 1045 | 0.40–0.50 | 0.60–0.90 | .045* | .055* | | | | | | |
| X1045 | 0.40–0.50 | 0.40–0.70 | .045* | .055* | | | | | | |
| 1050 | 0.45–0.55 | 0.60–0.90 | .045* | .055* | | | | | | |
| X1050 | 0.45–0.55 | 0.40–0.70 | .045* | .055* | | | | | | |
| 1055 | 0.50–0.60 | 0.60–0.90 | .040* | .055* | | | | | | |
| X1055 | 0.50–0.60 | 0.90–1.20 | .040* | .055* | | | | | | |
| 1060 | 0.55–0.70 | 0.60–0.90 | .040* | .055* | | | | | | |
| 1065 | 0.60–0.75 | 0.60–0.90 | .040* | .055* | | | | | | |
| X1065 | 0.60–0.75 | 0.90–1.20 | .040* | .055* | | | | | | |
| 1070 | 0.65–0.80 | 0.60–0.90 | .040* | .055* | | | | | | |
| 1075 | 0.70–0.85 | 0.60–0.90 | .040* | .055* | | | | | | |
| 1080 | 0.75–0.90 | 0.60–0.90 | .040* | .055* | | | | | | |
| 1085 | 0.80–0.95 | 0.60–0.90 | .040* | .055* | | | | | | |
| 1090 | 0.85–1.00 | 0.60–0.90 | .040* | .055* | | | | | | |
| 1095 | 0.90–1.05 | 0.25–0.50 | .040* | .055* | | | | | | |
| Free Cutting Steels | | | | | | | | | | |
| 1112 | 0.08–0.16 | 0.60–0.90 | 0.09–0.13 | 0.10–0.20 | | | | | | |
| X1112 | 0.08–0.16 | 0.60–0.90 | 0.09–0.13 | 0.20–0.30 | | | | | | |
| 1115 | 0.10–0.20 | 0.70–1.00 | 0.045* | 0.075–0.15 | | | | | | |
| 1120 | 0.15–0.25 | 0.60–0.90 | 0.045* | 0.075–0.15 | | | | | | |
| X1314 | 0.10–0.20 | 1.00–1.30 | 0.045* | 0.075–0.15 | | | | | | |
| X1315 | 0.10–0.20 | 1.30–1.60 | 0.045* | 0.075–0.15 | | | | | | |
| X1330 | 0.25–0.35 | 1.35–1.65 | 0.045* | 0.075–0.15 | | | | | | |
| X1335 | 0.30–0.40 | 1.35–1.65 | 0.045* | 0.075–0.15 | | | | | | |
| X1340 | 0.35–0.45 | 1.35–1.65 | 0.045* | 0.075–0.15 | | | | | | |
| Manganese Steels | | | | | | | | | | |
| T1330 | 0.25–0.35 | 1.60–1.90 | 0.040* | 0.050 | | | | | | |
| T1335 | 0.30–0.40 | 1.60–1.90 | .040* | .050 | | | | | | |
| T1340 | 0.35–0.45 | 1.60–1.90 | .040* | .050 | | | | | | |
| T1345 | 0.40–0.50 | 1.60–1.90 | .040* | .050 | | | | | | |
| T1350 | 0.45–0.55 | 1.60–1.90 | .040* | .050 | | | | | | |
| Nickel Steels‡ | | | | | | | | | | |
| 2015 | 0.10–0.20 | 0.30–0.60 | 0.040* | 0.050* | 0.40–0.60 | | | | | |
| 2115 | 0.10–0.20 | 0.30–0.60 | .040* | .050* | 1.25–1.75 | | | | | |
| 2315 | 0.10–0.20 | 0.30–0.60 | .040* | .050* | 3.25–3.75 | | | | | |
| 2320 | 0.15–0.25 | 0.30–0.60 | .040* | .050* | 3.25–3.75 | | | | | |
| 2330 | 0.25–0.35 | 0.50–0.80 | .040* | .050* | 3.25–3.75 | | | | | |
| 2335 | 0.30–0.40 | 0.50–0.80 | .040* | .050* | 3.25–3.75 | | | | | |
| 2340 | 0.35–0.45 | 0.60–0.90 | .040* | .050* | 3.25–3.75 | | | | | |
| 2345 | 0.40–0.50 | 0.60–0.90 | .040* | .050* | 3.25–3.75 | | | | | |
| 2350 | 0.45–0.55 | 0.60–0.90 | .040* | .050* | 3.25–3.75 | | | | | |
| 2515 | 0.10–0.20 | 0.30–0.60 | .040* | .050* | 4.75–5.25 | | | | | |
| Nickel Chromium Steels‡ | | | | | | | | | | |
| 3115 | 0.10–0.20 | 0.30–0.60 | 0.040* | 0.050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3120 | 0.15–0.25 | 0.30–0.60 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3125 | 0.20–0.30 | 0.50–0.80 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3130 | 0.25–0.35 | 0.50–0.80 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3135 | 0.30–0.40 | 0.50–0.80 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3140 | 0.35–0.45 | 0.60–0.90 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| X3140 | 0.35–0.45 | 0.60–0.90 | .040* | .050* | 1.00–1.50 | 0.60–0.90 | | | | |
| 3145 | 0.40–0.50 | 0.60–0.90 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3150 | 0.45–0.55 | 0.60–0.90 | .040* | .050* | 1.00–1.50 | 0.45–0.75 | | | | |
| 3215 | 0.10–0.20 | 0.30–0.60 | .040* | .050* | 1.50–2.00 | 0.90–1.25 | | | | |

*See footnotes at end of table.*

*Table continued on following page.*

**Table 19.—Specifications of S.A.E. Steels—*Continued***

| S.A.E. No. | C | Mn | P | S | Ni | Cr | Mo | W | V† | Si |
|---|---|---|---|---|---|---|---|---|---|---|
| Nickel Chromium Steels‡—*Continued* | | | | | | | | | | |
| 3220 | 0.15–0.25 | 0.30–0.60 | .040* | .050* | 1.50–2.00 | 0.90–1.25 | ......... | ......... | ..... | ......... |
| 3230 | 0.25–0.35 | 0.30–0.60 | .040* | .050* | 1.50–2.00 | 0.90–1.25 | ......... | ......... | ..... | ......... |
| 3240 | 0.35–0.45 | 0.30–0.60 | .040* | .050* | 1.50–2.00 | 0.90–1.25 | ......... | ......... | ..... | ......... |
| 3245 | 0.40–0.50 | 0.30–0.60 | .040* | .050* | 1.50–2.00 | 0.90–1.25 | ......... | ......... | ..... | ......... |
| 3250 | 0.45–0.55 | 0.30–0.60 | .040* | .050* | 1.50–2.00 | 0.90–1.25 | ......... | ......... | ..... | ......... |
| 3312 | 0.17* | 0.30–0.60 | .040* | .050* | 3.25–3.75 | 1.25–1.75 | ......... | ......... | ..... | ......... |
| 3325 | 0.20–0.30 | 0.30–0.60 | .040* | .050* | 3.25–3.75 | 1.25–1.75 | ......... | ......... | ..... | ......... |
| 3335 | 0.30–0.40 | 0.30–0.60 | .040* | .050* | 3.25–3.75 | 1.25–1.75 | ......... | ......... | ..... | ......... |
| 3340 | 0.35–0.45 | 0.30–0.60 | .040* | .050* | 3.25–3.75 | 1.25–1.75 | ......... | ......... | ..... | ......... |
| 3415 | 0.10–0.20 | 0.30–0.60 | .040* | .050* | 2.75–3.25 | 0.60–0.95 | ......... | ......... | ..... | ......... |
| 3435 | 0.30–0.40 | 0.30–0.60 | .040* | .050* | 2.75–3.25 | 0.60–0.95 | ......... | ......... | ..... | ......... |
| 3450 | 0.45–0.55 | 0.30–0.60 | .040* | .050* | 2.75–3.25 | 0.60–0.95 | ......... | ......... | ..... | ......... |
| Molybdenum Steels‡ | | | | | | | | | | |
| 4130 | 0.25–0.35 | 0.50–0.80 | 0.040* | 0.050* | ......... | 0.50–0.80 | 0.15–0.25 | ......... | ..... | ......... |
| X4130 | 0.25–0.35 | 0.40–0.60 | .040* | .050* | ......... | 0.80–1.10 | 0.15–0.25 | ......... | ..... | ......... |
| 4135 | 0.30–0.40 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | 0.15–0.25 | ......... | ..... | ......... |
| 4140 | 0.35–0.45 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | 0.15–0.25 | ......... | ..... | ......... |
| 4150 | 0.45–0.55 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | 0.15–0.25 | ......... | ..... | ......... |
| 4340 | 0.35–0.45 | 0.50–0.80 | .040* | .050* | 1.50–2.00 | 0.50–0.80 | 0.30–0.40 | ......... | ..... | ......... |
| 4345 | 0.40–0.50 | 0.50–0.80 | .040* | .050* | 1.50–2.00 | 0.60–0.90 | 0.15–0.25 | ......... | ..... | ......... |
| 4615 | 0.10–0.25 | 0.40–0.70 | .040* | .050* | 1.65–2.00 | ......... | 0.20–0.30 | ......... | ..... | ......... |
| 4620 | 0.15–0.25 | 0.40–0.70 | .040* | .050* | 1.65–2.00 | ......... | 0.20–0.30 | ......... | ..... | ......... |
| 4640 | 0.35–0.45 | 0.50–0.80 | .040* | .050* | 1.65–2.00 | ......... | 0.20–0.30 | ......... | ..... | ......... |
| 4815 | 0.10–0.20 | 0.40–0.60 | .040* | .050* | 3.25–3.75 | ......... | 0.20–0.30 | ......... | ..... | ......... |
| 4820 | 0.15–0.25 | 0.40–0.60 | .040* | .050* | 3.25–3.75 | ......... | 0.20–0.30 | ......... | ..... | ......... |
| Chromium Steels‡ | | | | | | | | | | |
| 5120 | 0.15–0.25 | 0.30–0.60 | 0.040* | 0.050* | ......... | 0.60–0.90 | ......... | ......... | ..... | ......... |
| 5140 | 0.35–0.45 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | ..... | ......... |
| 5150 | 0.45–0.55 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | ..... | ......... |
| 52100 | 0.95–1.10 | 0.20–0.50 | .030* | .035* | ......... | 1.20–1.50 | ......... | ......... | ..... | ......... |
| Chromium Vanadium Steels‡ | | | | | | | | | | |
| 6115 | 0.10–0.20 | 0.30–0.60 | 0.040* | 0.050* | ......... | 0.80–1.10 | ......... | ......... | 0.18 | ......... |
| 6120 | 0.15–0.25 | 0.30–0.60 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6125 | 0.20–0.30 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6130 | 0.25–0.35 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6135 | 0.30–0.40 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6140 | 0.35–0.45 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6145 | 0.40–0.50 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6150 | 0.45–0.55 | 0.60–0.90 | .040* | .050* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| 6195 | 0.90–1.05 | 0.20–0.45 | .030* | .035* | ......... | 0.80–1.10 | ......... | ......... | .18 | ......... |
| Tungsten Steels‡ | | | | | | | | | | |
| 71360 | 0.50–0.70 | 0.30* | 0.035* | 0.040* | ......... | 3.00–4.00 | ......... | 12.00–15.00 | ..... | ......... |
| 71660 | 0.50–0.70 | .30* | .035* | .040* | ......... | 3.00–4.00 | ......... | 15.00–18.00 | ..... | ......... |
| 7260 | 0.50–0.70 | .30* | .035* | .040* | ......... | 0.50–1.00 | ......... | 1.50–2.00 | ..... | ......... |
| Silicon Manganese Steels | | | | | | | | | | |
| 9255 | 0.50–0.60 | 0.60–0.90 | 0.040* | 0.050* | ......... | ......... | ......... | ......... | ..... | 1.80–2.20 |
| 9260 | 0.55–0.65 | 0.60–0.90 | .040* | .050* | ......... | ......... | ......... | ......... | ..... | 1.80–2.20 |
| Corrosion- and Heat-resisting Alloys | | | | | | | | | | |
| 30905 | 0.08* | 0.20–0.70 | 0.030* | 0.030* | 8.00–10.00 | 17.00–20.00 | ......... | ......... | ..... | 0.75* |
| 30915 | 0.09–0.20 | 0.20–0.70 | .030* | .030* | 8.00–10.00 | 17.00–20.00 | ......... | ......... | ..... | .75* |
| 51210 | 0.12* | 0.60* | .030* | .030* | ......... | 11.50–13.00 | ......... | ......... | ..... | .50* |
| X51410 | 0.12* | 0.60* | .030* | 0.15–0.50 | ......... | 13.00–15.00 | ......... | ......... | ..... | .50* |
| 51335 | 0.25–0.40 | 0.60* | .030* | 0.030* | ......... | 12.00–14.00 | ......... | ......... | ..... | .50* |
| 51510 | 0.12* | 0.60* | .030* | .030* | ......... | 14.00–16.00 | ......... | ......... | ..... | .50* |
| 51710 | 0.12* | 0.60* | .030* | .030* | ......... | 16.00–18.00 | ......... | ......... | ..... | .50* |

* Maximum. † Desired content; minimum = 0.15.

‡ Silicon range of all S.A.E. basic open-hearth steels shall be 0.15–0.30. For electric and acid open-hearth alloy steels silicon content shall be 0.15 minimum.

**S.A.E. X1040.**—A low-manganese variant of S.A.E. 1040, used where deep hardening properties are not required. More suitable for water quenching than S.A.E. 1040.

**S.A.E. 1045.**—A medium carbon steel used for larger sizes of automotive forgings, as crank-shafts, starter ring gears, axles and spline shafts. Quenching of thin sections or small diameter parts in water should be done with caution.

**S.A.E. X1045.**—A low-manganese variant of S.A.E. 1045 where deep hardening properties are not required. See also S.A.E. X1335.

**S.A.E. 1050.**—A medium carbon steel for larger sections than S.A.E. 1045. See also S.A.E. X1335.

**S.A.E. X1050.**—A low-manganese variant of S.A.E. 1050 for heavier forgings and sections.

**S.A.E. 1055.**—Used for hard-drawn, oil-tempered or patented spring wire, and for general coil spring purposes.

**S.A.E. X1055.**—Used for snap rings, cushion springs, lock and thrust washers, hard-drawn spring wire, etc.

**S.A.E. 1060.**—Used for the same purposes as S.A.E. X1055.

**S.A.E. 1065.**—Used for smaller sections of valve springs, hard-drawn spring wire and general purpose coil springs.

**S.A.E. X1065.**—Used for larger sections of coil springs of hard-drawn or oil-tempered wire, and heat-treated springs.

**S.A.E. 1070.** Used in cold rolled or planished form for clutch discs; in agricultural industry for mower sections, plow beams, etc.

**S.A.E. 1075.**—Used in agricultural industry for cultivator discs, coulter blades and plows.

**S.A.E. 1080.**—Used in agricultural industry for plowshares, twine holders, knotter discs, springs, molds, shovels, harrow and plow discs.

**S.A.E. 1085.**—Used for bumper bars, lock pins, clutch discs, leaf springs, music wire; in agricultural industry for mower knives.

**S.A.E. 1090.**—Used for steel balls, pins, keys, bumper bars, leaf and coil springs; in agricultural industry, for harrow and seeder discs, seat springs, hay rake teeth and bundle carrier teeth for binders.

**S.A.E. 1095.**—For uses similar to those of S.A.E. 1090.

**S.A.E. 1112.**—A Bessemer steel, often called screw stock. Used for studs, screw and automatic machine products. Has good machining properties, but is cold short and should not be used for vital parts. May be carburized and cyanided, but open-hearth steels are recommended where heat treating is required.

**S.A.E. X1112.**—A variant of S.A.E. 1112, in which sulphur is added to improve finish and machinability. Should not be used in vital parts.

**S.A.E. 1115 and S.A.E. 1120.**—Known as open-hearth screw stock; inferior in machining properties to S.A.E. 1112 and S.A.E. X1112, but have a better combination of strength and toughness. Used for case-hardened parts and for bending, swaging, riveting and forming operations.

**S.A.E. X1314 and S.A.E. X1315.**—Used for case-hardened parts where good machining properties are desired. S.A.E. X1314 is preferable for drastic quenching of light sections where core toughness is essential.

**S.A.E. X1330.**—A free-cutting manganese steel that may be substituted for S.A.E. 1035 and S.A.E. 1040 where improved machining, deeper hardening and higher physical properties are required.

**S.A.E. X1335 and S.A.E. X1340.**—Free cutting manganese steels that may be substituted for S.A.E. 1045 and 1050 where improved machining, deeper hardening, and higher physical properties are desired. They are oil-hardening steels, and care should be used in water quenching.

**S.A.E. T1330.**—A manganese alloy steel that for certain applications can be used interchangeably with other medium alloy steels of similar carbon content.

**S.A.E. T1335 and S.A.E. T1340.**—Manganese oil quenching steels that for certain applications can be used interchangeably with other medium alloy steels of similar carbon content. Great care should be used in water quenching.

**S.A.E. T1345 and S.A.E. T1350.**—See note T1335.

**NICKEL STEELS.—S.A.E. 2015.**—A 0.5% nickel steel applicable as a substitute for carburizing grades of plain carbon steel. Will harden from an oil quench and may replace carbon steel for carburized parts if distortion is a factor, and water quenching undesirable.

**S.A.E. 2115.**—A carburizing grade of 1.5% nickel steel, used as an economical substitute for S.A.E. 2315. Quenches readily in oil, but does not have toughness or strength equal to 3 $^1/_2$% nickel steels.

**S.A.E. 2315.**—Primarily intended for carburizing. Should be given a single quench after carburizing when maximum hardness only is required, and a double quench when maximum core refinement and case hardness are necessary. Has low critical points and the case quenches to file hardness in oil. If used for gears of high accuracy and high strength, carburizing preferably should be preceded by normalizing at least 50 deg. F. above carburizing temperature.

**S.A.E. 2320.**—Similar to and sometimes used interchangeably with S.A.E. 2315. In the higher carbon range, it shows marked increase in core hardness and should not be used for thin sections. Machinability better than S.A.E. 2315. See S.A.E 2315 for note on use in gears.

**S.A.E. 2330 and 2335.**—For agricultural, automotive and aviation parts requiring considerable strength and toughness. Have good machining qualities, but are not carburizing. Used for keys, heat-treated butts, screws, studs, nuts, levers, etc.

**S.A.E. 2340 and S.A.E. 2345.**—Used for structural purposes where great strength and toughness are required. Are sensitive to heat treatment, and water quenching is not recommended. Typical uses are propeller shafts, spline shafts, axle shafts, etc., and (S.A.E. 2345) gears.

**S.A.E. 2350.**—Used for oil-hardened gears and heat-treated parts of large section.

**FIVE PERCENT NICKEL STEELS.—S.A.E. 2515 and S.A.E. 2512.**—Used for carburized parts requiring exceptionally tough core after heat treatment. Will not give the surface hardness of some of the other steels. For excellent core toughness, carbon content should be held to 0.17% max. The steel then is known as S.A.E. 2512.

**NICKEL-CHROMIUM STEELS.—S.A.E. 3115 and S.A.E. 3120.**—Intended primarily for carburized parts, but are more responsive to heat treatment than the carbon steels. May be heat treated for structural parts requiring a certain degree of strength. Carburizing should be preceded by normalizing at 50° F. above carburizing temperature. Applications are ring gears and pinions, spline shafts and countershafts, transmission gearing, piston pins, etc.

**S.A.E. 3125 and S.A.E. 3130.**—Used for water-quenched parts where greater strength and toughness are required than is possible with plain carbon steels. Used for cylinder studs, steering arm bolts, keys, nuts, connecting-rod bolts and structural parts requiring good physical qualities.

**S.A.E. 3135 and S.A.E. 3140.**—Used for heat-treated structural parts where good physical properties are required. Typical applications are crank-shafts, axle shafts, spline shafts, links for drive chains, studs, screws, etc.

**S.A.E. X3140.**—A higher-chromium variant of S.A.E. 3140, used where improved hardening characteristics are desired for parts of large section.

**S.A.E. 3145 and S.A.E. 3150.**—Used for oil-hardened gears and structural parts of heavy section.

**S.A.E. 3215 and S.A.E. 3220.**—Used for carburized parts requiring greater core strength than is possible with S.A.E. 3115 and S.A.E. 3120. S.A.E. 3220 should be used only for massive sections. These steels are not recommended for water quenching. Should be preceded by normalizing at 50° F. above carburizing temperature, followed by an anneal for machinability.

**S.A.E. 3230 and S.A.E. 3240.**—For parts requiring higher physical properties than are possible with steels of lower alloy content, and especially for sections over 1 1/2 in. diameter. Not recommended for water quenching.

**S.A.E. 3245 and S.A.E. 3250.**—For oil-hardened machined or forged parts that require high physical properties, as axle shafts, gears, and similar heavy duty parts.

**S.A.E. 3312.**—For carburized parts requiring a core of high strength and toughness.

**S.A.E. 3325.**—An oil-hardening steel, sometimes used interchangeably with S.A.E. 3230 for parts requiring high physical properties and capable of resisting severe dynamic stresses. Not recommended for water quenching.

**S.A.E. 3335 and S.A.E. 3340.**—Used for structural parts requiring high physical qualities and ability to resist severe dynamic stress.

**S.A.E. 3415.**—For carburized parts with a core of very high strength and toughness.

**S.A.E. 3435.**—For parts subject to severe service conditions.

**S.A.E. 3450.**—Used interchangeably with S.A.E. 3240 and S.A.E. 3340 for heat-treated machined or forged parts that require exceptionally high physical properties and ability to resist severe dynamic stress.

**MOLYBDENUM STEELS.—S.A.E. 4130.**—A chromium molybdenum steel used for axles, steering knuckles, steering knuckle arms, and sheet and strip for aircraft. Care should be used in water quenching.

**S.A.E. X4130.**—A higher-chromium and lower-manganese variant of, and used for the same purposes as, S.A.E. 4130.

**S.A.E. 4140.**—Used for rear axles, front axle tubing, propeller shafts, transmission shafts and forgings for aircraft.

**S.A.E. 4150.**—Used for parts of heavy section.

**S.A.E. 4340.**—Suitable for oil-hardening gears, axle shafts, Diesel engine crank-shafts, and heavy duty shafting over 2-in. diameter, where high strength and anti-fatigue properties are required.

**Table 20.—Physical Properties of Certain S.A.E. Steels with Various Heat Treatments**

| Draw Temp., deg. F. | Tensile Strength, 1000 lb. per sq.in. | Yield Point, 1000 lb. per sq.in. | Reduction of Area, percent | Elongation in 2 in., percent | Rockwell Hardness | Tensile Strength, 1000 lb. per sq.in. | Yield Point, 1000 lb. per sq.in. | Reduction of Area, percent | Elongation in 2 in., percent | Rockwell Hardness |
|---|---|---|---|---|---|---|---|---|---|---|
| | S.A.E. 1035. Quenched in water at 1525–1575° F. | | | | | S.A.E. 1035. Quenched in oil at 1525–1575° F. | | | | |
| 800 | 106 | 75 | 52 | 23 | B96 | 97.5 | 66 | 48.5 | 19 | B93 |
| 900 | 98 | 70 | 55 | 25 | B94 | 92.5 | 63.5 | 52 | 21 | B92 |
| 1000 | 95 | 77.5 | 56.5 | 27.5 | B92 | 90 | 60 | 55 | 23 | B91 |
| 1100 | 93 | 73 | 60 | 28.5 | B91 | 88.5 | 56 | 58 | 25 | B90 |
| 1200 | 89.5 | 57 | 64 | 30 | B90 | 87 | 53 | 61 | 27 | B89 |
| 1300 | 85 | 49 | 68 | 33 | B88 | 83 | 51 | 63 | 28.5 | B87 |
| | S.A.E. 1045. Quenched in water at 1475–1525° F. | | | | | S.A.E. 1045. Quenched in oil at 1475–1525° F. | | | | |
| 800 | 121 | 91 | 46.5 | 16 | C24 | 116 | 80 | 42 | 16 | B99 |
| 900 | 115 | 84 | 48.5 | 20 | B99 | 110 | 84 | 45 | 18 | B98 |
| 1000 | 111 | 77 | 51 | 22.5 | B98 | 107 | 70 | 48 | 20 | B97 |
| 1100 | 107.5 | 71 | 52.5 | 25 | B96 | 104 | 67.5 | 50 | 21.5 | B96 |
| 1200 | 103 | 66 | 55 | 27.5 | B96 | 99 | 65 | 52 | 22 | B95 |
| 1300 | 97.5 | 60 | 56 | 28 | B93 | 95 | 62 | 55 | 22.5 | B92 |
| | S.A.E. T1330. Normalized at 1650–1750° F. Quenched in water at 1525–1575° F. | | | | | S.A.E. T1340. Normalized at 1600–1700° F. Quenched in oil at 1475–1525° F. | | | | |
| 800 | 161.5 | 137.5 | 42.5 | 13 | C35 | 162 | 135 | 40 | 10 | C35 |
| 900 | 140 | 117.5 | 48 | 16 | C30 | 148.5 | 120 | 47 | 12.5 | C32 |
| 1000 | 122 | 100 | 52 | 19 | C24 | 136 | 106.5 | 51.5 | 16 | C29 |
| 1100 | 108 | 86 | 57 | 21.5 | B96 | 123 | 92.5 | 57 | 18 | C24 |
| 1200 | 98.5 | 73 | 59.5 | 23.5 | B96 | 112 | 81 | 59 | 19 | C20 |
| 1300 | 92 | 68 | 61.5 | 25.5 | B91 | 102 | 70 | 60 | 20 | B95 |
| | S.A.E. 2330. Normalized at 1625–1675° F. Quenched in water at 1450–1500° F. | | | | | S.A.E. 2340. Normalized at 1625–1725° F. Quenched in oil at 1425–1475° F. | | | | |
| 800 | 152.5 | 126.5 | 52.5 | 14.5 | C32 | 165 | 148 | 46.5 | 14 | C35 |
| 900 | 137 | 110 | 58 | 16 | C29 | 148 | 127.5 | 50 | 16 | C32 |
| 1000 | 121 | 95 | 62.5 | 18 | B99 | 130 | 108 | 54 | 20 | C24 |
| 1100 | 108 | 80 | 66.5 | 20 | B96 | 115 | 90 | 58 | 25 | C21 |
| 1200 | 95 | 77.5 | 68 | 22 | B91 | 102.5 | 75 | 60 | 27 | B96 |
| 1300 | 92 | 56 | 69 | 26 | B86 | 93 | 63 | 61 | 28 | B90 |
| | S.A.E. 3130. Normalized at 1650–1750° F. Quenched in water at 1500–1550° F. | | | | | S.A.E. 3140. Normalized at 1600–1700° F. Quenched in oil at 1475–1525° F. | | | | |
| 800 | 167 | 142 | 55 | 10 | C33 | 175 | 151 | 45 | 12 | C35 |
| 900 | 145 | 123 | 60 | 17 | C29 | 157 | 132.5 | 50 | 16 | C32 |
| 1000 | 127 | 115 | 64.5 | 21 | C24 | 148 | 110.6 | 55 | 18 | C28 |
| 1100 | 110 | 90 | 68 | 25 | B96 | 122.5 | 100 | 58.5 | 19 | C24 |
| 1200 | 100 | 77.5 | 71.5 | 27.5 | B94 | 111 | 85 | 62 | 20 | C20 |
| 1300 | 92.5 | 67.5 | 73 | 28.5 | B91 | 102.5 | 70 | 65 | 20 | B96 |
| | S.A.E. 3230. Normalized at 1625–1675° F. Quenched in oil at 1500–1575° F. | | | | | S.A.E. 3240. Normalized at 1625–1725° F. Quenched in oil at 1475–1525° F. | | | | |
| 800 | 178 | 152 | 48 | 16 | C37 | 202 | 180 | 44 | 16 | C41 |
| 900 | 157 | 132 | 55 | 18 | C33 | 177.5 | 152.5 | 50 | 17.5 | C36 |
| 1000 | 137 | 113 | 59.5 | 20.5 | C28 | 155 | 132.5 | 56 | 19 | C34 |
| 1100 | 119 | 96 | 63 | 22.5 | C23 | 137.5 | 116 | 60 | 21 | C30 |
| 1200 | 105 | 80.5 | 67 | 25 | B97 | 122.5 | 100 | 62.5 | 22 | C25 |
| 1300 | 97 | 70 | 68.5 | 27.5 | B94 | 110 | 87.5 | 65 | 23 | C21 |
| | S.A.E. 3250. Normalized at 1600–1700° F. Quenched in oil at 1425–1475° F. | | | | | S.A.E. 3325. Normalized at 1600–1700° F. Quenched in oil at 1475–1525° F. | | | | |
| 800 | 220 | 200 | 39 | 12.5 | C44 | 170.5 | 142.5 | 55 | 18 | C37 |
| 900 | 192 | 172 | 47 | 15 | C40 | 148 | 126 | 58.5 | 20 | C32 |
| 1000 | 168 | 148.5 | 52 | 17.5 | C35 | 130 | 110 | 62 | 22.5 | C28 |
| 1100 | 149 | 128.5 | 57 | 19 | C31 | 117.5 | 99 | 65 | 25 | C24 |
| 1200 | 132 | 112 | 59 | 20 | C28 | 110 | 92 | 67 | 27 | C21 |
| 1300 | 118 | 98 | 60 | 21 | C23 | ...... | ...... | ...... | ...... | ...... |
| | S.A.E. 3335. Normalized at 1600–1700° F. Quenched in oil at 1425–1475° F. | | | | | S.A.E. 3435. Normalized at 1550–1650° F. Quenched in oil at 1425–1475° F. | | | | |
| 800 | 188 | 168.5 | 50 | 12.5 | C40 | 188 | 162.5 | 55 | 15 | C36 |
| 900 | 167 | 146 | 56 | 17.5 | C35 | 166 | 144 | 57.5 | 16.5 | C33 |
| 1000 | 148 | 127.5 | 59.5 | 20 | C32 | 157 | 127 | 60 | 18.5 | C30 |
| 1100 | 132 | 112 | 61.5 | 21 | C29 | 130 | 110 | 62.5 | 20 | C26 |
| 1200 | 118.5 | 100 | 62.5 | 22 | C23 | 119.5 | 97.5 | 65 | 22 | C21 |

**Table 20.**—(*Continued*)

| Draw Temp., deg. F. | Tensile Strength, 1000 lb. per sq.in. | Yield Point, 1000 lb. per sq.in. | Reduction of Area, percent | Elongation in 2 in., percent | Rockwell Hardness | Tensile Strength, 1000 lb. per sq.in. | Yield Point, 1000 lb. per sq.in. | Reduction of Area, percent | Elongation in 2 in., percent | Rockwell Hardness |
|---|---|---|---|---|---|---|---|---|---|---|
| | S.A.E. 3450. Normalized at 1550–1650° F. Quenched in oil at 1400–1450° F. | | | | | S.A.E. 4140. Normalized at 1650–1750° F. Quenched in oil at 1525–1625° F. | | | | |
| 800 | 200 | 175 | 52 | 12.5 | C40 | 180 | 156 | 36 | 10 | C38 |
| 900 | 177.5 | 157.5 | 53 | 15 | C37 | 170 | 146 | 40 | 11 | C37 |
| 1000 | 156.5 | 138 | 55 | 17.5 | C34 | 157.5 | 132 | 45 | 12 | C34 |
| 1100 | 140 | 120 | 58 | 18.5 | C31 | 142 | 118 | 51 | 14 | C31 |
| 1200 | 126 | 117.5 | 61.5 | 21 | C28 | 126.5 | 101 | 56 | 17.5 | C25 |
| 1300 | ........ | ........ | ........ | ........ | ....... | 110 | 86 | 59 | 21 | C20 |
| | S.A.E. 4340. Normalized at 1625–1725° F. Quenched in oil at 1475–1550° F. | | | | | S.A.E. 4640. Normalized at 1650–1700° F. Quenched in oil at 1450–1500° F. | | | | |
| 800 | 210 | 190 | 40 | 11 | C44 | 178.5 | 147.5 | 47 | 10 | C38 |
| 900 | 195 | 175 | 42 | 11 | C42 | 158.5 | 130 | 49 | 11 | C34 |
| 1000 | 178 | 157.5 | 45 | 12 | C38 | 142 | 115 | 51 | 12 | C31 |
| 1100 | 160 | 138 | 48.5 | 15 | C35 | 138.5 | 102.5 | 55 | 15 | C26 |
| 1200 | 140 | 119 | 52.5 | 18 | C30 | 119 | 93 | 58 | 19 | C23 |
| 1300 | 120 | 95 | 60 | 20 | C23 | 115 | 88 | 61.5 | 22 | C21 |
| | S.A.E. 5140. Normalized at 1625–1725° F. Quenched in oil at 1500–1600° F. | | | | | S.A.E. 6140. Normalized at 1650–1750° F. Quenched in oil at 1550–1650° F. | | | | |
| 800 | 180 | 160 | 42.5 | 13 | C38 | 180 | 160 | 42.5 | 13 | C38 |
| 900 | 160 | 140 | 47.5 | 14 | C35 | 160 | 140 | 47.5 | 14 | C35 |
| 1000 | 141 | 120 | 52 | 15 | C31 | 142 | 121 | 52 | 15 | C31 |
| 1100 | 126 | 102 | 56 | 17 | C25 | 127.5 | 115 | 56 | 17.5 | C26 |
| 1200 | 112 | 88 | 59 | 19 | C20 | 115 | 92.5 | 59 | 19 | C21 |
| 1300 | 100 | 78 | 62 | 21 | B94 | 108 | 87 | 62 | 20.5 | B96 |

**S.A.E. 4345.**—Used for large gears and shafting, rear axle shafts and heavy sections and parts where high tensile and anti-fatigue properties are essential. Often specified for axle shafts over 2 in. diameter.

**S.A.E. 4615 and S.A.E. 4620.**—Used for carburized gears, shafts and parts where high fatigue resistance and high tensile properties are requisite.

**S.A.E. 4640.**—See S.A.E. 4615.

**S.A.E. 4815 and S.A.E. 4820.**—Used interchangeably with 5% nickel steel for gears and carburized parts. Also used for steering knuckle pins, roller bearings, chain pins, and high strength bolts and studs. Have excellent core and surface hardness and good wear resistance.

**CHROMIUM STEELS.—S.A.E. 5120.**—A plain chromium alloy steel of deep hardening type. Hardens satisfactorily in oil. Equal in physical properties to S.A.E. 2320 and S.A.E. 3120, except that it does not have the tough fibrous core of nickel-chromium-steels.

**S.A.E. 5140.**—May be used interchangeably with S.A.E. 2340 and 3140 for heat-treated forgings requiring greater strength and toughness than are obtainable in plain carbon steel. Used for gears and shafts hardened by direct cyaniding.

**S.A.E. 5150.**—An oil-hardening chromium steel used for gears, shafts, thrust washers and springs. Is a deep hardening type.

**S.A.E. 52100.**—Used for races and balls or rollers of anti-friction bearings.

**CHROMIUM VANADIUM STEELS.—S.A.E. 6115 and S.A.E. 6120.**—Normally are fine-grained steels used for carburized parts such as shafts, rear axle and transmission gearing. Are interchangeable with other alloy steels of similar carbon content.

**S.A.E. 6125 and S.A.E. 6130.**—Suitable for oil- or water-quenched parts. Steels of over 0.30 carbon should not be quenched in water.

**S.A.E. 6135 and S.A.E. 6140.**—Used for studs, screws, connecting rods, axle shafts, propeller shafts and similar applications. S.A.E. 6140 sometimes is used for gears hardened by direct cyaniding.

**S.A.E. 6145 and S.A.E. 6150.**—Used for heat-treated forgings and machined parts subjected to severe conditions, and requiring high strength and anti-fatigue properties. Also used for leaf and coil springs, gears, shafts and heavy forgings.

**S.A.E. 6195.**—Principally used for balls, rollers and races of anti-friction bearings.

**SILICON MANGANESE STEELS.—S.A.E. 9255 and S.A.E. 9260.**—Used principally for leaf springs. Mn on the high side of the range is specified for leaf spring sections 3/8 in. and over, and on the low side of the range for sections under 3/8 in.

**CORROSION AND HEAT-RESISTING STEELS.**—(See also p. 3-19). **S.A.E. 30905 and S.A.E. 30915.**—Standard so-called 18-8 Cr-Ni stainless steel, suitable for heat resistance up to 1500 ° F. May be forged, riveted, welded, cold-worked and deep drawn. Do not respond to hardening treatment. Somewhat difficult to machine.

**S.A.E. 51210.**—A stainless steel used for strip, rod and wire, shafting, forging, tubing, nuts, rivets, etc., where corrosion resistance is required. Can be hammered, forged, cold-worked, welded and machined at reduced speeds. Resists oxidation up to 1200° F.

**S.A.E. X51410.**—A free-machining, high-sulphur variant of S.A.E. 51210. Contains also Zr, Se, Mo and Cu. Applicable to the same uses as S.A.E. 51210 but machines easily at the speeds and feeds used for ordinary steel. May be readily polished.

**S.A.E. 51335.**—Standard cutlery grade of stainless steel. Used only in the hardened condition. Should be forged between 2000 and 1600° F. Has air hardening properties, and must be annealed after forging if it is to be machined. Used for shafts, bushings and bearings subject to corrosive conditions. Will resist tarnish from vegetable and mineral oil, moist air, water, steam and alkalis. Will not resist hydrochloric, sulphurous or hydrofluoric acid.

**S.A.E. 51510.**—Used for same purposes as S.A.E. 51210, but only in annealed condition. Is somewhat superior to S.A.E. 51210 in corrosion resistance. Can be forged, cold-worked and welded, but is not a good machining steel. Used for automobile trim and for bright surfaces maintained under exposure to oxidizing conditions.

**S.A.E. 51710.**—Used solely for ornamental purposes. Is more corrosion resistant than S.A.E. 51510. Not susceptible to heat treatment except annealing.

# WELDING OF IRON AND STEEL

**By J. C. Hodge and R. E. Hiller**

## 1. TYPES OF WELDS

**WELDING** has been defined as a localized consolidation of metals, accomplished by the application of either heat or pressure or a combination of the two. The temperature for accomplishing welding may vary; the source of the welding heat may differ; and pressure may be applied by various means or methods. The several welding processes involve different sources and degrees of welding heat, and also may involve different methods of applying pressure. These variations serve as a basis of classifying welding processes. See Fig. 1.

**WELDING PROCESSES** are divided into two general groups: 1. Methods in which the joint surfaces are heated to within the plastic temperature range and pressure applied. 2. Methods in which the joint surfaces are heated above the melting temperature of the parent metal and fusion obtained without pressure (fusion welding).

Soldering and brazing are joining processes allied to fusion welding, in which junction is accomplished by fusion of an intermediate metal of relatively low melting temperature between the parts to be joined without fusing their surfaces. These processes will not be further described.

The two main divisions of the welding processes are further classified according to the source of heat; for example, the plastic welding processes into forge welding, electric resistance welding, and Thermit pressure; the fluid or fusion welding processes into gas, electric arc, and Thermit welding.

**FORGE WELDING** is the oldest form of welding. The parts to be joined are heated locally in a furnace to the plastic temperature range and then subjected to pressure by hammering or rolling either by hand or by machine. This process is used for a wide variety of applications, from the miscellaneous work of the blacksmith to the joining of steel plates up to approximately 2 in. thick in the fabrication of boiler drums and other pressure vessels. Large tonnages of lap-welded and butt-welded tubes have been produced by passing heated formed skelp through pressure rolls. Forge welding has been largely superseded by other more recent welding processes.

**RESISTANCE WELDING.**—The various electric resistance welding processes are in principle similar to forge welding, in that the weld is made by application of pressure while the surfaces being joined are heated to plastic temperature. All resistance welding processes involve heating the edges of the parts to be joined by the passage of a heavy current across the joint. Heat is generated locally at the joint by reason of the greater electrical resistance of the circuit at the joint contact surfaces. Either direct or alternating current may be used, but generally it is most convenient to use alternating current, since the desired high amperage at low voltage is readily supplied through suitable transformers. Because of the high current required and the necessary "push-up" forces, the surfaces which may be joined by this method are limited. For a given cross-section to be welded, the power requirements vary widely with the time interval of current

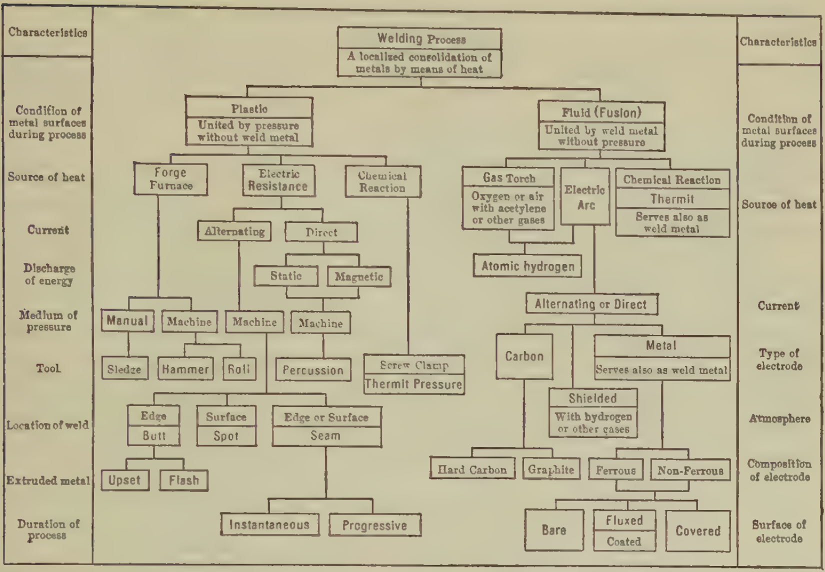


FIG. 1. Classification of the Principal Welding Processes

application to produce the weld. For the usual resistance weld, with low-carbon steel, requiring a heating period of several seconds, 30 kva. or more are required per square inch of weld surface. As the time interval is decreased the power requirement increases and, where welds are made in a fraction of a second, 1500 to 2000 kva. per sq. in. are not uncommon. The necessary pressure or push-up force applied after the joint surfaces have been heated to plastic temperature is approximately 5000 lb. per sq. in. of weld surface.

Resistance welding processes are particularly applicable to mass production, because special equipment is required for each type of welding job.

**Electrodes** for introducing the welding current into the work generally are faced with a hard copper alloy to avoid excessive replacement, which would be necessary if soft copper alone were used.

**Resistance Butt Welding.**—In resistance butt welding the parts to be joined are pressed into contact and electric current is passed through the grips (electrodes) and across the junction of the two parts. When the temperature of the contact surfaces has reached the plastic range, pressure is applied, and upsetting of the plastic metal to a greater or less degree accompanies the making of the weld.

**Electric Resistance Flash Welding** is a modification of resistance butt welding and is carried out in much the same manner. Contact edges are slightly separated, however, and arcing or flashing occurs from edge to edge of joint. This arcing may burn or flash off a considerable amount of metal at the joint before the ends become sufficiently hot to form the weld by pressure. During the flashing period this loss in metal is compensated by feeding the two edges together to provide a uniform distance between them. Pressure is applied after flashing has heated up the contact surfaces to welding temperature. Flash welding is applicable to the welding of larger joint areas than resistance butt welding. This method also requires less care in the preparation of joint surfaces and smaller current densities for satisfactory welding than does resistance butt welding. A hydrogen atmosphere often is maintained at the weld to prevent oxidation of the joint surfaces.

Both butt and flash resistance welding processes are widely used for joining ends of wires, bars and tubes. They have been more recently developed for producing tubing from flat strips by a continuous longitudinal butt weld as the formed strip passes through electrical contact rolls. In such tube welding processes very high currents are utilized, the weld being effected at point of contact in a fraction of a second.

After completion of the weld, it may be heat treated in the resistance welding machine, using the current to bring the portion between the grips or electrodes to the desired heat treating temperature.

**Resistance Spot Welding** is applied to the welding of overlapped metal sheets or other comparatively thin sections. The overlapped parts are held between electrodes, which apply pressure while electric current is maintained through them, and the current cut off before the pressure is released. The area welded is approximately the same size as the electrode tip, usually $^1/_8$ to $^1/_4$ in. diameter. Spot welding is particularly applicable to fabrication of metal sheets up to $^1/_4$ in. thick. Automatic equipment makes possible the production of many welds per minute.

**Shot Welding** is spot welding, in which an extremely high welding current is applied for a short period of time. This method is used, and is particularly valuable, for the welding of certain alloy steels, as nickel-chromium steels, where time at welding temperature is harmful to the metal. In spite of the large welding currents employed, the total heat required is relatively small because of the small time interval of current application.

**Resistance Seam Welding** may be considered as a series of overlapping spot welds, the electrodes generally being in the form of rolls which also transmit pressure as the overlapping sheets pass between them. Automatic equipment for this type of welding generally utilizes a current interrupter which so times the current impulses with relation to the speed of travel of the sheets that a series of overlapping spot welds actually results.

**Percussive Welding Processes** are a form of resistance welding in which electrical energy is discharged suddenly at the junction, accompanied by the application of pressure. Closing of the discharge circuit often accompanies the bringing of the two parts into contact with pressure. The energy discharge may be obtained from a condenser, or by the transformation, by the collapse of the field, of the energy stored in a magnetic field.

**THERMIT WELDING.**—In the Thermit welding process, superheated steel is obtained by the exothermic reaction which results when a mixture of iron oxide and aluminum is ignited. The iron oxide is reduced by the aluminum to iron. The resulting superheated steel is poured into a mold around the joint to be made, mold and joint having been preheated. The superheated steel resulting from the iron oxide-aluminum reaction has a temperature far in excess of the melting point of steel, and hence there is sufficient heat to produce fusion with the surfaces of the parts to be joined. The composition of the metal may be varied by the introduction of alloying elements. The process has been applied to all types of wrought or cast steel and iron. It is particularly applicable to repair of heavy sections and for making butt welds in pipe and rails.

**Thermit Pressure Welding.**—A modification of the Thermit process is the Thermit pressure method of butt welding. This is applicable to joining such parts as rails, tubes or bars which may be conveniently pressed together by end pressure. The surfaces to be joined are butted together and the Thermit slag and metal poured into the mold around the joint. The alumina slag produced in the Thermit reaction is poured into the mold first and covers the surfaces of the parts. The superheated metal does not come into direct contact with the parts, but heats them to the plastic temperature range. When the ends are heated, as judged by a suitable time interval, pressure is applied and produces what is essentially a forge weld.

**GAS WELDING.**—In this fusion welding process the welding heat is obtained from a gas flame, oxygen and acetylene being generally the gases used. Fuel gases other than acetylene may be used in combination with oxygen. With the oxygen-acetylene combination a flame temperature of 5,000 to 6,000° F. is obtained. By varying the proportions of oxygen and acetylene, a neutral, reducing, or oxidizing atmosphere may be obtained. For most purposes a neutral or slightly reducing atmosphere is used. The process is applied successfully to the welding of practically all metals. It is of particular advantage in making welds of any shape or position on comparatively thin sections, especially at sharp edges and corners. It is not, however, limited to thin sections, but may be satisfactorily used to produce welds in heavy metal parts. For joining thin parts, a filler metal may or may not be required, but for thicker sections a filler rod of suitable composition must be used. The reducing flame protects the molten metal from oxidation, but in the welding of some materials a flux is necessary to help remove oxides.

**ARC WELDING.**—In arc welding the welding heat is obtained from an electric arc formed between the base metal and an electrode, or between two electrodes. Welding current may be either alternating or direct. Depending on the character of the electrode, arc welding is designated as either *carbon arc welding* or *metal arc welding*. In both processes the heat is highly localized and the temperature of the molten metal is considerably higher than in gas welding. Arc voltages will vary from 15 to 45 volts and the arc current from 20 to 600 amperes, although some processes use current values as high as 2000 amperes. The magnitude of the current which can be successfully used depends on the size of the electrodes, thickness of parts being welded, and rate of weld travel.

In arc welding a molten pool forms on the work at the arc location. By manipulation of the electrode the molten pool is made to travel along the joint as desired. Additional metal is introduced by a filler rod with the carbon arc process. In the metal arc process, the molten metal is supplied by melting from the tip of the electrode. Metal arc welding is more readily adaptable to a wider variety of applications than carbon arc welding.

**The Carbon Arc Method** always is used with direct current, and for most purposes with the electrode negative in order to avoid excessive consumption of electrode and absorption of carbon in the weld metal. The procedure is essentially a puddling one, and is not adapted to welding in the vertical or overhead positions. Its best application is in automatic welding in the flat position, in which it is general practice to shield the arc and molten metal from the atmosphere by introducing a combustible material and fluxing agents into the arc. The method has had wide application for high-speed welding of thin sections in the fabrication of steel barrels, tanks and light tubular products. It also has been used for welding heavy sections in pipe and pressure vessels.

**The Metal Arc Method** carries metal across the arc from the electrode to the work in very small globules and deposits them in the molten pool on the work. The electrode is consumed and the metal deposited at a rate dependent on the current used. By proper choice of electrodes, either direct or alternating current may be used. The procedure is adaptable for welding of sections 1/16 in. thick or more, and may be carried out in flat, vertical, or overhead position. Best quality welding is obtained by protecting metal in the arc from oxygen and nitrogen of the atmosphere by covering the electrodes with suitable fluxing materials and combustible agents. These exclude oxygen and nitrogen from the arc and give a slag covering to the pool of molten metal. A wide variety of electrodes are available for particular applications. The method is the most versatile and most widely used of the various fusion processes.

**ARC WELDING EQUIPMENT.**—Either alternating or direct current may be used for arc welding, depending on the type of electrodes. Carbon electrodes, bare wire, and covered electrodes, with covering principally of combustible materials, generally require direct current. Covered electrodes, with covering principally of minerals are, in general, satisfactory for direct or alternating current. Alternating current has the advantage of eliminating arc disturbances created by magnetic fields, and is conveniently supplied from power lines through transformers.

**Motor-Generator Sets.**—The most satisfactory source of direct current is a motor-generator set. The most common type is a portable unit, supplying one arc. The generator is of the variable voltage type which adjusts its output voltage to the arc demands. A less common unit is a larger set, suitable for supplying several arcs. This type usually has a constant voltage type generator to supply 70 to 100 volts. The voltage for individual arcs is adjusted by resistances in each line.

Alternating current is supplied from single-phase transformers, either with variable voltage taps or with a variable inductance in each arc circuit to obtain desired current values. To obtain the necessary constant-current characteristics, inductance always is employed, although sometimes it is inherent in the transformer design. The magnitude of the constant current may be adjusted by varying the inductance.

**The Amount of Current Required** for each size of electrode will vary considerably with the type of coating and the mass of metal being welded. Approximate values are listed in Table 1. These values are for usual manual or automatic welding procedures. Some special methods use currents as high as 1500 to 2000 amperes with metal electrodes of 1/2 in. diameter or larger. One such method uses a bare electrode and keeps the arc completely buried in powdered flux.

**Electrodes.**—All important metal arc welds now are made with covered electrodes, because of the greater strength and ductility of the weld metal as compared to that obtained from bare wire. A large variety of materials is used for covering electrodes, but coverings may be grouped as those consisting principally of combustible material, those consisting principally of mineral substances, and those using a combination of

**Table 1.—Approximate Current for Various Electrodes for Welding Steel**

| Electrode Diameter, in. | Current Required, Amperes | | |
|---|---|---|---|
| | Carbon | Bare Wire | Heavy Coated Electrodes |
| 3/32 | ....... | 50– 80 | 60–100 |
| 1/8 | ....... | 75–125 | 80–150 |
| 5/32 | 40 | 100–175 | 125–200 |
| 3/16 | 75 | 150–225 | 150–250 |
| 1/4 | 100–200 | 200–300 | 225–400 |
| 5/16 | 200–400 | ....... | 275–500 |
| 3/8 | 300–500 | ....... | 350–700 |
| 1/2 | 400–700 | ....... | ....... |

combustible and mineral materials. The combustible type gives protection to the arc largely by producing a gaseous envelope about the arc which excludes oxygen and nitrogen. The mineral-coated types give protection by forming a sheath of molten slag about the arc and a heavy slag layer over the weld metal pool. The action of this slag may be considered similar to that of slag in steel-melting furnaces. The coverings of electrodes also aid in stabilizing the arc, and, to a very large degree, determine the welding characteristics of the electrode. Electrode coverings also may be used to some extent for introducing alloying elements into the weld metal.

**ATOMIC HYDROGEN WELDING** is actually a combination of both arc welding and gas welding. A stream of hydrogen is made to pass through an electric arc established between two tungsten electrodes. The heat of the arc causes disassociation of the molecular hydrogen to the atomic form. After passing through the arc the atomic hydrogen recombines into the molecular form, liberating the heat obtained originally from the arc. The atomic hydrogen flame is used in fundamentally the same manner as the oxy-acetylene flame. This process is of particular advantage for welding along edges and corners of thin sections. It is used for welding stainless steels. The hydrogen flame gives excellent protection to the readily oxidizable chromium in these steels. Filler metal of suitable composition may be melted in the atomic hydrogen flame as required.

## 2. PHYSICAL PROPERTIES OF FUSION WELDS IN STEEL

**CONDITIONS FOR SOUND WELDS.**—The physical properties of fusion welds depend, of course, on the relative freedom of the weld metal from defects such as porosity, entrapped slag, cracks, incomplete fusion, etc. Assuming that a sound weld has been obtained, physical properties will depend on the degree of protection which has been given to the molten metal from the gases of the atmosphere. Avoidance of contamination by nitrogen and oxygen is especially important in metal arc welding. Welds deposited with bare wire electrodes with no arc protection will contain approximately 0.10 to 0.15% N, and approximately 0.3% O. With a satisfactorily designed covered electrode, the deposited weld metal will contain less than 0.03% N, with oxygen correspondingly low. The effect of such contamination on weld metal physical properties may be best indicated by the following typical properties of metal arc welds in plain carbon steels using bare electrodes and covered electrodes:

| | Bare Electrodes | Covered Electrodes |
|---|---|---|
| Yield point, lb. per sq. in. | 35–40,000 | 45–55,000 |
| Ultimate tensile strength, lb. per sq. in. | 40–55,000 | 60–75,000 |
| Elongation in 2 in., percent | 0–10 | 20–38 |
| Endurance limit, lb. per sq. in. | 15–20,000 | 28–31,000 |
| Density | 7.50–7.70 | 7.80–7.85 |
| Charpy impact, ft-lb. | 3–8 | 20–45 |

The tensile strength of steel welds may be considerably increased over the above values by addition of suitable alloying elements, but in all cases, for a given composition, superior properties will be obtained in metal arc welding by using covered electrodes.

**SOUNDNESS OF FUSION WELDS.**—Producing welds by the gas or arc fusion processes introduces many variables of technique and operator skill which may cause mechanical defects in the weld. These may be gas pockets, entrapped slag from the electrode covering, zones of incomplete fusion between weld metal and base material, or cracks resulting from contraction stresses.

Several methods have been utilized for inspection of welds to detect such defects. The common medical stethoscope has been used to a limited extent, by detecting variation in sound issued at positions along the welds when struck by a hammer. This method is not very sensitive and is limited to welds up to about 1 in. thick. Magnetic methods have been used quite successfully in magnetic materials, to detect cracks at or near the surface of welds. They have not proved adaptable to detect defects far below the surface. The simplest of the magnetic methods consists of establishing a magnetic field approximately perpendicular to suspected defects and dusting a magnetic powder over the surface. The powder tends to concentrate at locations of fissures or cracks. The magnetic method may be used to advantage in some cases to supplement radiographic methods for the detection of small surface cracks which can not be disclosed readily by the x-ray method. The magnetic method also may be used to advantage for examining welds which can not for one reason or another be x-rayed.

The most thorough and reliable method of inspection is x-ray examination of the complete weld. The x-ray tube is placed on one side of the weld and photographic film on the opposite side. Excellent quality radiographs can be obtained which register the presence of all types of defects and give a clear picture of their size, location and distribu-

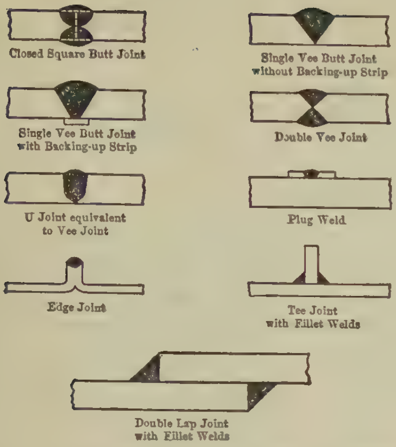

Closed Square Butt Joint
Single Vee Butt Joint without Backing-up Strip
Single Vee Butt Joint with Backing-up Strip
Double Vee Joint
U Joint equivalent to Vee Joint
Plug Weld
Edge Joint
Tee Joint with Fillet Welds
Double Lap Joint with Fillet Welds

FIG. 2. Forms of Joints for Fusion Welding

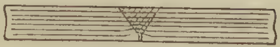
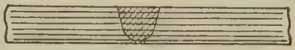
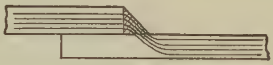


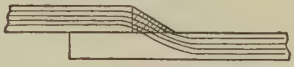
FIG. 3. Undesirable Stress Concentration with Incomplete Fusion



A.- Uniform Stress Distribution with Butt Joint



B.- Non-uniform Stress Distribution with 45° Fillet Weld



C.- More Uniform Stress Distribution with 30°- 60° Fillet Weld

FIG. 4. Stress Distribution in Various Types of Welds

tion. This method may be used for examination of steel sections 4 1/2 to 5 in. thick, and will detect defective areas which are in thickness equal to 2% or more of the steel section. Gamma rays arising from radio-active materials may be used in the same manner as x-rays. They are particularly applicable to heavy sections and to use in locations where x-ray equipment can not be placed. Use of both x-rays and gamma rays require suitable precautions to avoid excessive exposure of personnel to the radiation. See X-ray Protection (U. S. Bureau of Standards Handbook HB20) for recommended practice.

**STRESSES IN WELDS.**—The thermal contraction of fusion welds, and the expansion and contraction of metal parts from the heat of welding, produces stresses of high magnitude in welded structures. The effects of these stresses are to distort the structure during welding and to create residual internal or *locked-up* stresses in the completed welds. By control of procedure, the degree of distortion, and the magnitude of the stresses may be considerably reduced. In thick sections and rigid structures, however, the completed welds invariably have residual stresses equal to the yield point of the material. It is not uncommon for these stresses to act in more than one axial direction, and to be of such magnitude as to cause fracture of the weld without any external loading. During welding, assembly of parts so as to allow for deformation due to contraction, and peening of the layers of weld metal will reduce the stresses. Their removal, however, is best accomplished by uniform heating, followed by slow cooling. For steel welds, heating the entire structure to 1100 to 1200° F. for one hour per inch of thickness is recommended. If it is possible to heat only part of the structure at a time, the temperature gradient between heated and unheated portions should not exceed 200° F. per foot. This stress relieving practice may be expected to reduce residual welding stresses to the approximate yield point value of the steel at the stress-relieving temperature, *i.e.*, 3000 to 8000 lb. per sq. in. for mild steels.

## 3. WELDING DESIGN

**ADVANTAGES OF WELDED CONSTRUCTION.**—The design and assembly of a structure with welded connections offer a number of advantages as compared to mechanical joints, such as riveting. The connections are simplified in that the joint generally is made direct, and no third member, as a butt-strap, is required. Welded joints permit placing the added metal exactly where needed, and produce an integral structure which, with mild steel, can develop 100% joint efficiency on any thickness of section. Elimination of extra material results in a weight saving of up to 25%. This saving in weight is an important consideration in transportation fields.

**FORMS OF WELDED JOINTS.**—Many forms of joints are used in fusion welding processes. Some of the more widely used are shown in Fig. 2. Drawings of welded construction should contain full information as to weld details, type of joint, dimensions of welds, etc. The American Welding Society has issued welding symbols which may be conveniently used on drawings specifying welding. See Figs. 5–8. In designing a welded structure, consideration also should be given to the following points:

**1.** The joint should be made as accessible as possible to the welding operator. This means that wherever possible welding should be done in the downhand position rather than taxing the skill of the operator by welding in the overhead position.

**2.** The dimensions of the joint should be such as to permit complete fusion through the thickness of the sections being welded. If incomplete fusion is obtained an undesirable stress concentration will be present. See Fig. 3.

**3.** Butt joints are preferable from the viewpoint of uniform stress distribution across the weld. In the case of fillet welds a 30 deg–60 deg. weld provides a more uniform transfer of stress through the weld than a 45 deg.–45 deg. weld. See Fig. 8.

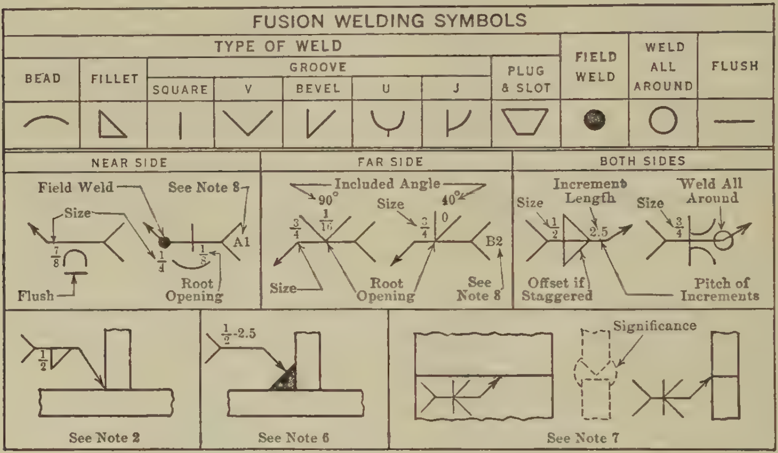


Fig. 5. Fusion Welding Symbols

1. In plan or elevation, near, far and both sides locations refer to nearest member parallel to plane of drawing and not to others farther behind.
2. In section or end views only, when weld is not drawn, the side to which arrow points is considered near side.
3. Weld symbols on both sides are of same size unless otherwise shown.
4. Symbols govern to break in continuity of structure or to extent of hatching or dimension lines.
5. All welds are continuous and of user's standard proportions, and all except V- and bevel-grooved welds are closed unless otherwise shown.
6. When welds are drawn in section or end views, obvious information is not given by symbols.
7. In joints in which one member only is to be grooved arrows point to that member.
8. Tail of arrow used for specification reference.

Note: All dimensions are in inches.

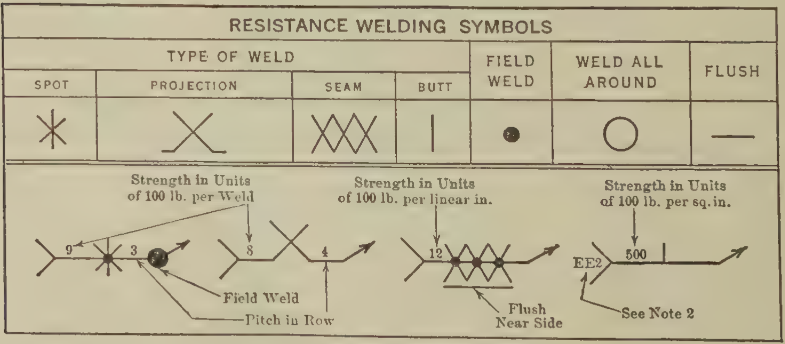

RESISTANCE WELDING SYMBOLS
TYPE OF WELD
SPOT
PROJECTION
SEAM
BUTT
FIELD WELD
WELD ALL AROUND
FLUSH
Strength in Units of 100 lb. per Weld
9
3
8
4
Field Weld
Pitch in Row
Strength in Units of 100 lb. per linear in.
12
Flush Near Side
Strength in Units of 100 lb. per sq. in.
EE2
500
See Note 2

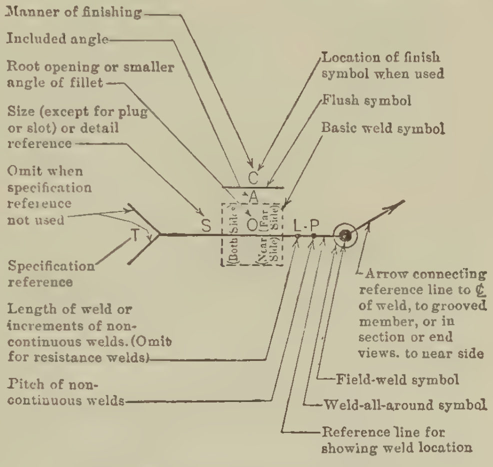
Fig. 6. Resistance Welding Symbols

1. Symbols govern to break in continuity of structure or to extent of dimension lines. 2. Tail of arrow used for specification reference. Note: All dimensions are in inches.

**4.** Where the structure previously has been a casting or built by riveting, and where the structure is being redesigned for welded construction, it should be appreciated that welding cannot be used interchangeably with the older riveted or casting design, and the designer should adapt his structure as far as possible to utilize the full advantages of welded construction.

**5.** Where distortion of the welded structure by reason of the contraction of the weld metal is liable to occur, and where the distortion is objectionable, the welds should, if possible, be so positioned that the contraction of the welds will tend to balance each other on each side of some central plane through the structure.

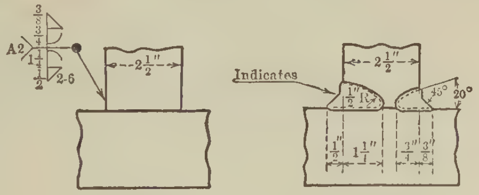
The design stresses will, of course, vary, depending on the quality of the welds and the conditions of service of the structure. For example, pressure vessels may be built on the basis of factors of safety of 4 or 5, corresponding to an allowable working stress of about 10,000 to 12,000 lb. per sq. in. (55,000 lb. per sq. in. tensile strength steel), while an open storage tank could reasonably employ much higher stresses. The allowable working stresses for welds of various types have been set by engineering codes governing different types of construction. A brief résumé of each code, including permissible working stresses for different types and classes of welded construction, is given below:


Manner of finishing
Included angle
Root opening or smaller angle of fillet
Size (except for plug or slot) or detail reference
Omit when specification reference not used
Specification reference
Length of weld or increments of non-continuous welds. (Omit for resistance welds)
Pitch of non-continuous welds
Location of finish symbol when used
Flush symbol
Basic weld symbol
Arrow connecting reference line to ℄ of weld, to grooved member, or in section or end views. to near side
Field-weld symbol
Weld-all-around symbol
Reference line for showing weld location
C
A
S
T
O
L-P
(Both Sides)
(Far Side)
(Near Side)

Note:- Relative position of data and symbols shown; near, far and both sides significance as given in instructions.

Fig. 7. Standard Location of Information on Welding Symbols

**A.S.M.E. Boiler Construction Code—Unfired Pressure Vessel Section.**—Rules for the fusion process of welding in this section are so drawn as to provide for three different classes of vessels as follows:



Sketch 1

Sketch 2

Fig. 8. Comparison between Symbolic and Verbal Methods of Welding Information

Interpretation of symbol: Double-fillet-welded, partially grooved, double-J, tee-joint with incomplete penetration. (Type of joint shown by drawing.) Grooves of standard proportions (which may be $1/2$ in. rad., 20° included angle, root edges in contact before welding) $3/4$ in. deep on far side and 1 $1/4$ in. deep on near side. $3/8$ in. continuous fillet weld on far side and $1/2$ in. intermittent fillet weld on near side with increments 2 in. long, spaced 6 in. center-to-center. All fillets standard 45° fillets. All welding done in field in accordance with welding specification number A2 (which may require, for example, that weld be made by manual D.C. shielded metal-arc process using high-grade, covered mild steel electrode; that root be unchipped and welds unpeened but that joint be preheated before welding).

I.—Paragraph U–68. Vessels may be used for any purpose when constructed in accordance with requirements, which are identical with those of the Power Boiler Section.

II.—Paragraph U–69. Vessels constructed in accordance with the rules of this paragraph may be used for any purpose except for containing lethal gases or liquids and/or liquids operating at a temperature in excess of 300° F., providing the plate thickness does not exceed 1 $1/2$ in., and the maximum pressure does not exceed 400 lb. per sq. in., nor at a temperature in excess of 700° F. This pressure limitation does not apply to vessels operated under hydraulic pressure at atmospheric temperature.

Requirements are as follows: 1. Main joints of double-welded butt type; 2. Vessels to be stress relieved when both the wall thickness is greater than 0.58 in. and the shell diameter less than 20 in., and for other wall thicknesses and shell diameters where the diameter is less than 120 $t - 50$, where $t$ = thickness, in.; 3. Qualification tests of welding operators at intervals of 6 or 12 months: *a*, tensile strength of weld not less than minimum of specified tensile range of plate for full section tension specimen, and not less than 95%

for reduced section tension specimen; *b*, bend test, ductility across weld, 20% minimum. 4. Factor of safety, 5; 5. Joint efficiency, 80%. 6. Allowable working stress as follows:

| | | |
|---|---|---|
| Minimum Tensile Strength of Plate | 55,000 lb. per sq. in. | 70,000 lb. per sq. in. |
| Allowable working stress, lb. per sq. in. | 8,800 | 11,200 |

III.—Paragraph U–70. Vessels constructed in accordance with the rules of this paragraph may be used for the storage of gases or liquids, except lethal gases or liquids, at temperatures not materially exceeding their boiling temperature at atmospheric pressure, and at pressures not to exceed 200 lb. per sq. in. and/or not to exceed 250° F.

Requirements are as follows: 1. Plate thickness not over $^5/_8$ in.; 2. Qualification tests of welding operators at intervals of 6 to 12 months; *a*, tensile strength of weld not less than 90% of minimum of specified tensile range of plate used for full section tension specimen and not less than 85% for reduced section tension specimen; *b*, bend test ductility across weld, 10% minimum.

Allowable working stresses are as follows:

| | | |
|---|---|---|
| Double-welded butt joints for all joints | 8,000 | lb. per sq. in. |
| Single-welded butt joints for girth or head joints | 6,500 | " |
| Double full-fillet lap joints for girth joints only | 7,000 | " |
| Plug or intermittent welds for girth or head joints | 5,600 | " |
| Single-welded butt joints and double full-fillet welds for longitudinal joints: | | |
| *a*, For thickness of less than $^1/_4$ in | 5,600 | " |
| *b*, " " " $^1/_4$ to $^3/_8$ in | 7,000 | " |

**A.P.I.–A.S.M.E. Unfired Pressure Vessel Code.**—This code covers the construction of vessels for use with petroleum liquids and gases. Stress relieving is required for plate thicknesses above 1 $^1/_4$ in., or when radiographing is not mandatory. Factors permitting various joint efficiencies are employed, these factors varying with type of steel plate, type of joints, and whether or not radiographing of the welded joints and stress relieving of the vessel has been employed, as follows:

| Steel: | Specification | Factor |
|---|---|---|
| Group A | A.S.T.M. A70, firebox grade only | 1.00 |
| | " A30, " " " | 1.00 |
| | " A149, " " " | 1.00 |
| | " A150, " " " | 1.00 |
| Group B | " A70, flange " " | 0.97 |
| | " A30, " " " | 0.97 |
| | " A149, " " " | 0.97 |
| Group C | " A7<br>" A10<br>" A113<br>Not to be used for plates thicker than $^5/_8$ in. or for metal temperatures above 450° F | 0.92 |
| Radiographing: | | |
| | Radiographed | 1.12 |
| | Not radiographed | 1.00 |
| Stress Relieving: | | |
| | Stress relieved | 1.06 |
| | Not stress relieved | 1.00 |
| Weld Joints: | | |
| | Double-welded butt joint and single welded butt joint with backing-up strip | 0.80 |
| | Single-welded butt joint without backing-up strip | .70 |
| | Double-full-fillet lap joint and single-full-fillet lap joint with plug welds | .65 |
| | Single-full-fillet lap joint | .55 |

For service temperatures below 650° F. the allowable working stress in tension and shear for carbon steel in this code is 25% of the ultimate strength in tension at atmospheric temperature. The allowable working stress of a welded joint, therefore, is

$$^1/_4\, S \times f_q \times f_r \times f_s \times f_w$$

where $S$ = ultimate tensile strength of steel, and $f_q$, $f_r$, $f_s$ and $f_w$ = respectively, the factors for plate quality, radiographing, stress relieving and weld joint. Thus welded joints on plate of 55,000 lb. per sq. in. minimum tensile strength of Group A, of the double-welded butt type, radiographed and stress relieved, would have an allowable working stress of

$$^1/_4\,(55{,}000) \times 1.00 \times 1.12 \times 1.06 \times 0.80 = 13{,}060 \text{ lb. per sq. in.}$$

Table 2 gives the probable combinations of construction factors and joint efficiency.

**Table 2.—Joint Efficiency of Various Types of Construction**

| Type of Joint | Joint Efficiency, percent | Radiograph Factor (1.12) | Stress Relief Factor (1.06) | Final Joint Efficiency for Groups of Steel, percent | | |
|---|---|---|---|---|---|---|
| | | | | A, Factor = 1.00 | B, Factor = 0.97 | C, Factor = 0.92 |
| Double-welded butt joint and single-welded butt joint with backing-up strip. | 80 | .......... | .......... | 80 | 78 | 74 |
| | | .......... | * | 85 | 82 | 78 |
| | | * | .......... | 90 | 87 | .......... |
| | | * | * | 95 | 92 | .......... |
| Single-welded butt joint without backing-up strip. | 70 | .......... | .......... | 70 | 68 | 64 |
| | | .......... | * | 74 | 72 | 68 |
| | | * | .......... | 78 | 76 | .......... |
| | | * | * | 83 | 81 | .......... |
| Double-full-fillet lap joint and single-full-fillet lap joint with plug welds. | 65 | .......... | .......... | 65 | 63 | 60 |
| | | .......... | * | 69 | 67 | 63 |
| Single-full-fillet lap joint. | 55 | .......... | .......... | 55 | 53 | 51 |
| | | .......... | * | 58 | 57 | 54 |

Note.—The asterisks in the columns Radiograph Factor and Stress Relief Factor indicate that these operations have been performed. For example, in Line 4, the factor under Column A = 80 × 1.12 × 1.06 = 95, and under Column B = 95 × 0.97 = 92.

**A.S.M.E. Boiler Construction Code—Power Boiler Section.**—The use of forge welding, electric resistance butt welding, or pressure Thermit welding, has been sanctioned for many years. The ultimate strength of the joint is considered as 35,000 lb. per sq. in. Allowable working stress on factor of safety of 5 is 7000 lb. per sq. in.

The rules for the fusion process of welding, adopted in 1931, apply only to drums and shells. The chief requirements are as follows: 1. Main joints are of double-welded butt type. 2. Main joints are radiographed. 3. Completed welded drum is stress relieved. 4. Test plate welded with drum: *a*, tensile strength of transverse specimen and all-weld-metal specimen to be at least that of minimum of the range of the plate; *b*, elongation of all-weld-metal tension specimen, 20% in 2 in. minimum; *c*, elongation of outer fibers of transverse bend test, 30% minimum across weld; 5. Carbon content of plate material, 0.35% maximum. 6. Factor of safety, 5. 7. Joint efficiency, 90%. 8. Allowable working stress of plate as follows:

| | | |
|---|---|---|
| Minimum Tensile Strength of Plate....... | 55,000 lb. per sq. in.* | 70,000 lb. per sq. in.* |
| Allowable working stress, lb. per sq. in., at temperatures up to 700° F.......... | 9,900 | 12,600 |

**Bureau of Marine Inspection and Navigation, U. S. Dept. of Commerce.**—The welding rules of this code, covering the construction of marine boiler drums and unfired pressure vessels, are similar to the requirements of the A.S.M.E. Boiler Construction Code.

**Bureau of Engineering, U. S. Navy Dept.**—These welding specifications establish for purposes of requirements as to design and tests of welded assemblies a classification of welded structures as follows:

- Class A—Pressure vessel welds.
  - Class A1—Fired pressure vessels such as boiler drums.
  - Class A2—Unfired pressure vessels.
  - Class A3— " " "
  - Class A4— " " "
- Class B—Structural welds.
  - Class B1—Moving machinery parts.
  - Class B2—Stationary machinery parts.
  - Class B3—Non-pressure vessels.
  - Class B4—Tack welding.

A joint efficiency of 90% is permitted on welds which have been radiographed. Otherwise the joint efficiency is taken as 80%.

**A.S.A. Code for Pressure Piping.**—This code covers construction requirements for various kinds of piping including piping for steam, gas, air and oil. Welding requirements involve qualification of welding operators. Stress relieving of pipe joints is mandatory when the pipe wall thickness is greater than 3/4 in., and the service temperature is in excess of 250° F. See Kent's Mechanical Engineers' Handbook—Power, p. 5–23, forming Vol. 2, of this series.

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* Intermediate tensile strengths in proportion.

**National Board, Rules for Repair of Boilers by Fusion Welding.**—Rules for the repair of boilers by fusion welding have been issued by the National Bureau of Casualty and Surety Underwriters in co-operation with the National Board of Boiler and Pressure Vessel Inspectors. Copies of these rules may be obtained upon application to the National Bureau of Casualty and Surety Underwriters, 1 Park Avenue, New York City.

A number of other codes exist covering various types of welded construction. In general, all codes require qualification of welding operators, and most of them embody filler metal specifications in accordance with the requirements of the American Welding Society specification, issued August, 1933.

Some of the constructional codes issued by the American Welding Society are as follows: Code for Fusion Welding and Gas Cutting in Building Construction (*Jour.*, Am. Welding Soc., March, 1935). Specification for the Design, Construction, Alteration and Repair of Railway and Highway Bridges. Tentative Code for Fusion Welding and Flame Cutting in Machinery Construction. Marine Code for Welding and Gas Cutting, Part D, Rules for the Fusion Welding of Hulls and Hull Parts (*Jour.*, Am. Welding Soc., Feb., 1935).

## 4. WELDING OF VARIOUS MATERIALS

**STEELS.—General.**—The difficulty which may be encountered in the welding of various steels is, in general, in proportion to the hardening developed in the heat-affected zone along the weld. With high hardness, fine cracks or even complete fracture may result from welding stresses. As the carbon content of steel increases, the potential hardness increases, and the addition of alloying elements in general emphasizes the hardening effect.

Excessive hardening can be somewhat controlled by regulating the cooling rate from the welding temperature. With Thermit welding and Thermit pressure welding, a slow cooling rate can be obtained by the insulating effect of the mold surrounding the entire weld. With gas welding, particularly on thin sections, the torch can be used to heat the base materials and keep the zone adjacent to the weld sufficiently hot to retard cooling. With arc welding, the area of high temperature is very localized, and preheating of the parts sometimes necessary to obtain slow cooling and to reduce welding stresses. With a composition giving high hardness, more difficulty is likely to occur with heavy sections, and whatever welding method is used preheating will be necessary. Heating to 400 to 500° F. is effective and, depending on the steel and the particular structure, this temperature may need to be maintained until the weld has been stress relieved.

Variation in the quality of steel may result in welding difficulties even for grades normally considered weldable, such as plain carbon steels below 0.30% C. Fortunately, such difficulties are exceptional. No definite information is available as to the reasons for such difficulties, although it is generally conceded that trouble may occur because of excessive segregation in the steel, or because of the presence of abnormally large amounts of residual elements such as arsenic, copper, etc. Laminations in rolled steel products will, of course, produce welding difficulties, especially if the lamination extends along the joint edge.

**CASTINGS.**—Castings may be satisfactorily welded, provided rolled or forged steel parts of the same composition can be satisfactorily welded. This assumes that the casting adjacent to the weld region is sound and free from shrinkage cracks which otherwise would open under the contraction stresses during cooling from welding.

**IRON AND CARBON STEELS UP TO 0.35 PERCENT CARBON.**—Iron and plain carbon steels up to 0.35% C. are readily welded by all processes, and completed welds can be obtained with strength equivalent to the base materials. Preheating is not necessary except for extremely rigid joints where welding stresses become exceptionally high. In fusion welding, with gas or electric arc, filler metal of approximately 0.15% C. is recommended. With shielded arc methods (carbon or metal arc), ductile welds with strength of 60,000 to 70,000 lb. per sq. in. will be obtained. This strength can be increased to 70,000 to 80,000 lb. per sq. in., without loss of ductility, by small additions of alloying elements. Molybdenum in amounts of 0.2 to 0.4% is very effective.

Where stress relieving is required, heating at 1100 to 1200° F. for a period of one hour per inch of thickness is recommended.

**MEDIUM CARBON STEELS OF 0.35 TO 0.60 PERCENT CARBON.**—As the carbon content of steel increases above 0.35%, difficulty in welding is likely to develop. With thin sections and a flexible design, gas or arc welding may be used without preheating, but in joints over $^3/_8$ to $^1/_2$ in. thick, preheating is necessary. In oxygen-acetylene gas welding, better welds are obtained with a flame having excess acetylene. To develop equivalent strength in the weld, a low-alloy filler metal should be used for both gas and arc welding methods. A rod containing chromium and molybdenum is recommended. These steels are readily welded by the Thermit process.

Welds should be stress relieved at 1100 to 1200° F. and, where preheating has been necessary or with the higher carbon ranges, this stress relieving should be carried out before permitting the weld to cool below 300° F.

**HIGH CARBON STEELS OF 0.60 TO 1.50 PERCENT CARBON.**—Steels above 0.60 percent carbon may be welded quite satisfactorily by the Thermit process, and this method has been used considerably for butt welds in rails. Gas and arc welding can be done only with considerable difficulty, and the resulting welds are only fairly satisfactory. Gas welding is more easily performed. A joint can be made by arc welding by depositing low-carbon metal over the surface of each part and then making the arc weld between these low carbon surfaces. The resulting weld will, of course, have the properties of a low carbon weld.

**LOW-ALLOY HIGH-YIELD-STRENGTH STRUCTURAL STEELS.**—There are a considerable number of low-alloy steels especially designed to produce yield point values of 50,000 to 70,000 lb. per sq. in. and with ultimate strengths of 70,000 to 100,000 lb. per sq. in. Most of these steels have been produced with a low carbon content and no particular difficulty is encountered in welding them. In general, where carbon contents are 0.20% or less, gas or arc welding can be performed without preheating. With higher carbon contents, preheating will often be necessary. To develop weld strengths equivalent to the base material, it is necessary to use filler metal with some alloy content. Low carbon filler metal with alloy composition similar to the base materials usually can be used. Stress relieving should be carried out at 1100 to 1200° F.

**S.A.E. ALLOY STEELS.**—For welding of the S.A.E. alloy steels, the same general procedure may be used as for the alloy structural steels. The ease of welding will depend on the hardness developed in the heated zone. Where high hardness occurs, preheating to 300 to 500° F. will be necessary. Filler metal of similar alloy composition to the base material usually can be used.

**CHROMIUM STEELS OF 4 TO 6 PERCENT CHROMIUM** are definitely air hardening, even with low carbon content, and preheating at 300 to 500° F. is essential. Welds generally must be kept above 300° F. until annealed or stress relieved. Heat treatment after welding should be a full anneal or long time stress relieving at 1300 to 1350° F. The same procedure should be used with steels of 2 1/2% Cr. With such procedure, welds in these steels with carbon less than 0.20% are very ductile and equal to the base metal in every respect.

**CORROSION- AND HEAT-RESISTING STEELS** have high percentages of chromium or chromium and nickel, with low carbon content. Welding steels with 12 to 30% Cr requires preheating to above 200° F., and an annealing treatment after welding in order to obtain ductility. The lower chromium ranges, 12 to 16% Cr, may be welded without great difficulty if preheating precautions are taken. As the chromium content is increased, the welds are much less ductile, and the very high chromium range becomes very difficult to weld. Heat treatment should consist of annealing at 1450 to 1550° F. To maintain equivalent chromium content in the welds, filler metal should have a slightly higher chromium content than the base metal. Due to rapid carbon absorption in high-chromium steels, the carbon arc method is not suitable.

The stainless steels containing high percentages of both chromium and nickel are not subject to hardening from heat treatment and may be welded without difficulty. Typical steels of this group are: 18% Cr–8% Ni, 25% Cr–12% Ni, 25% Cr–20% Ni; there are many other variations of analysis. Welds in all of these materials are soft and ductile, and no heat treatment after welding is required, except as may be needed to improve corrosion resistance. In this respect the 18% Cr–8% Ni type is subject to serious loss of corrosion resistance, due to precipitation of carbides in the temperature range of 900 to 1500° F. For this reason, very low carbon contents (less than 0.07%) are desirable for these steels if they are to be welded. The carbides may be stabilized by addition of titanium or columbium, and steels so stabilized are recommended for welding. Filler metal also should be of the stabilized analysis. Titanium should be used in amounts of at least 6 times the carbon content, and columbium in amounts of at least 10 times the carbon content. If the 18% Cr–8% Ni steels are not of the stabilized type, heat treatment after welding should consist of heating to 1900 to 2100° F. followed by rapid cooling. In the higher alloy steels, carbide precipitation is not serious.

Because of higher electrical resistance and lower heat conductivity, electric welding of these steels requires about 10% lower current than carbon steels. Also, the high coefficient of expansion requires additional care to prevent excessive distortion.

**MANGANESE AUSTENITIC STEEL** with 12 to 14% Mn and approximately 1 to 1 1/2% C is quite brittle in the as-cast condition, because of large carbide areas, but upon heating to 1900° F. and quenching, the carbides are retained in solid solution and a tough ductile material results. On reheating to 1100 to 1400° F., carbides are again precipitated and the material becomes hard and brittle. In welding this steel it is necessary to deposit weld metal which is ductile and to avoid embrittlement of the base material by excessive heating.

Welding may be accomplished by either the gas or arc methods. A filler metal of 12 to 14% Mn and 3 to 5% Ni is recommended to obtain a good quality weld. Care should be taken to avoid excessive heating of the base metal.

**CAST IRON.**—Cast irons, either gray, white or malleable, are not considered adaptable to welding, but may be welded with proper care and procedure. White cast iron is the most easily welded. The process may be the gas or metal arc, using a white cast-iron filler rod. Resistance welding may be used where shape permits. Gray or malleable castings may be fusion welded with proper precautions, and the same general procedure can be applied for both types of castings. Whatever method is used, because of the low tensile strength and poor ductility of these materials, it is essential to avoid excessive variations in temperature throughout the casting to prevent cracking from thermal stresses.

**Gas Welding** is best performed by preheating the casting to approximately 1000° F. and completing the weld while the castings are hot. After welding, the casting must be cooled slowly. A suitable flux such as borax should be used, and the filler metal should be of a high-silicon cast-iron composition. Welds made in this manner will be machinable.

**Carbon Arc Welding** can be performed in a manner similar to that used for gas welding. Use of cast-iron filler rod and borax flux is recommended.

**Metallic Arc Welding** may be performed using either coated cast-iron composition electrodes or coated steel electrodes. Preheating is necessary in using the cast-iron electrodes, and this method will produce a machinable weld. The method is best adapted for welding in the flat position. Using coated steel electrodes, welds may be made without preheating the casting, but a hard unmachinable zone will be present. Small-diameter electrodes ($^1/_8$ in. maximum) should be used and welding should progress slowly with intermittent peening. This will keep thermal stresses in the casting, and contraction stresses in the weld metal, at a minimum. Welds made with steel electrodes will be considerably stronger than the cast-iron base metal. Welds not so strong, but very satisfactory and machinable, may be made, using an alloy electrode of high nickel composition such as Monel metal (68% Ni, 29% Cu). Similarly, welds may be made with copper or copper alloy filler metal.

## 5. SURFACING BY WELDING

For special applications, steels often are surfaced with another material in order to obtain resistance to wear or corrosion. These materials are conveniently welded to the steel base by gas or metal arc methods. For corrosion resistance, stainless steels or Monel metal commonly are used. For wear resistance or hard surface, deposits of plain high-carbon steels, alloy steels, high-speed steels, alloys of cobalt, chromium, and tungsten (Stellite), high-carbon austenitic steels such as Hadfield's manganese steel, and preparations of tungsten or boron carbides, are available. All of these materials may be applied by metal arc or oxygen-acetylene welding.

## 6. QUALIFICATION OF WELDERS

In all welding processes, but particularly in fusion welding processes such as gas welding and electric welding, employing manual welding, the quality of the welds for a given welding process depends largely on the skill of the welding operator. Weld quality depends less on the operator with automatic welding equipment. For manual welding by any of the processes, the welding operator should have received special training in making welds under conditions similar to those encountered on his production work. The skill of the welding operator should be checked from time to time. This generally is accomplished by what are known as qualification tests. The operator deposits a weld using the same process and under conditions similar to the production work which he is to undertake. Specimens from these qualification test plates are obtained and tested. Requirements for qualification tests of welding operators are quite varied as to detail. See Qualification of Welding Process and Testing of Welding Operators, *Jour.* Am. Welding Soc., 1936, Oct. Supplement.

## Section 3

# CORROSION AND CORROSION RESISTANT METALS

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## CORROSION

By Frank N. Speller

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## CORROSION RESISTANT METALS

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# CORROSION OF METALS

By Frank N. Speller

## 1. CORROSION IN GENERAL

Corrosion of metals may be defined broadly as the chemical action of their environment, often resulting in their deterioration or destruction. Under most ordinary conditions of exposure the corrosion products comprise mainly oxides (more or less hydrated), carbonates, and sulphides. At high temperatures the products consist mainly of oxides, though indications are that corrosion of this type also progresses more rapidly when some moisture is present. Corrosion may be entirely superficial or may attack the intercrystalline boundaries and cause disintegration in the metal, as dezincification of brass or so-called caustic embrittlement in boiler steel.

It may be taken as established that in nearly all cases of corrosion at ordinary temperatures, comprising those most important in practice, the driving force of the corrosion reaction between metal and environment is electrochemical.

**ESTABLISHED FACTS REGARDING CORROSION.**—*a.* In most cases both moisture and oxygen are necessary for corrosion. *b.* The initial rate of corrosion usually is comparatively rapid, slowing as a protective film forms. *c.* Surface films are important in controlling the rate and distribution of corrosion. *d.* Increased rate of motion increases corrosion in water up to a certain velocity; the rate then may drop where strong protective films are formed. *e.* Dissimilar metals in electrical contact accelerate corrosion of the ones that happen to be anodic. Galvanic action is a most active agent of corrosion. It occurs when two metals, one electronegative to the other, are placed in contact and exposed to moisture. *f.* The composition of ordinary iron and steel has little or no effect on their relative rates of corrosion under water or under ground. Under these conditions the particular kind of metal usually is not so important as environment. *g.* Variation in the concentration of a solution in contact with a metal tends to localize corrosion. Where this is due to oxygen in solution, the surface to which the oxygen diffuses most rapidly is cathodic to the area that is protected from diffusion of oxygen. The smaller the anodic areas in relation to cathodic areas the greater is the rate of penetration at anodic points. Corrosion, like other chemical reactions, increases with temperature.

The above, and other facts (See F. N. Speller, Corrosion Causes and Prevention, McGraw-Hill Book Co., 1926), are the basis of the electrochemical mechanism which now is generally accepted to explain corrosion in the presence of water.

## 2. MECHANISM OF CORROSION

**CORROSION OF IRON OR STEEL IN GENERAL.**—While there is a vast difference in the amount of corrosion of iron and steel under different conditions, and while the relative corrosion varies with surrounding conditions, certain fundamental causes underlie all cases of corrosion.

Every metal when placed in water is subjected to a fixed tendency to go into solution. This is wholly a matter of electrochemical activity and varies to a definite extent with each metal. The initial reaction in the process of corrosion is analogous to solution in acid. The acidity of the water determines the initial speed of attack. As in all electrochemical reactions, however, the initial speed of solution soon slows down as the electrochemical couples which form over the surface of the metal become "polarized" due to the accumulation of hydrogen on their cathodic surfaces, ultimately stopping the solution of the metal. In the case of iron a small amount of the metal is dissolved as ferrous hydrate, which also retards the rate of solution of the metal. This first stage of corrosion of iron, while essential, is not serious if the reaction between water and iron can be stopped at this point. In fact, it cannot proceed unless, in some manner, the hydrogen film protecting the metal is removed. This is usually accomplished by some depolarizing agent, generally free oxygen, which also combines with ferrous hydrate forming insoluble ferric hydrate, commonly known as rust.

This second stage of corrosion, is thus caused by the oxidation of hydrogen and ferrous hydrate in solution by free oxygen from the atmosphere. Oxygen is soluble in water at normal temperature to the extent of about ten parts by weight per million, and as water readily absorbs more oxygen, the reaction will continue until the metal has been destroyed, or the reaction may be stopped by exhaustion of the available supply of oxygen. The latter case is illustrated in any closed water system, where in a metallic water pipe the amount of free oxygen will decrease as the water travels from the inlet. If, then, the pipe is long enough the corrosion will finally be reduced to a negligible amount.

Iron, like all other metals, when in contact with water or a solution has a definite inherent tendency to go into solution in the form of electrically-charged particles (iron-ions.). The solution must remain electrically neutral, and these positive ions can enter it only if an equivalent number of positive ions of some other elements are displaced. Thus, iron immersed in a solution of copper sulphate begins to go into solution, but simultaneously, an equivalent amount of copper is plated out on the surface of the iron, and protects it somewhat from further action. Similarly when iron is immersed in water, hydrogen is plated out and forms a thin invisible film on the iron surface.

**FACTORS CONTROLLING CORROSION.**—Primary factors are those having to do with the initial tendency toward solution, and are mainly associated with the metal itself. Secondary factors are those that influence the subsequent rate of corrosion, and usually are much more important than primary factors. They mainly depend on the environment of the metal. The main factors influencing corrosion are listed below, regardless of their relative importance which changes materially with environment.

**Primary Factors Having to Do Mainly with the Metal.**—1. Effective electrode potential of the metal in solution. 2. Chemical and physical homogeneity and texture of its surface. 3. Its inherent ability to form an insoluble protective film. 4. Overvoltage of hydrogen on the metal surface.

**Factors Which Vary with Environment.**—5. Hydrogen-ion concentration ($p$H) in the solution. 6. Effective supply of oxygen in the solution adjacent to the metal. 7. Distribution of available oxygen on the metal surface. 8. Specific nature, concentration, and distribution of other ions in the solution. 9. Effective rate of flow of the liquid past the metal surface. 10. Presence of solid particles (dirt), or a coating of any kind, as mill scale, on the metal surface, or contact with other conducting material in the presence of an electrolyte. 11. Temperature. 12. Whether the metal is under static stress conditions or exposed to cycles of alternating stress (the so-called corrosion fatigue). 13. The ability of environment to build up protective films on the metal.

While all the above factors are important under some conditions, the following deserve special attention and should be more clearly recognized:

**Oxygen.**—In most cases, oxygen concentration and the rate at which oxygen diffuses to the metal are controlling factors when the metal is immersed in water or buried in soil. The part to which the oxygen diffuses most readily is cathodic to the adjacent areas that are shielded by protective coatings or other material. That is, the points that are less accessible to oxygen corrode more rapidly than areas to which the oxygen has freer access. This and the galvanic effect from contact between dissimilar materials generally causes local corrosion or pitting.

**Surface Films.**—Clean metallic surfaces exposed to air quickly acquire a film of oxide, which at ordinary temperature is thin and invisible. At higher temperatures, it is thick enough to give well-known characteristic colors, varying with the temperature. The nature and properties of the film depend on the composition of the metal itself and its environment. Aluminum and stainless steels, for instance, owe their high durability, in certain environment, to the formation of a continuous and permanent film that is stable under the conditions of exposure. In other environment the film may not form, or is quickly dissolved. The metal, therefore, has short life under such conditions. The addition of 0.25% copper to ordinary steel increases the life of the steel in atmosphere two to four times, due to the formation of a dense adherent film apparently consisting of iron oxide with a little copper oxide on the surface of the metal. When copper steel is immersed in water or buried in soil, the protective film does not form. Under such conditions, copper steel has been no more durable than ordinary steel without copper.

If the metal has no inherent power to form a protective film in solution, a film may be built up by adding film-forming reagents, as the chromates or alkalies, to the water. When removed from the solution, however, these films are only temporary. To secure permanent protection it is necessary to maintain in the water the proper concentration of chromate, or other material, used for this purpose. Films, when formed, should be self healing, so that if injured or removed in any way, a new film will form immediately. Chromium will form a very resistant invisible film under oxidizing conditions, but in dilute hydrochloric acid, where such films cannot exist, chromium dissolves even more rapidly than pure iron. Metals which acquire a stable film by contact with an oxidizing solution are termed passive. In this condition they are at least temporarily resistant to corrosion when removed from the film-forming solution. The presence of thin invisible oxide films on passive material has been determined by the change in potential of the metal and by the removal of the films from the surface by selective dissolution of the underlying metal. See U. R. Evans, Thin Films in Relation to Corrosion Problems, *Jour.* Inst. Metals, 46, 7–23, 1931. Where the corrosion products form spongy layers of irregular thickness, they do not protect, and may even accelerate the normal action by interfering with the rate of diffusion of oxygen to the surface of the metal.

**Corrosion Fatigue.**—Metals subjected to repeated application of stress fail at considerably below their static ultimate tensile strength. However, the metal may be cyclically stressed indefinitely without apparent injury, below the so-called fatigue or endurance limit. This endurance limit is about one-half the tensile strength in the case of carbon steels. Even a small amount of corrosion present with cyclic stress will reduce considerably this endurance limit. Concentrated stress tends to accelerate the rate of corrosion. A notch is soon formed at the point of maximum stress and after this reaches a certain depth, the metal under cyclic stress fails by ordinary fatigue. The combined action of stress and corrosion is termed *corrosion fatigue.* The amount of cyclic stress that a metal will stand under these conditions depends on the character of the corrosion and on the nature of the protective films formed by the metal or its environment. Hence, it is difficult to conceive of a fixed corrosion fatigue limit for any metal under all conditions of exposure. Corrosion factors predominate in corrosion fatigue. Therefore, if there is a so-called corrosion fatigue limit, it usually is independent of the tensile strength. Heat treatment of steel raises the tensile strength and the air endurance limit, but if it does not increase resistance to corrosion it will have no material influence on the resistance to corrosion fatigue. Even 18% chromium steel and 18–8% chromium-nickel steels show a considerable reduction in endurance limit in corrosive environment. Carbon steels in the as-rolled condition may fail at less than one-half of their air-endurance limit when kept continuously wet, depending on the composition of the water and the amount of oxygen present.

An exceedingly small amount of corrosion is sufficient to depress the endurance limit of the metal under stress. In practice, therefore, it is important to prevent contact between stressed portions of metal and corrosive solutions. When inhibitors have been added to water that ordinarily causes failure at a relatively low stress by corrosion fatigue, the endurance of the steel has been maintained at the same stress as in dry air. This apparently is due to the formation and maintenance of a protective film on the surface of the metal under stress. See D. J. McAdam, Stress-Strain Cycle Relationship and Corrosion Fatigue of Metals, *Proc.*, A.S.T.M., Vol. 26, Pt. II, pp. 224–254, 1926; Corrosion Fatigue of Non-ferrous Metals, *Proc.* A.S.T.M., Vol. 27, Pt. II, p. 102, 1927; Some Factors Involved in Corrosion and Corrosion-Fatigue of Metals, *Proc.* A.S.T.M., Vol. 28, Pt. II, pp. 117–158, 1928; and Stress Corrosion of Metals, presented at the Convention of the New International Assoc. for the Testing of Materials, Zurich, Switzerland, 1931; also F. N. Speller, I. B. McCorkle, and P. F. Mumma, Influence of Corrosion Accelerators and Inhibitors on Fatigue of Ferrous Metals, *Proc.* A.S.T.M., Vol. 28, Pt. II, pp. 159–167, 1928.

**ELECTROLYTIC CORROSION DUE TO OVERSTRAIN.**—(C. F. Burgess, *El. Rev.*, Sept. 19, 1908).—Mild steel bars overstrained in their middle portion were subjected to corrosion by suspension in dilute hydrochloric acid solutions, and others by making them the anode in neutral solutions of ammonium chloride and causing current to flow under low current density. In all cases a marked difference was noted in the rate at which the strained portions corroded as compared with the unstrained. Differences of potential of from five to nine millivolts were noted between two electrodes, one of which constituted the strained portion and one the unstrained. The more rapid electrolytic corrosion of the strained portion appears to be due to the fact that the strained metal is electro-positive to the unstrained, the current finding the easier path through the surface of the electro-positive metal. That the strained metal is the more electro-positive is also shown by a liberation of hydrogen bubbles on the unstrained portion.

**EFFECT OF CORROSION ON WROUGHT MANGANESE BRONZE UNDER TENSILE STRESS.** (Paul D. Merica and R. W. Woodward, *Proc.* A.S.T.M., Vol. 19, 1919).—An investigation to determine what service stresses may, with safety, be applied to physically normal wrought brass and bronze showed that brass rods exposed to the action of concentrated ammonium hydroxide and subject at the same time to a gradual application of tensile stress, would break with little elongation at any values of the stress greater than the yield point of the material. It appears that the effect of combined tensile stress and surface corrosion is to decrease the stress at which fracture will occur. This discovery is important in that brass and bronze are substituted for steel usually only on account of their supposed superior resistance to corrosion. If this superior resistance is not as great as has been assumed, the field of brass and bronze for structural purposes is largely limited. The general conclusion seems to be that brass or bronze should not be subject to corrosion in ammonium hydroxide while under a tensile stress greater than 20,000 lb. per sq. in. or more than 5000 lb. per sq. in. above the yield point without danger of failure. The tests indicate that the proportional limit is to be regarded as the maximum safe stress for manganese bronze of harder tempers but it is not certain that this limit may not be slightly exceeded in materials that are soft; *i.e.*, free from work hardness.

## 3. PRINCIPLES OF TESTING

**RELATIVE CORROSION.**—Difference of opinion regarding the relative corrosion of iron and steel and other metals apparently is due to failure to appreciate that different factors control in different environments. Factors external to the metal usually determine the corrosion rate, and these vary considerably with environment. The relative rate, therefore, may vary under different conditions of exposure. This should be the basis on which all corrosion tests (laboratory or service) are designed. Thus, in a study of metals, under atmospheric conditions, the dominant factor should be an excess of oxygen. If the endurance limit of metal under water is in question, the test should be made in that environment alone. It is a mistake to take the results of tests made under one type of corrosion, *i.e.*, in the atmosphere or in acids, and to apply the conclusions to the conditions in which entirely different factors control. An extreme instance is the well-known acid test made by exposing the metals to dilute sulphuric acid, in regard to which the A.S.T.M. has issued the following statement:

> The continued functioning during the past 25 years of the Committee on Corrosion of Iron and Steel of the A.S.T.M. has developed no further information as to the relation of the accelerated acid test to corrosion by the elements, or the trustworthiness of the acid test as a measure of natural corrosion.
>
> Studies have been made to determine whether consistent results can be obtained in testing iron and steel sheets in 20% sulphuric acid, and if the results in acid compare favorably with the results of corrosion tests in the atmosphere. See Report on Total Immersion Tests, *Proc.* A.S.T.M., Vol. 31, Part I, p. 176, 1931.
>
> A representative average set of samples from two sheets each of ten materials of iron and steel, both copper bearing and non-copper bearing, part of each sheet having been exposed to the atmosphere, was tested by five co-operating laboratories following an exact accelerated acid test procedure. The life in acid of similar sheets in atmospheric corrosion tests at Pittsburgh, Pa., and Key West, Fla., indicated that an acid test conducted by the procedure followed by the committee is not capable of consistent repetition. Further, the data from such an acid test should not be used to forecast the relative life of ordinary ferrous materials in the atmosphere.

Clearly, the subject of relative corrosion should be studied on the basis of type of service. The service test is the final criterion of the value of the metal, although judiciously made laboratory tests, with proper consideration given to the factors involved, will show under accelerated conditions a fair approximation of results in service. Three common types of service to be considered are atmospheric, water and soil corrosion.

**ATMOSPHERIC CORROSION.**—In atmospheric corrosion, oxygen is the dominant factor. In general, results of tests indicate little difference between the various kinds of iron and steel except where copper is present in excess of 0.07%. With copper content of from 0.2 to 0.5%, the durability of steel in air is two to four times that of steel without copper. Fig. 1 shows the influence of copper on corrosion. The greatest contrast between copper steel and non-copper steel is found in industrial atmospheres. Wrought iron without copper shows slightly longer life than non-copper steel under outdoor conditions. This probably is due to the better adherence of the products of corrosion to the rough surface of the metal. Sulphur in excess of 0.06% tends to increase the rate of corrosion in industrial air. For results of authoritative work on this type of corrosion, see reports of Committee A-5 in *Proc.* A.S.T.M. dating from 1915 to the present time.

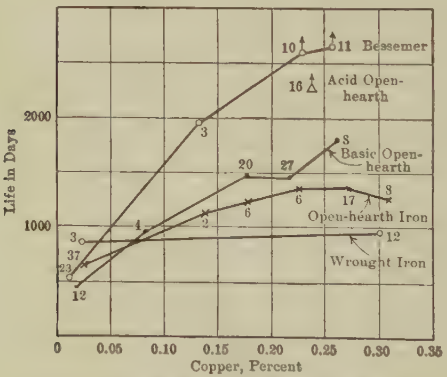

Fig. 1. Influence of Copper on Corrosion

A critical study of A.S.T.M. tests (V. V. Kendall and E. S. Taylerson, A Critical Study of the A.S.T.M. Corrosion Data on Uncoated Commercial Iron and Steel Sheets, *Proc.* A.S.T.M., Vol. 29, Pt. II, pp. 204–219, 1929), is summarized as follows:

> A study of the Pittsburgh and Fort Sheridan results confirms previous conclusions as to the

effect of copper. At the close of eleven years of observations of these tests, the sub-committee would direct attention to the fact that failures at the Fort Sheridan location confirm the findings at the Pittsburgh location, that copper-bearing metal shows marked superiority in rust-resisting properties as compared to non-copper bearing metal of substantially the same general composition under atmospheric exposure. (Report of Committee A-5 on Corrosion of Iron and Steel, *Proc.* A.S.T.M., Vol. 28, Pt. I, p. 156, 1928.) This paper also points out that at the Pittsburgh location, phosphorus combined with copper has beneficial effects from an atmospheric corrosion standpoint. Manganese and sulphur, within the limits of composition involved, seem to have little, if any, effect either way. The Fort Sheridan data, although limited on account of the incompleteness of the test, show similar beneficial results for high-copper material. The A.S.T.M. total immersion tests have been thoroughly studied, and, contrary to the atmospheric tests, the results indicate that the variation of the common elements, either singly or in combination, within range of composition and under the conditions of these tests, has no material influence on corrosion resistance and is entirely secondary to external conditions.

**Tests of Vent Lines** on the roofs of buildings were carried out by the National Tube Co., with wrought-iron, steel, and copper-steel pipe. Results based on the decrease in wall thickness indicated the relative loss of these metals to be as follows, taking steel as 100: wrought-iron, 92%, copper-steel, 36%.

**Tests on Hopper and Gondola Cars,** constructed one-half of copper steel and the other half of plain steel, showed after 13 years that the copper steel had about 200% longer life in the hopper cars (subjected to some abrasion with corrosion), and 300% longer life for the side sheets of the gondolas (subject to corrosion only) in comparison with plain steel of the same composition. The number of cars tested was 100. U. S. Unger (Results of Using Copper Bearing Steel in Freight Cars, *Rwy. Age*, Vol. 84, p. 1062, May, 1928), summarizes the experience with these cars as follows:

1. Paint adheres much better to copper-bearing steel in a car body than to plain open-hearth steel. 2. The saving in repainting cars, due to better adherence of paint to copper-bearing steel, would be sufficient to justify the use of copper-bearing steel in the bodies of steel freight cars. 3. Where mechanical abrasion has not been a serious factor, as in the side sheets of gondolas, the loss in thickness for the copper-bearing steel was only one-third as great as for the plain open-hearth steel. 4. Where the steel was subjected to severe mechanical abrasion as well as corrosion, the loss in thickness for the copper-bearing steel was approximately 60% as great as for the plain open-hearth steel. 5. From the results of this investigation, it can be conservatively stated that the use of all copper-bearing steel in the body of the cars would increase the life of the car body from 33 1/3 to 50%.

**Service Tests on Copper Steel Tie Plates,** containing 0.25% Cu, over a period of 11 years, show 5% loss in weight for copper steel and 20% loss for plain steel. Similar results have been obtained on cut spikes. See Report of Committee on Iron and Steel Structures, Am. Ry. Engrs. Assoc., *Rwy. Age*, Vol. 86, 562 pp. 166–167, 1929. Tests of 730 plain and a similar number of copper steel ties, in track for 9 years, indicate at least double the life for copper steel ties with a sub-grade of broken slag and ashes for ballast. See J. S. Unger, Comparison of Corrosion Resistance of Copper Steel and Plain Steel Railroad Ties, *Steel Maker*, Vol. 2, No. 1, pp. 93–95, Summer 1931.

**WATER CORROSION.**—When material is in continuous contact with water, the available oxygen is limited to its solubility in water. This affects the nature of the film formed by corrosion products, and therefore the rate of corrosion. Immersion tests were made by A.S.T.M. on the materials referred to above, in their atmospheric tests. The results show no material difference between the copper steels and similar steels without copper. Wrought-iron, steel and ferrous alloys commonly used in the manufacture of pipe, tested in service in warm water and return steam pipe lines, show the results given in Table 1. They are summarized according to condition of service and material in Table 2. In connection with these tables the following should be noted: 1. Tests are arranged in chronological order, 1910 to 1931. 2. Table 1 covers the results of 57 tests expressed in terms of measured pitting. 3. Table 2 is a classification of the tests listed in Table 1, based on character of service and material, and gives 78 comparisons of other ferrous material against steel. 4. For convenience, wherever possible, the amount of corrosion also has been figured in percentage, the corrosion being taken as 100.

Examination of these test samples from various parts of the country indicate no fundamental difference in the rate of corrosion of any of the materials *in water*, except that the copper steels are somewhat more durable for return steam lines, where corrosion at times is of the wet and dry type as in air. For further information and practical experience with steel pipe for water service, see H. R. Redington, J. L. W. Birkinbine, F. N. Speller, Corrosion of Water Mains, *Jour.* A.W.W.A., Vol. 23, No. 11, Nov. 1931.

**SOIL CORROSION.**—Under ground, the dominant factor is local electrolytic action, resulting in pitting. The rate of penetration varies with moisture content, electrical resistivity, physical structure and chemical composition of the soil and ground water, and stray electric currents leaving the pipe (electrolysis).

(*Continued on p.* 3-11)

**Table 1.—Summary of Complete Record of Comparative Service Corrosion Tests of Pipe**

Tests conducted in co-operation with outside interests by Dept. of Metallurgy and Research, National Tube Co., Pittsburgh, Pa. For significance of symbols in Tables 1 and 2, see Notes A, B and C, p. 3–10.

| Location. See Note A | Year Removed | Duration y = years m = months d = days | Character of Service. See Note B | Pipe | | | Penetration of Corrosion | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Size, in. | No. in Test | Kind. See Note C | Mean of Average of 4 to 10 Deepest Pits in Each Pipe in Test | | | Average of the Deepest Pits. One Pit—the Deepest—from Each Pipe in Test | | |
| | | | | | | | Depth, in. | Per-cent. Steel = 100 | Kind. Note C | Depth, in. | Per-cent. Steel = 100 | Kind. Note C |
| 1 | 1910 | 2y | HWL | 2 | 2 | S | 0.081 | 100 | S | 0.108 | 100 | S |
| | | | | | 3 | SC$_{1.0}$ | .067 | 83 | SC$_{1.0}$ | .093 | 86 | CS$_{1.0}$ |
| 2 | 1910 | 6m–8y | BFL | 1–4 | 26 | S | .109 | 100 | S | .108 | 100 | S |
| | | | | | 26 | W | .118 | 108 | W | .112 | 104 | W |
| 3 | 1911 | 6m–8y | BFL | 1 1/2 & 2 | 8 | W | .086 | 125 | W | .110 | 118 | W |
| | | | | | 6 | S | .069 | 100 | S | .085 | 100 | S |
| 4 | 1911 | 8y | HWM | 1–1 1/2 | 42 | WG | .092 | 98 | WG | .105 | 100 | WG |
| | | | | | 27 | SG | .094 | 100 | SG | .105 | 100 | SG |
| 5 | 1913 | 6y–10y | HWS & SRL | ..... | 9 | W | ....... | ..... | ....... | .095 | 142 | W |
| | | | | ..... | 9 | S | ....... | ..... | ....... | .067 | 100 | S |
| 6 | 1913 | 2y 1m | HWR | 2 | 8 | W | .069 | 105 | W | .089 | 103 | W |
| | | | | | 4 | S | .066 | 100 | S | .086 | 100 | S |
| | | | | | 4 | SC | .067 | 102 | SC | .086 | 100 | SC |
| 7 | 1915 | 11 1/2m | HWM | 2 | 1 | W | .080 | 94 | W | .104 | 96 | W |
| | | | | | 1 | S | .085 | 100 | S | .108 | 100 | S |
| 7 | 1915 | 11 1/2m | HWM | 1 1/4 | 3 | S | .089 | 100 | S | .102 | 100 | S |
| | | | | | 3 | SF | .057 | 64 | SF | .063 | 62 | SF |
| 8 | 1915 | 9m | MRW | 3/4 | 2 | W | ....... | ..... | ....... | .017 | ..... | W |
| | | | | | 4 | WG | ....... | ..... | ....... | .053 | ..... | WG |
| | | | | | 2 | S | ....... | ..... | ....... | ....... | ..... | ....... |
| | | | | | 2 | SG | ....... | ..... | ....... | .000 | ..... | SG |
| | | | | | 2 | SC | ....... | ..... | ....... | .014 | ..... | SC |
| | | | | | 2 | SF | ....... | ..... | ....... | .000 | ..... | SF |
| 9 | 1916 | 10m | HWM | 2 | 4 | W | .116 | 105 | W | .125 | 99 | W |
| | | | | | 7 | S | .110 | 100 | S | .126 | 100 | S |
| 9 | 1916 | 11m | HWM | 2 | 1 | W | .042 | 117 | W | .047 | 118 | W |
| | | | | | 1 | S | .036 | 100 | S | .040 | 100 | S |
| | | | | | 1 | SF | .016 | 44 | SF | .019 | 48 | SF |
| | | | | | 1 | SG | .035 | 94 | SG | .045 | 113 | SG |
| 10 | 1916 | 1y 1m | HWR | 3/4 | 6 | W | .043 | 119 | W | .057 | 127 | W |
| | | | | | 6 | S | .036 | 100 | S | .045 | 100 | S |
| | | | | | 3 | SF | .023 | 64 | SF | .031 | 69 | SF |
| 11 | 1916 | 55d | MW | 2 | 1 | W | .048 | 209 | W | .070 | 226 | W |
| | | | | | 1 | S | .023 | 100 | S | .031 | 100 | S |
| 12 | 1916 | 2y 9 1/2m | HWM | 2 | 3 | W | .057 | 110 | W | .073 | 114 | W |
| | | | | | 3 | S | .052 | 100 | S | .064 | 100 | S |
| 13 | 1916 | 2y 5m | HWM | 2 | 2 | W | .075 | 107 | W | .085 | 97 | W |
| | | | | | 4 | S | .070 | 100 | S | .088 | 100 | S |
| 14 | 1916 | 2y 6m | HWM | 2 | 2 | W | .061 | 179 | W | .077 | 179 | W |
| | | | | | 4 | S | .034 | 100 | S | .043 | 100 | S |
| 15 | 1917 | 2y 6m | SRL | 2 1/2 | 1 | S | .021 | 100 | S | .027 | 100 | S |
| | | | | | 1 | SC | .022 | 105 | SC | .027 | 100 | SC |
| 9 | 1917 | 1y 2m | HWV | 2 | 6 | W | .117 | 103 | W | .138 | 105 | W |
| | | | | | 6 | S | .114 | 100 | S | .131 | 100 | S |
| 16 | 1917 | 2y | HWL | 1 | 4 | WG | .057 | 102 | WG | .071 | 99 | WG |
| | | | | | 4 | SG | .056 | 100 | SG | .072 | 100 | SG |
| 17 | 1918 | 2y | HSW | 1 | 3 | W | .088 | 138 | W | .135 | 157 | W |
| | | | | | 3 | WG | .049 | 77 | WG | .083 | 97 | WG |
| | | | | | 3 | S | .064 | 100 | S | .086 | 100 | S |
| | | | | | 3 | SG | .030 | 47 | SG | .061 | 71 | SG |
| 18 | 1918 | 11m | HWM | 2 | 2 | W | .070 | 121 | W | .080 | 108 | W |
| | | | | | 2 | S | .058 | 100 | S | .074 | 100 | S |
| | | | | | 1 | SG | .057 | 98 | SG | .068 | 92 | SG |
| | | | | | 2 | SC | .070 | 121 | SC | .083 | 112 | SC |
| | | | | | 1 | SCG | .105 | 181 | SCG | .118 | 159 | SCG |
| 19 | 1919 | 3y | HWM | 1 1/2 | 2 | W | ....... | ..... | ....... | .073 | 162 | W |
| | | | | | 2 | SF | ....... | ..... | ....... | .045 | 100 | SF |
| | | | | | 2 | SG | ....... | ..... | ....... | .078 | 173 | SG |
| | | | | | 2 | SC | ....... | ..... | ....... | .068 | 151 | SC |

*Table continued on following page.*

**Table 1.**—*Continued*

| Location. See Note A | Year Removed | Duration y = years m = months d = days | Character of Service, See Note B | Pipe | | | Penetration of Corrosion | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Mean of Average of 4 to 10 Deepest Pits in Each Pipe in Test | | | Average of the Deepest Pits. One Pit—the Deepest—from Each Pipe in Test | | |
| | | | | Size, in. | No. in Test | Kind. See Note C | Depth, in. | Percent. Steel = 100 | Kind. Note C | Depth, in. | Percent. Steel = 100 | Kind. Note C |
| 9 | 1919 | 2y | HWM | 2 | 4 | WG | ....... | ..... | ....... | 0.114 | 104 | WG |
| | | | | | 4 | SG | ....... | ..... | ....... | .110 | 100 | SG |
| 15 | 1919 | 2y 5m | HWM | 2 | 1 | W | 0.051 | 91 | W | .082 | 92 | W |
| | | | | | 4 | S | .056 | 100 | S | .089 | 100 | S |
| 1 | 1921 | 5y | HWV | 2 | 2 | W | .039 | 98 | W | .062 | 95 | W |
| | | | | | 4 | S | .040 | 100 | S | .065 | 100 | S |
| 12 | 1921 | 4y 6m | HWV | 2 | 6 | W | .041 | 103 | W | .059 | 102 | W |
| | | | | | 14 | S | .040 | 100 | S | .058 | 100 | S |
| 12 | 1921 | 10y | HWV | 2 | 1 | W | .061 | 115 | W | .089 | 97 | W |
| | | | | | 5 | S | .053 | 100 | S | .092 | 100 | S |
| 13 | 1921 | 4y 7m | HWV | 2 | 2 | W | .035 | 106 | W | .098 | 172 | W |
| | | | | | 4 | S | .033 | 100 | S | .057 | 100 | S |
| 20 | 1921 | 5y 8m | HWV | 2 | 3 | W | .047 | 84 | W | .065 | 75 | W |
| | | | | | 2 | S | .056 | 100 | S | .087 | 100 | S |
| 21 | 1921 | 4y 7m | HWM | 2 | 4 | W | .098 | 131 | W | .110 | 117 | W |
| | | | | | 4 | S | .075 | 100 | S | .094 | 100 | S |
| 22 | 1923 | 6y 8m | HWM | 2 | 3 | W | .074 | 97 | W | .088 | 99 | W |
| | | | | | 3 | S | .076 | 100 | S | .089 | 100 | S |
| | | | | | 4 | SF | .065 | 86 | SF | .074 | 84 | SF |
| 14 | 1924 | 7y 6m | HWV | 2 | 2 | W | .045 | 145 | W | .059 | 123 | W |
| | | | | | 4 | S | .031 | 100 | S | .048 | 100 | S |
| 23 | 1926 | 9y 8m | BFL | 1 1/2 | 1 | W | .100 | 130 | W | .130 | 126 | W |
| | | | | | 1 | S | .077 | 100 | S | .103 | 100 | S |
| 24 | 1927 | 8y | SRL | 1 1/4 | 1 | W | .041 | 68 | W | .051 | 72 | W |
| | | | | | 1 | S | .060 | 100 | S | .071 | 100 | S |
| 25 | 1927 | 1y 2m | HWS | 2 | 2 | W | .125 | 130 | W | .147 | 130 | W |
| | | | | | 2 | S | .096 | 100 | S | .113 | 100 | S |
| | | | | | 2 | SS | .109 | 114 | SS | .132 | 117 | SS |
| | | | | | 2 | $SSC_{1.0}$ | .114 | 119 | $SSC_{1.0}$ | .123 | 109 | $SSC_{1.0}$ |
| 26 | 1927 | 12y | HWS | 1 | 2 | W | .083 | 124 | W | .097 | 109 | W |
| | | | | | 2 | S | .067 | 100 | S | .089 | 100 | S |
| 27 | 1927 | 309d | SRL | 1 | 1 | W | .044 | 129 | W | .050 | 111 | W |
| | | | | | 1 | S | .034 | 100 | S | .045 | 100 | S |
| 28 | 1927 | 160d | CCW | 1 1/2 | 1 | W | .051 | 142 | W | .057 | 110 | W |
| | | | | | 1 | S | .036 | 100 | S | .052 | 100 | S |
| 28 | 1927 | 1y 2m | CCW | 1 1/2 | 1 | W | .047 | 127 | W | .058 | 138 | W |
| | | | | | 1 | S | .037 | 100 | S | .042 | 100 | S |
| 29 | 1928 | 11m | HWS | 2 | 2 | W | .034 | 126 | W | .044 | 126 | W |
| | | | | | 1 | S | .027 | 100 | S | .035 | 100 | S |
| 30 | 1929 | 1y 1 1/2m | SRL | 1 | 2 | W | .064 | 108 | W | .107 | 157 | W |
| | | | | | 2 | S | .059 | 100 | S | .068 | 100 | S |
| 31 | 1929 | 10y | SRL | 1 1/4 | 2 | W | .050 | 143 | W | .067 | 163 | W |
| | | | | | 2 | S | .035 | 100 | S | .041 | 100 | S |
| | | | | | 2 | SC | .014 | 40 | SC | .019 | 46 | SC |
| 32 | 1929 | 2y 3m 24d | SRL | 3/4 | 2 | W | .026 | 113 | W | .040 | 118 | W |
| | | | | | 1 | S | .023 | 100 | S | .034 | 100 | S |
| | | | | | 1 | SC | .019 | 83 | SC | .026 | 76 | SC |
| 32 | 1929 | 2y 4m 11d | SRL | 3/4 | 2 | W | .020 | 83 | W | .029 | 73 | W |
| | | | | | 1 | S | .024 | 100 | S | .040 | 100 | S |
| | | | | | 1 | SC | .015 | 63 | SC | .026 | 65 | SC |
| 33 | 1929 | 1y 2 1/2m | HWS | 1 1/4 | 1 | W | .034 | 87 | W | .054 | 93 | W |
| | | | | | 1 | WG | .021 | 54 | WG | .035 | 60 | WG |
| | | | | | 1 | S | .039 | 100 | S | .058 | 100 | S |
| | | | | | 1 | SG | .026 | 67 | SG | .060 | 103 | SG |
| | | | | | 1 | SC | .029 | 74 | SC | .068 | 117 | SC |
| 27 | 1929 | 2y 2 1/2m | SRL | 1 | 1 | W | .045 | 237 | W | .082 | 205 | W |
| | | | | | 1 | S | .019 | 100 | S | .040 | 100 | S |
| 29 | 1929 | 2y 4 1/2m | HWS | 2 | 2 | W | .051 | 128 | W | .062 | 124 | W |
| | | | | | 1 | S | .040 | 100 | S | .050 | 100 | S |
| 29 | 1929 | 2y 4 1/2m | HWS | 2 | 2 | W | .052 | 127 | W | .059 | 107 | W |
| | | | | | 1 | S | .041 | 100 | S | .055 | 100 | S |

*Table continued on following page.*

Table 1.—*Continued*

| Location. See Note A | Year Removed | Duration y = years m = months d = days | Character of Service. See Note B | Pipe | | | Penetration of Corrosion | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Size, in. | No. in Test | Kind. See Note C | Mean of Average of 4 to 10 Deepest Pits in Each Pipe in Test | | | Average of the Deepest Pits. One Pit—the Deepest—from Each Pipe in Test | | |
| | | | | | | | Depth, in. | Per-cent. Steel = 100 | Kind. Note C | Depth, in. | Per-cent. Steel = 100 | Kind. Note C |
| 34 | 1930 | 4y | HWS | 2 | 6 | W | 0.101 | 146 | W | 0.121 | 136 | W |
| | | | | | 3 | S | .069 | 100 | S | .089 | 100 | S |
| | | | | | 3 | SS | .099 | 143 | SS | .109 | 122 | SS |
| | | | | | 3 | $SSC_{0.30}$ | .078 | 113 | $SSC_{0.30}$ | .097 | 109 | $SSC_{0.30}$ |
| | | | | | 3 | $SSC_{1.02}$ | .069 | 100 | $SSC_{1.02}$ | .084 | 94 | $SSC_{1.02}$ |
| | | | | | 3 | $SSC_{1.42}$ | .077 | 112 | $SSC_{1.42}$ | .105 | 118 | $SSC_{1.42}$ |
| 35 | 1930 | 1y 6m | BFL | 2 | 3 | W | .066 | 84 | W | .084 | 84 | W |
| | | | | | 3 | WG | .074 | 94 | WG | .106 | 106 | WG |
| | | | | | 1 | S | .079 | 100 | S | .100 | 100 | S |
| | | | | | 2 | SG | .075 | 95 | SG | .115 | 115 | SG |
| | | | | | 2 | SC | .061 | 77 | SC | .082 | 82 | SC |
| | | | | | 1 | T | .095 | 120 | T | .114 | 114 | T |
| 36 | 1930 | 6 1/2m | CWS | 3/4 | 2 | W | .016 | 178 | W | ....... | ..... | ....... |
| | | | | | 2 | S | .009 | 100 | S | ....... | ..... | ....... |
| 37 | 1930 | 8m | SRL | 2 | 2 | W | .052 | 79 | W | .063 | 85 | W |
| | | | | | 2 | S | .066 | 100 | S | .074 | 100 | S |
| | | | | | 1 | SC | .035 | 53 | SC | .037 | 50 | SC |
| 38 | 1931 | 7y 9m | RBA | 1 1/4 | 2 | W | .045 | 57 | W | .072 | 76 | W |
| | | | | | 6 | S | .079 | 100 | S | .095 | 100 | S |
| 33 | 1931 | 2y<br>11 1/2m | HWS | 1 1/4 | 1 | W | .072 | 133 | W | .094 | 147 | W |
| | | | | | 1 | WG | .052 | 96 | WG | .057 | 89 | WG |
| | | | | | 1 | S | .054 | 100 | S | .064 | 100 | S |
| | | | | | 1 | SG | .046 | 85 | SG | .063 | 98 | SG |
| | | | | | 1 | SC | .041 | 76 | SC | .052 | 81 | SC |
| 27 | 1931 | 3y 9 1/2m | SRL | 1 | 1 | W | .067 | 152 | W | .082 | 146 | W |
| | | | | | 1 | S | .044 | 100 | S | .056 | 100 | S |
| 32 | 1931 | 3y 11m | HWM | 3/4 | 2 | W | .072 | 136 | W | .091 | 125 | W |
| | | | | | 1 | S | .053 | 100 | S | .073 | 100 | S |
| | | | | | 1 | SC | .038 | 72 | SC | .050 | 68 | SC |
| 1 | 1931 | 4y<br>10 1/2m | HWL | 2 | 1 | W | .079 | 141 | W | .095 | 130 | W |
| | | | | | 1 | S | .056 | 100 | S | .073 | 100 | S |
| | | | | | 1 | $SSC_{0.30}$ | .066 | 118 | $SSC_{0.30}$ | .092 | 126 | $SSC_{0.30}$ |
| | | | | | 1 | $SSC_{0.76}$ | .058 | 104 | $SSC_{0.76}$ | .079 | 108 | $SSC_{0.76}$ |
| | | | | | 1 | $SSC_{1.02}$ | .055 | 98 | $SSC_{1.02}$ | .075 | 103 | $SSC_{1.02}$ |

NOTES ON TABLE 1. NOTE A. Location of tests were: 1. New York City, Allen St. Bath House. 2. H. C. Frick Coke Co. Power Plants. 3. Cresson, Pa., Coal Fields. 4. Pittsburgh, Pa., Allegheny General Hospital. 5. New York City, Hotel. 6. Philadelphia, Pa., Pennsylvania Bldg. 7. Scranton, Pa., International Correspondence Schools. 8. West Homestead, Pa., C. F. Greiner Plumbing Co. 9. Pittsburgh, Pa., Irene Kaufmann Settlement. 10. St. Louis, Mo., Third National Bank. 11. Nanticoke, Pa., Susquehanna Coal Co. 12. New York City, W. 41st St. Bath House. 13. New York City, E. 76th St. Bath House. 14. New York City, E. 109th St. Bath House. 15. New York City, Cherry and Oliver St. Bath House. 16. McKeesport, Pa., National Works. 17. Atlantic City, N. J., Hotel Traymore. 18. Providence, R. I., Brown Univ. 19. Cambridge, Mass., Harvard Univ. 20. New York City, Rutgers Pl. Bath House. 21. Pittsburgh, Pa., Wm. Penn Hotel. 22. Philadelphia, Pa., Widener Bldg. 23. San Diego, Cal., San Diego Consolidated Gas & Elec. Co. 24. Auburn, R. I., Budlong Rose Co. 25. Pittsburgh, Pa., Mellon Inst. of Industrial Research. 26. Rochester, N. Y., Mechanics Inst. 27. Oakland, Cal., Athens Athletic Club. 28. Emeryville, Cal., General Cable Co. 29. San Francisco, Cal., Phelan Bldg. 30. Ithaca, N. Y., Cornell Univ. 31. Cromwell, Conn., A. N. Pierson Greenhouse. 32. Los Angeles, Cal., Biltmore Hotel. 33. Sacramento, Cal., Hotel Senator. 34. Ellwood City, Pa., National Tube Co. 35. Los Angeles, Cal., Bartlett Bldg. 36. New England Dressed Meat and Wool Co. 37. An Eastern engineering corporation (name confidential). 38. Ashland, Mass., Aniline Dye & Chemical Co.

NOTE B. Character of Service: HWL, Hot water line; BFL, Boiler feed water line; HWM, Hot water system; SRL, Steam return line; HWS, Hot water supply; HWR, Hot water return; MRW Monongahela River water; MW, Mine water; HSW, Hot salt water; HWV, Hot water service; CCW, Circulating cooling water, 100–140 deg. F.; CWS, Cold water service in grease cellar extractor, corrosion by fatty acids; RBA, Refrigerating brine and atmosphere alternately; AVP, atmosphere in vent pipe; A, Atmosphere.

NOTE C. Kind of Material in Pipe: W, wrought iron; WG, galvanized wrought iron; S, steel; SC, copper steel (subscript denotes percentage of copper); SG, galvanized steel; SF, "scale-free" steel; SCG, copper steel, galvanized; SS, seamless steel; SSC, seamless copper steel; T, toncan iron.

**Table 2.—Summary of Grand Averages of Complete Record of Comparative Service Corrosion Tests of Pipe**

Tests conducted in co-operation with outside interests by Department of Metallurgy and Research, National Tube Co., Pittsburgh, Pa.

Percentages based on steel = 100; and for galvanized materials, galvanized steel = 100.

| Character of Service * | No. of Tests | Steel | Wrought Iron | | Galvanized Steel | | Galvanized Wrought Iron | | Copper-steel | | Scale-free Steel | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Depth, in. | Depth, in. | Per cent | Depth, in. | Per cent | Depth, in. | Per cent | Depth, in. | Per cent | Depth, in. | Per cent |
| | | Mean Averages of 4 to 10 Deepest Pits in each Pipe in Test | | | | | | | | | | |
| HWS | 35 | 0.059 | 0.068 | 115 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| HWS | 6 | ....... | ....... | ....... | 0.055 | 100 | 0.058 | 105 | ....... | ..... | ....... | ..... |
| HWS | 10 | .069 | ....... | ....... | ....... | ..... | ....... | ..... | 0.062 | 90 | ....... | ..... |
| HWS | 4 | .059 | ....... | ....... | ....... | ..... | ....... | ..... | ....... | ..... | .040 | 68 |
| SR | 9 | .040 | .045 | 113 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| SR | 5 | .034 | ....... | ....... | ....... | ..... | ....... | ..... | .021 | 62 | ....... | ..... |
| MW | 1 | .023 | .048 | 209 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| HSW | 1 | .064 | .088 | 138 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| HSW | 1 | ....... | ....... | ....... | .030 | 100 | .049 | 163 | ....... | ..... | ....... | ..... |
| RBA | 1 | .079 | .045 | 57 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| AVP † | 9 | .01038 | .00956 | 92 | ....... | ..... | ....... | ..... | .00374 | 36 | ....... | ..... |
| A † | 2 | .00364 | .00155 | 43 | ....... | ..... | ....... | ..... | .00107 | 29 | ....... | ..... |
| | Average of Deepest Pits. (One pit, the deepest, from each pipe in test) | | | | | | | | | | | |
| HWS | 38 | 0.076 | 0.084 | 111 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| HWS | 7 | ....... | ....... | ....... | 0.084 | 100 | .082 | 98 | ....... | ..... | ....... | ..... |
| HWS | 11 | .084 | ....... | ....... | ....... | ..... | ....... | ..... | 0.080 | 95 | ....... | ..... |
| HWS | 4 | .069 | ....... | ....... | ....... | ..... | ....... | ..... | ....... | ..... | 0.047 | 68 |
| SR | 9 | .052 | .063 | 121 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| SR | 5 | .043 | ....... | ....... | ....... | ..... | ....... | ..... | .027 | 63 | ....... | ..... |
| MW | 1 | .031 | .070 | 226 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| HSW | 1 | .086 | .135 | 157 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |
| HSW | 1 | ....... | ....... | ....... | .061 | 100 | .083 | 136 | ....... | ..... | ....... | ..... |
| RBA | 1 | .095 | .072 | 76 | ....... | ..... | ....... | ..... | ....... | ..... | ....... | ..... |

* Character of Service: HWS = Hot water; SR = Steam return; MW = Mine water; HSW = Hot salt water; RBA = Refrigerating brine and atmosphere; AVP = Atmosphere in vent pipe; A = Atmosphere.

† The figures in these tests are grand averages for penetration of corrosion in inch per year, based either on weight lost or decrease in wall.

(*Continued from page* 3–07)

The U. S. Bureau of Standards has conducted an extensive investigation of steel corrosion from 1924 in typical soils in the U. S. This comprehensive test includes some 15,000 samples. For detailed records of the tests, see Research Papers Nos. 95 and 329, U. S. Bureau of Standards. A summary of the weighted average results, based on maximum pits on ferrous metals from all soils included in these tests over a period of eight years, is given in Fig. 2.

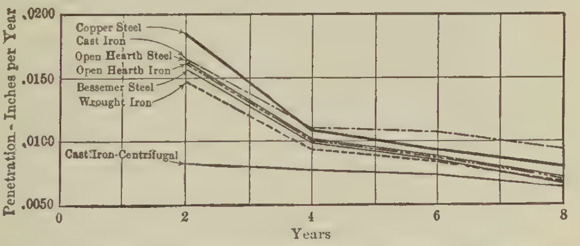

Fig. 2. Results of Bureau Standards Soil Corrosion Investigation

**CHEMICAL CORROSION** includes the action of acids and other chemicals. It is only possible to determine the relative durability of ferrous materials under such variable conditions by actual tests, in each case, in the laboratory, or preferably in service.

## 4. PREVENTION OF CORROSION

Deterioration of metals may be prevented or reduced in three ways: 1. By using a more resistant metal. 2. By application of a suitable protective coating. 3. By making the environment less corrosive. The amount of protection specifically depends on the corrosive nature of the environment and must be chosen to suit the conditions and with a view to economy.

**Protection Against Atmospheric Corrosion.**—A large variety of paints, lacquers, and similar protective materials has been developed, which cannot be summarized here. See publications of American Paint and Varnish Mfrs. Assoc., distributed by the Institute of Paint and Varnish Research Laboratory, Washington, D. C. Many products suitable for atmospheric protection have been developed in the past few years, some being decidedly superior to the old type of linseed oil paints. These are mostly compounds of synthetic resins, applied with or without baking, self-curing rubber compounds, and the nitrocellulose lacquers. See H. A. Gardner, Physical and Chemical Examination of Paints, Varnishes, Lacquers, and Colors; Report of Inst. of Paint and Varnish Research, Oct. 1930.

## Paints and Varnishes

**References.**—Technology of Paint and Varnish, A. H. Sabin (John Wiley & Sons): Reports of Committee, D-1, Am. Soc. Test. Mat.; Modern Pigments and Their Vehicles, Maire (John Wiley & Sons).

**PAINT.**—Composed of a vehicle or binder, usually linseed oil with some tung oil, or varnish (enamel paints), and a pigment, which is a more or less inert solid in the form of a powder, either mixed or ground together. Nearly all paint contains paint drier or japan, which is a lead or (and) manganese compound soluble in oil, and acts as a carrier of oxygen; as little should be used as possible. Boiled oil contains drier; no additional drier is needed. None should be used with varnish paints, nor with "ready-mixed paints" in general.

The principal pigments are white lead (carbonate or oxy-sulphate) and white zinc (oxide), red lead (peroxide), oxides of iron, hydrated and anhydrous, graphite, lampblack, bone black, chrome yellow, chrome green, ultramarine and Prussian blue, and various tinting colors. White lead has the greatest body or opacity of white pigments; three coats of it equal five of white zinc; zinc is more brilliant and permanent, but it is liable to peel, and it is customary to mix the two. These are the standard white paints for all uses, and the basis of all light-colored paints. Anhydrous iron oxides are brown and purplish-brown, hydrated oxides are yellowish-red to reddish-yellow, with more or less brown; most iron oxides are mixtures of both sorts, and often contain a little manganese and much clay. They are cheap, and are serviceable paints on wood and are often used on iron, but for the latter use are falling into disrepute. Graphite used for painting iron contains from 10 to 90% foreign matter, usually silicates. It is very opaque, hence has great covering power and may be applied in a very thin coat, which is to be avoided. The best graphite paints give very good results. There are many grades of lampblack; the cheaper sorts contain oily matter and are especially hard to dry; all lampblack is slow to dry in oil. In a less degree this is true of all paints containing carbon, including graphite. Lampblack is used with advantage with red lead; it is also an ingredient of many "carbon" paints, the base of which is either bone black or artificial graphite. Red lead dries by uniting chemically with the oil to form a cement; it is heavy, and makes an expensive paint, and is often highly adulterated. Pure red lead has long had a high reputation as a paint for iron and steel, and is still used extensively, especially as a first coat; but of late years some of the new paints and varnish-like preparations have displaced it to a considerable extent, even on the most important work.

**VARNISHES** are made by melting fossil resin, to which is then added from half its weight to three times its weight of refined linseed oil, and the compound is thinned with turpentine; they usually contain a little drier. They are chiefly used on wood, being more durable and more brilliant than oil, and are often used over paint to preserve it. Asphaltum is sometimes substituted in part or in whole for the fossil resin, and in this way are made black varnishes which have been used on iron and steel with good results. Asphaltum and substances like it have also been simply dissolved in solvents, as benzine or carbon disulphide, and used for the same purpose.

All these preservative coatings are supposed to form impervious films, keeping out air and moisture; but in fact all are somewhat porous. On this account it is necessary to have a film of appreciable thickness, best formed by successive coats, so that the pores of one will be closed by the next. The pigment is used to give an agreeable color, to help fill the pores of the oil film, to make the paint harder, so that it will resist abrasion, and to make a thicker film. In varnishes these results are sought to be attained by the resin which is dissolved in the oil. There is no sort of agreement among practical men as to which coating is best for any particular case; this is probably because so much depends on the preparation of the surface and the care with which the coating is applied, and also because the conditions of exposure vary so greatly.

**APPLICATIONS.**—Cast-iron water pipes usually are coated by dipping in a hot mixture of coal-tar and coal-tar pitch; riveted or welded steel pipes by dipping in hot asphalt. Ships' bottoms are coated with a red lead or other inhibitive primer and a varnish paint to prevent rusting. Over this is a second coat containing a poison, as mercury chloride or a copper compound, or a greasy copper soap applied hot; these tend to prevent the accumulation of marine growths. Galvanized iron and tin surfaces should be thoroughly cleaned with benzine and scrubbed before painting. When new, they are partly covered with grease and chemicals. These must be removed or paint will not adhere. Weathering for six months prior to painting serves the same purpose in preparing galvanized steel surfaces.

## 5. CORROSION-RESISTING PROCESSES

**CEMENT.**—Iron-work often is bedded in concrete; if free from cracks and voids it is an efficient protection. The metal should be cleaned and then washed with neat cement before embedding.

**BOWER-BARFF PROCESS.**—Dr. Barff's method was to heat articles to be coated to about 1800° F. and inject steam heated to 1000° F. into the muffle. George and A. S. Bower used air instead of steam, then carbon monoxide (producer gas) to reduce the red oxide. In the combined process, the articles are heated to 1600° F. in a closed retort; superheated steam is injected for 20 min., then producer gas for 15 to 25 min.; the treatment can be repeated to increase the depth of oxidation. Less heat is required for wrought- than for cast-iron or steel. By a later improvement, steam heated above the temperature of the articles was injected during the last one to two hours. By a further improvement known as the Wells Process, the work is finished in one operation, the steam and producer-gas being injected together. Articles are slightly increased in size by the treatment. The surface is gray, changing to black when oiled; it will chip off if too thin; it will take paint or enamel and may be polished, but cannot be either bent or machined; the coating itself is incorrodible and resists sea-water, mine-water and acid fumes; the strength of the metal is slightly reduced. The process is extensively used for small hardware. (See F. S. Barff, *Jour.* I. & S. Inst., 1877, p. 356; A. S. Bower, *Trans.* A.I.M.E., 1882, p. 329; B. H. Thwaite, *Proc.* Inst. C. E., 1883, p. 255; George W. Maynard, *Trans.* A.S.M.E., iv, p. 315, 1883.)

**RUSSIA AND PLANISHED IRON.**—Russia iron is made by cementation and slight oxidation. W. Dewees Wood (U. S. Pat. No. 252,166 of 1882) treated planished sheets with hydrocarbon vapors or gas and superheated steam within an air-tight and heated chamber.

**NITER PROCESS.**—An old process improved by Col. A. R. Buffington in 1884. The articles are stirred about in a half-and-half mixture of fused sodium and potassium nitrate (saltpeter) to which is added manganese dioxide in the ratio of 1 part oxide to 50 parts niter by volume. They then are suspended in the vapors, dipped and washed in boiling water and finally immersed in hot oil. Pure chemicals are essential. Used for small arms and pieces which cannot stand the high heat of other processes. The temperature of the bath should be from 600 to 650° F. to yield a temper blue, and 1000° F. to yield the dark gun-metal finish used for rifle parts.

**PHOSPHATIC COATINGS FOR RUST-PROOFING IRON AND STEEL.** (L. E. Eckelmann, *Chem. & Met. Engg.*, Dec. 24, 1919).—Recent phosphatic rust-proofing differs from other methods in that the dimensions, sharpness of edges of the articles are not altered, and the physical properties of tempered steel are not affected. Rust-proofing is effected by immersing the clean objects in a hot solution of iron phosphates in weak $H_3PO_4$. Phosphoric acid solutions will dissolve ferrous phosphate within certain limits in direct proportion to the concentration of the acid present. If any base or metal, as iron, be added—capable of combining with the free acid in such a solution, ferrous phosphate forming on the metallic surfaces will precipitate this salt back on the iron at the point of solution. The action proceeds with the evolution of hydrogen until no more iron is exposed, the article being then completely coated with a basic iron phosphate.

**Coslett Process.**—The foregoing reaction was the basis of the process developed by Coslett in 1907, in which ferrous phosphate, made by mixing iron filings with concentrated $H_3PO_4$ sufficient to form a semi-fluid paste was added to weak boiling phosphoric acid. Iron or steel articles immersed in this solution, for three or four hours, become covered with a rust-resisting light gray deposit of basic ferrous phosphate. The difficulty of obtaining uniform coatings, the great amount of suspended salt adhering to the work, the necessity of rinsing work after treatment, and the non-uniformity of the finish, made the adoption of this process slow.

**Parkerizing** is the process by which a coating of insoluble phosphates is produced on the surface of iron, using a bath containing some free phosphoric acid and a mixture of ferrous monohydrogen phosphate and manganese dihydrogen phosphate. The coating produced is less than 0.0002 in. thick and has maximum rust-resisting ability when the ratio of iron to manganese in the coating is 1 : 1. The free phosphoric acid is produced by the hydrolysis on boiling of ferrous dihydrogen orthophosphate. The bath is maintained at 208–210° F. The ratio of iron to manganese in the bath is 2 : 1. About one hour is required. A small amount of copper phosphate can be added as an accelerator to speed up the reaction and reduce the time required to 10 minutes. The resultant coating forms an excellent base for paints, lacquers, and organic enamels. It can be electroplated either before or after buffing. The coating is not as rust-resistant as the one obtained from the plain phosphate bath but its value can be greatly enhanced by stoving or by short immersion in a hot chromate solution.

**Bonderizing** is the term used when the preliminary accelerated phosphate treatment is applied to act as a base for paints or lacquers. It is claimed that should the surface be scratched or injured during subsequent service, rust cannot spread beneath the Bonderizing coating.

**GALVANIZING** is a method of coating steel or iron articles with zinc, the product being known as galvanized steel or iron. Galvanized coatings usually resist, for a reasonable time, ordinary corrosive conditions such as in fairly pure air, moderately hard domestic waters, sea water, brines, etc. The zinc, particularly under atmospheric conditions, often becomes hardened with a protective layer or film believed to be a zinc-hydroxide-carbonate mixture. Zinc is anodic to iron and therefore protects the base metal, where exposed, until the adjacent outer layer of zinc is dissolved. Galvanized coatings are unsuitable where the corroding agent is very acid, as in certain mine waters and industrial solutions, or where the solutions are excessively alkaline, *e.g.*, calcium brines with an excess of caustic alkalinity. Favorable $p$H values are given between 5 and 8.5 and temperatures below 160° F. In brines the $p$H should not exceed 8. Very hot soft water attacks galvanizing.

Galvanized coatings are commercially applied: *a*, by dipping the article in molten zinc; *b*, by electroplating; *c*, by heating in contact with zinc powder.

**Hot-dip Galvanizing.**—Articles to be hot-dipped galvanized usually are cleaned by acid pickling, washed, and fluxed by immersion in a bath of zinc ammonium chloride or hydrochloric acid. They then are dipped in molten zinc maintained at 830 to 920° F. When the steel has reached approximately the zinc temperature, requiring a few seconds in the case of wire and several minutes for heavier sections, it is drawn slowly from the bath, allowing time for draining. In the case of wire, excess zinc often is removed and the coating smoothed mechanically by asbestos or charcoal wipers. Long, slightly corrugated rolls are similarly used for sheets. The article may cool in air, producing a spangled (crystalline) zinc surface, or it may cool rapidly in water, producing a matte surface.

Adherence of zinc coatings depends on the formation of suitable zinc-iron alloys. Some grades of wire and sheets, conduit pipe, etc., requiring maximum bending qualities are made with light coatings consisting largely of zinc-iron alloy formed by wiping after dipping. Where bending requirements are secondary, and maximum corrosion resistance is desired, as for other grades of iron and sheets, galvanized water pipe, galvanized ware, etc., the galvanizing should be considerably heavier, and with more pure zinc in the coating. The average weight of coating applied varies from $^1/_2$ to 2 $^1/_2$ oz. per sq. ft. of covered surface, depending on the thickness of base metal, temperature and time in bath, and amount of wiping.

In the process of galvanizing, tin is at times added to the zinc bath to improve the spangle, while aluminum may be added to improve the fluidity of the bath and brightness and smoothness of the coating. The vapor of burning sulphur or ammonium chloride dust on the zinc surface as it comes from the bath has been used to improve brightness.

**Electrogalvanizing.**—Electrogalvanized zinc coatings are applied to conduit pipe, wire and intricate shapes. Electrogalvanized coatings on the threads of pipe couplings prevent sticking or galling of the threaded joints, and generally are used for tubular goods in petroleum production. Articles to be electrogalvanized may be cleaned by pickling with or without the aid of electric current, or machined surfaces may be cleaned with alkaline grease removers. The zinc is deposited from sulphate or cyanide baths using low-voltage plating current.

**GALVANIZING BY CEMENTATION. SHERARDIZING.** (H. A. Calderwood, Natl. Elec. Prod. Corp.).—The characteristic feature of the cementation process, sherardizing with zinc being a well-known example, is a definite and appreciable alloying of the surface of the treated metal. The treatment does not result in a layer of the simple secondary metal adhering to the base metal, but in an alloy of the two metals which varies in richness, there being more of the secondary metal at the surface of the coating.

Cementation, such as sherardizing, involves the use of the coating or cementing metal in the form of fine powder. The articles to be treated are placed in a drum or retort, which then is nearly filled with the metal powder, sealed and brought to the proper temperature. This temperature is in some cases well below the melting point of the cementing metal, in others, close to or slightly above its melting point, depending upon the metals involved, the character and texture of surface finish desired, and other considerations such as required speed of operation.

Sherard Cowper-Cowles (England) refined and perfected the process of cementation with zinc, and British and U. S. patents were granted to him in 1902 for the method known as Sherardizing. It consists in packing the articles to be sherardized, usually steel, in a retort with zinc powder known as *blue dust*. The retort or drum is tightly sealed, and by suitable means so heated that its contents attain a temperature approaching the melting point of zinc. The melting point of zinc is 786° F. and sherardizing usually is carried on at from 600 to 700° F., although it can be done at temperatures above or below this range. Better results are obtained if the drums are slowly rotated, and this generally is done in order to bring a fresh supply of dust into intimate contact with the articles being treated. The time required is from two to four hours. After drums are removed from the heating oven, they are allowed to cool before being opened. The time required for cooling is one to several hours, depending on the size of the retort. Prolonged heating and slow cooling of steel articles embedded in zinc dust, which is a poor conductor of heat, tends to soften them, which is highly desirable for objects which must be cut and bent.

**Blue Dust** is a natural by-product of smelting zinc, but when especially made of greater purity and uniformity for the purpose of sherardizing, is known as sherardizing zinc dust. It consists of minute particles of zinc metal, each encased in a thin skin, shell, or pellicle of zinc oxide. Commercial sherardizing zinc contains about 95 to 97% metallic zinc. The remainder is chiefly zinc oxide, with traces of impurities such as iron and lead. The resultant coating is a true zinc-iron alloy, ranging from $Fe\ Zn_3$ at the base metal line to $Fe\ Zn_{10}$ near the surface, and a layer of relatively pure zinc on the exterior; exact analysis naturally is difficult.

**Characteristics of a Sherardized Coating** are: *Color*, a rich gray; *texture*, which is matte, caused by innumerable minute fissures not visible to the unaided eye, and which has a velvety feel; *ability to receive and retain paints*, varnishes, enamels, etc., due to the bonding effect in the minute fissures; *uniform distribution and penetration*, making possible the effective coating of irregularly-shaped and hollow articles, screw threads, holes, etc., without building up on corners or actually filling up depressions, although allowance must be made for thickness on threads and other closely fitted parts: *ductility and tenacity*, which prevents flaking and slivering when sherardized articles such as pipes and sheets are bent; ability to take a high polish, the surface being relatively hard.

**Corrosion Resistance of Sherardizing.**—The principal object of metallic coatings being to prevent corrosion, performance of sherardizing in this respect is important. Concerning exposure to ordinary attacks indoors, and when exposed to weather and the zinc is oxidized, Rawdon (Protective Metallic Coatings, p. 112, Chem. Catalog, 1928) says:

> The coating gradually assumes a dark appearance as the iron likewise is oxidized, finally becoming a very dark gray and almost black if the iron content is not too high. Coatings high in iron, and hence containing the iron-rich compound, usually show a red dusty surface on weathering from the ferric oxide produced. A wrong impression often is formed concerning such coatings, that is, that the coating has been entirely corroded away and that the red dust indicates an exposure of the base metal. It can generally be shown that such is not the case.

When this color is not due to rusting of the base metal, the red or yellowish formation usually disappears in a short time, and the surface darkens. The black coating is believed to be the magnetic oxide of iron. When sherardizing acquires this surface formation it becomes highly resistant to further corrosion. The relation of thickness of coating to durability is important, since a heavy coating is, for mechanical reasons, objectionable on some articles, particularly those subject to bending and flexing.

Opinions differ regarding the mechanism of the sherardizing (cementation) process. Rawdon says:

> All agree that once the zinc is deposited on the surface of the iron article embedded in the zinc dust, alloying results by diffusion in the solid state. The method by which the zinc is deposited on the surface is the disputed point. Some (A. Sange, *Electrochem. and Metal. Ind.*, Vol. 7, p. 485; L. McCulloch, Factors Influencing the Process of Sherardizing, *Elec. Jour.*, Vol. 19, p. 156, *Trans.* A.I.M. & M.E., Vol. 68, p. 757) consider that the slight vaporization of the zinc which occurs even at the relatively low temperature used in this process is sufficient to account for the transportation of the zinc from the dust particles to the iron surface. Others (J. W. Hinchley, Some Practical Experience of the Sherardizing Process, *Trans.* Faraday Soc., Vol. 6, p. 133, and O. W. Storey, The Sherardizing Process, *Met. and Chem. Engg.*, Vol. 14, p. 683) hold that intimate contact of the zinc particles and iron surface is sufficient, and that the zinc particles become attached bodily to the surface. The discussion is one of academic rather than practical interest. Both factors undoubtedly contribute to the coating process as carried out in practice.

**TESTS AND SPECIFICATIONS FOR GALVANIZED COATINGS.**—Galvanized coatings usually are specified as a given weight of zinc per unit area. If bending or forming is anticipated, bend tests may be specified. Coatings may be expressed as ounces of zinc per square foot of surface, and determined by weighing a suitable sample of the galvanized article, removing the zinc, and reweighing. The usual means of removing the zinc is hydrochloric acid containing sufficient antimony chloride to prevent attack on the base metal. The minimum thickness of galvanized coatings, particularly on wire, also is expressed in terms of the Preece test, which consists of dipping the zinc coated article in a neutral solution of copper sulphate (sp. gr. 1.186 at 18° C.) for periods of one minute, washing and wiping between each dip until the zinc is penetrated, as shown by the formation of adherent copper on the exposed surface of the steel. Coating weight is expressed as number of dips to cause failure For details of both methods, see Standard Methods of Determining Weight of Coating on Zinc Coated Articles, A.S.T.M. Specification, Designation A 90–30.

Specifications for most commercial products covering weight of coating and bend tests when required are available in A.S.T.M. Standards, 1936, Part 1, Metals.

On sheets, coating weights are given as weight per sq. ft. of sheet, *i.e.*, 2 sq. ft. of coated surface. Ordinary coatings on sheets range from 1.25 to 2.75 oz. per sq. ft. of sheet coated on both sides. The weight of zinc per single covered surface would be one-half this amount. Zinc coatings on pipe, structural shapes, etc., generally range between 1.5 and 3.0 oz. per sq. ft. of covered surface. The weight of coating on wire varies with the size of wire and process of coating, and, except in fine sizes, will range from 0.15 to 1.25 oz. per sq. ft. of covered surface.

**LEAD COATINGS.**—Lead is very resistant to some types of corrosion, and lead coatings on steel have been used to some extent. A thin coating of lead, however, is difficult to apply, either by electroplating or hot dipping, so as to be sufficiently free from pin holes to be effective when submerged in corrosive water. The atmospheric resistance of such coatings usually is quite good. Hot-dipped lead sheets, known as terne plate, are extensively used. Wrought steel or iron pipes protected on the interior by thick lead tubes mechanically applied, when carefully installed with suitable fittings and valves, are effective for carrying corrosive acids and similar liquids.

**RUST-RESISTING METALS.**—**Iron.**—Various modifications of iron and steel have been introduced to give better service under atmospheric conditions. High purity iron and ingot iron (see p. 2–06) have considerable resistance to corrosion, but the purity of iron has been shown to be a minor factor in most types of natural corrosion, as external conditions usually predominate. This grade of iron is included in the A.S.T.M. corrosion tests referred to on p. 3–06.

**Copper Steel** has found a wide commercial application on account of its slow rusting properties under atmospheric and similar conditions of exposure. See D. M. Buck, A Review of the Development of Copper Steel, Year-Book Am. Iron and Steel Inst., pp. 373–379, 1920. Over 500,000 tons of copper steel in sheets, plates, and similar products have been manufactured for use where the metal is exposed to the weather. The addition of 0.25% copper at an increased price of $3.00 per ton usually will prolong the life of the metal two to four times under such conditions. (See also p. 3–06.)

**Corrosion-resistant Alloys,** containing high percentages of chromium are widely used. See pp. 3–19 to 3–24.

## 6. PROTECTION AGAINST UNDER WATER CORROSION

The main factors controlling the rate of corrosion in water are dissolved oxygen, temperature, rate of motion, and the composition of the water. The chemical composition of ferrous metals, within rather wide limits, does not seem to have a marked influence on the character or amount of corrosion. The high chromium series of steels are an exception to this. With cold water the rate of corrosion is usually slow, but in hot water the corrosive action is accelerated according to the temperature, the rate being doubled for every 25 deg. F. ± increase in temperature.

Localized corrosion or pitting usually is much deeper near mill scale, especially where the scale is thick and firmly adherent, undoubtedly due to the electro-negative character of the scale with respect to the iron. Old rust acts somewhat similarly but is less harmful. Removal of mill scale and rust greatly reduces the tendency to pitting, providing the metal itself is fairly homogeneous and free from strains. Surface finish is a greater controlling factor in corrosion than is the variation in the composition of the metal itself.

Within certain limits, corrosion varies directly with the temperature and the rate of flow and also with the composition of the water. Corrosion is likely to be more rapid on unprotected metal with increasing velocity of the water. The amount of corrosion is almost directly proportional to the amount of oxygen in the solution in the water. Hot water heating systems show practically no corrosion after 35 or 40 years of use, whereas hot water supply systems, operating at the same temperatures, have deteriorated in from 6 to 8 years. The only difference between the two is that the heating system carried no oxygen while the water in the supply system usually is saturated with it.

In some cases it is more economical to prevent internal corrosion by removing dissolved oxygen from the water, especially with hot water. Oxygen removal may be accomplished: 1. By mechanical deaeration for which there are two or three satisfactory designs of apparatus in use. 2. By the chemical combination of the oxygen with some material such as scrap and sheet iron. The two methods may be combined by using mechanical deaeration to remove most of the oxygen and chemical treatment for the remainder.

**MECHANICAL DEAERATING OF WATER** is accomplished by passing the water over baffles or spraying into a high-vacuum chamber at normal temperature, or by increasing the temperature and controlling the pressure and temperature so that more or less complete removal of all the gases is obtained. One system involves heating the water above the boiling point corresponding to the pressure and spraying it into a chamber at lower pressure. The air and aqueous vapor are drawn through a condenser where the vapor is condensed with a simultaneous transfer of heat to the incoming cold water.

For practical protection of water pipe from internal corrosion at 180° F., the oxygen need not be lower than 0.4 c.c. per liter (cold water carries from 5 to 10 c.c. per liter, depending on the season of the year and temperature). To avoid appreciable corrosion in steam boilers, superheaters and economizers operating at a much higher temperature, the oxygen should be below 0.03 c.c. per liter. This may be accomplished by means of deaerators now on the market.

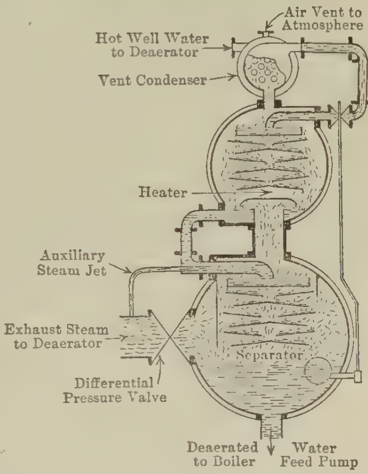
Air Vent to Atmosphere
Hot Well Water to Deaerator
Vent Condenser
Heater
Auxiliary Steam Jet
Exhaust Steam to Deaerator
Separator
Differential Pressure Valve
Deaerated Water to Boiler
Feed Pump

Fig. 3. Principle of Deaerator

Fig. 3 shows diagrammatically a type of deaerator in which a current of steam sweeps over the surface of the heated water. The water and steam are brought to the same temperature, which facilitates complete removal of dissolved gases (mostly free O, $CO_2$ and N). After extracting the residual gases from the water, the steam flows upwards into the heater, and gives up its heat to the incoming water. Where conditions warrant complete recovery of the heat, a condenser cooled by the incoming water to the heater, may be installed at the vent. An auxiliary steam jet brings the temperature of water leaving the heater chamber to the temperature of the steam in the separator chamber. The cycles of operation are: Water from the supply line passes through the vent condenser, and, controlled by the float valve, enters the heating chamber, where it is heated to the desired temperature by steam previously used for deaeration. The heated water then flows through the piping between heater and separator, where it comes in contact with the auxiliary steam jet. This injects into the water sufficient steam to compensate for the temperature depression of the heater, and the water enters the separator or deaerating chamber, at substantially the temperature of the steam. It cascades over the trays and is agitated in the presence of the steam. The steam sweeps through the separating chamber, deaerates the water, and flows up through the steam passage between the separator and the heater. The water drops into the lower part of the separator, and is pumped to the boilers, economizers, or service line. The gases are vented into the vent condenser.

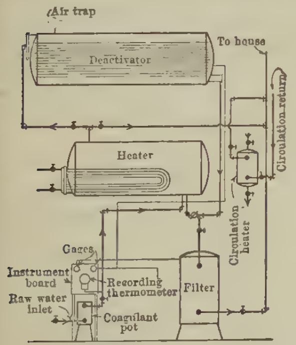

Fig. 4. Principle of Apparatus for Fixing Free Oxygen

**FIXING OF FREE OXYGEN IN WATER BY CHEMICAL COMBINATION** is effected in practice by passing heated water through a storage tank carrying a mass of expanded sheet iron, giving about 60 to 70 sq. ft. of surface per cu. ft. of space. At 170° F. the oxygen may be reduced in this way in about 1 hr. to 0.3 c.c. per liter. If, as is usual, the water carries free $CO_2$ and bicarbonates, the free $CO_2$ often is a little higher under this treatment, notwithstanding which, practically no corrosion is found. The first plant of considerable size based on this principle was built in Pittsburgh in 1915. The essential features are indicated in Fig. 4. For domestic use, the water must be filtered after treatment.

**COMBINATION OF MECHANICAL AND CHEMICAL DEAERATION.**—Deactivation (a term suggested to designate chemical deaeration process) may supplement mechanical deaeration. Such combination seems to be the most economical means for complete oxygen removal from hot water on a large scale under average conditions. Water so treated is practically inactive toward iron, and may be used in boilers or steel economizers without fear of corrosion.

Other metals in scrap form, as zinc, may be used for oxygen removal, but iron is the most economical, not only because of low first cost, but also because the residual ferrous hydrate formed renders the water slightly more alkaline. Sodium sulphite sometimes is used effectively for this purpose. In high pressure steam boiler practice, in addition to the complete removal of the oxygen, the water should be kept slightly alkaline with *p*H of about 9.

**CORROSION IN STEAM BOILERS.**—Internal corrosion may be caused by the use of water containing free acid, by water containing chloride of magnesium, which decomposes when heated, liberating acid, or by water containing air or carbonic acid in solution. External corrosion rarely takes place when a boiler is kept hot, but when cold it is apt to corrode rapidly in those portions where it joins the brickwork or where it may be covered by dust or ashes, or wherever dampness may lodge.

**CORROSION OF NON-FERROUS PIPES.**—The corrosion of brass, zinc, and other metals is greatly retarded or entirely stopped by removing oxygen from the water in contact with these metals. Brass and copper piping are sometimes seriously attacked by heated water under pressure. Water heated in a copper boiler of the usual type loses nearly one-half of its free oxygen in passing through the heater. Brass pipes after 10 or 12 years frequently have lost nearly all their zinc content. The same has been observed with alloy condenser tubes. It is now generally recognized that no one metal will meet all requirements of service. Ordinary brass pipe (30 Zn, 70 Cu) is adequate where the water is fairly pure and almost free from chlorides; red brass (15 Zn, 85 Cu) is much more durable where the water carries considerable amounts of soluble salts and is less likely to suffer dezincification. This grade is preferable for sea water. Copper pipe usually is more durable than brass for practically all domestic water service. See J. R. McDermot, Separation of Dissolved Gases from Water, *Trans.* A.S.M.E., xlii, p. 267, 1920. Also F. N. Speller, W. H. Walker, and R. G. Knowland, *Trans.* A.S.H.V.E., 1918–1920.

## 7. PROTECTION AGAINST SOIL CORROSION

**SOIL CORROSION** is highly localized and usually is caused by electrochemical action resulting from variations in the soil and solutions therein. Low alloy steels, *i.e.*, steels with less than 10% Cr do not show enough advantage, from an economic standpoint, under severe soil corrosion to warrant their use and the solution of the problem of longer life of underground pipe is practically limited to a selection of suitable protective coatings.

Soil corrosion varies considerably from point to point on pipe lines. On long lines serious damage is confined to 5% or less of the line as a rule. (W. T. Smith, Economy in Wider Use of Protective Coatings on Pipes, *Eng. News-Record*, April 21, 1932.) It, therefore, is advantageous to have a soil survey of the territory through which the pipe line will run, before deciding on the type of protective coating, if any, to be used. Much progress has been made in the study of soil factors at the U. S. Bureau of Standards, so that a careful survey, judiciously interpreted, with regard to all of the factors involved, should give an approximate idea as to the corrosivity of the soil. See U. S. Bureau of Standards Research Papers Nos. 95, 298, 329, 359, and 363. A survey, together with experience on pipe lines situated in similar soils, affords a fairly sound basis for selecting the most economical coating for a particular location.

**BITUMINOUS COATINGS** have been mostly used for underground protection. These consist of the well-known coal tar and asphalt dip coatings; preferably applied, after removing all grease, dirt, and loose mill scale, by vertically dipping the pipe in a bituminous bath at about 350° F. The melting point of the asphalt mixture should be varied between 140° and 190° to suit climatic conditions. Coatings of this type, properly applied at the mill, are suitable only for mild corrosive conditions, but afford good protection on the inside of water mains. In wet, marshy places, or in alkaline or acid soils, more substantial protection often is required. A thicker coating may be applied at the trench, but the pipe either should receive an asphalt or coal tar dip coating, or a coating of a suitable bituminous primer applied cold at the plant before shipment to prevent rusting in transit. Bituminous coatings usually are reinforced with fabric or with saturated felt wound spirally on the pipe over the dip coat. Asbestos-base reinforcement is preferable. In certain soils, unless these bituminous coatings have a high melting point and are unusually rigid, they are subject to soil stress; that is, some soils have a tendency to cling to the coating and pull it away from the pipe when the soil shrinks by drying; or irregular pressure in the trench may distort the bitumen and cause thin spots in the coating. In such soils a shielding material should be used to prevent damage to the coating by soil stress. Experience shows that many bituminous coatings that otherwise would have afforded adequate protection have been destroyed or rendered ineffective in this way. Thin, stiff, rolled steel sheet (26 gage) or copper foil may be used as a shield. Asphalt mastics consisting of 30% bitumen and 70% of a graded sand have been used with good results, especially for protection of pipe conveying hot oil through very corrosive soils. These mixtures are applied about $^3/_8$ in. thick, but are too expensive for ordinary use, costing about 60 cents per ft. on 8-in. pipe.

In soils where the electrical conductivity is high and fairly uniform, as in New Orleans, it has been found practicable to maintain the coated pipe lines at lower potential than the soil by imposing a current on the pipe to prevent the destruction of coating wherever it happens to be broken.

**PORTLAND CEMENT CONCRETE** has been used for over 40 years for protecting pipe in highly corrosive soil. It has a record as one of the most practical means of affording permanent protection. The minimum thickness usually is about 1 in., the mixture being applied in a wooden box surrounding the pipe. A portable steel form has been developed so that a concrete mixture consisting of 3 sand and 1 Portland cement can be poured around the pipe with a minimum thickness of $^5/_8$ in. Portions of the Hetch Hetchy steel conduits have been protected with such a mixture applied $^1/_2$ in. thick, wrapped with fabric. (*Construction Methods*, Dec. 1931, pp. 22–26.)

Portland cement coatings are applied in such thicknesses at about the same cost as fabric-wrapped bituminous coatings, and are highly durable in most soils, the cost of application being about 13 cents per lineal foot on 8-in. pipe. Concrete coatings offer some resistance to stray electric currents but should not be relied upon to protect pipe against electrolysis. Where pipe lines pass under electric railways, or in situations where complete electrical insulation is necessary, the line has been effectively protected by boxing and coating with hot asphalt with a minimum thickness of about 1 in. for a distance of 100 feet on each side of the track.

A comprehensive test of pipe coatings is now (1935) being conducted under the auspices of the U. S. Bureau of Standards with the co-operation of committees of the Am. Gas Assoc. and A.P.I. These tests are being made on short lengths of pipe, and also on operating oil lines. They should give useful information in a few years concerning the durability of all the material now available for such protection.

# CORROSION RESISTANT METALS*

## 1. STAINLESS STEEL

**STAINLESS STEEL.**—Alloys of iron, chromium and nickel are grouped under the general name of stainless steel. They are in three general groups. 1. Chromium-iron, with moderately high carbon. 2. Chromium-iron, with carbon under 0.12%. 3. Chromium-nickel-iron with low carbon. The various grades have been standardized by the producers of stainless steel under type numbers as shown in Table 1.

**Table 1.—Analyses of Standard Grades of Stainless Steel**

| Type No. | C, percent, max. | Cr, per-cent | Ni, per-cent | Type No. | C, per-cent, max. | Cr, percent | Ni, per-cent | Other Elements, percent | Type No. | C, per-cent, max. | Cr, per-cent | Other Elements, percent |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 302 | 0.08–.20* | 17–19 | 7–9 | 309 | 0.20 | 22–26 | 11–13 | ......... | 410 | 0.12 | 12–14 | ....... |
| 303 | 0.08–.20*† | 17–19 | 7–9 | 310 | .25 | 24–26 | 19–21 | ......... | 416 | .12 | 12–15 | ..¶... |
| 304 | 0.11 | 17–19 | 7–9 | 316 | .11 | 17–19 | 7–11 | Mo, 2–4 | 418 | .12 | 12–15 | W, 2.5–3.5 |
| 305 | 0.08–.20* | 18–20 | 8–10 | 320 | .20 | 17–19 | 7–9 | Ti = 4C | 425 | .12 | 14–16 | ....... |
| 306 | 0.11 | 18–20 | 8–10 | 321 | .20 | 18–20 | 8–10 | Ti = 4C | 430 | .12 | 16–18 | ....... |
| 307 | 0.08–.20* | 19–22 | 9–12 | 403 | .12 | 11.5–13.0 | ...... | ......... | 438 | .12 | 15–18 | W, 2.5–3.5 |
| 308 | 0.11 | 19–22 | 9–12 | 405 | .08 | 11.5–13.5 | ...... | Al, 0.10–.20‡ | 442 | .35 | 18–23 | ....... |
| ..... | ....... | ...... | ...... | 406 | .12 | 12–14 | ...... | Al, 4–4.5§ | 446 | .35 | 23–30 | ....... |

* Carbon may be specified to a 4-point range within these limits.
† Free-machining quality. Phosphorus, sulphur and selenium, min., 0.15%.
‡ Non-hardening quality. § High electrical resistance.
¶ Free machining quality; contains phosphorus, sulphur and selenium.

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* Staff contribution.

These alloys are resistant to oxidation and generally to attack by acids, salts and organic substances, with the exception of carbolic, hydrochloric, hydrofluoric, phosphoric and dilute sulphuric acids, bleaching powder, the calcium, sodium and stannous hypochlorites, and the copper, ferric and mercuric chlorides. They also are subject to attack by wet or dry chlorine gas and molten zinc. The resistance to corrosion and attack of the various alloys by various reagents differ, and manufacturers should be consulted regarding the particular alloy for a given purpose.

A wide range of physical properties may be obtained with these alloys. The 18 Cr, 8 Ni group has a modulus of elasticity of (annealed) 28,600,000 which may be reduced to 22,000,000 by cold working. That of the 22 Cr, 11 Ni group is 29,000,000 annealed, and 22,000,000 cold worked. The modulus of shear of the two groups is respectively 11.9 and 11.2 million. Tensile properties at elevated temperatures are given in Table 2. Other physical properties are given in Table 3.

The effect of annealing on the physical properties of alloys containing nickel is given in Table 4. The effect of hardening and tempering at various temperatures of alloys containing no nickel, is given in Table 5. Creep stress is given in Table 6 and the fields of service or uses of the several alloys are given in Table 7.

The characteristics of the various groups vary widely in their reaction to heat treatment. In the first two groups, the steels with up to 18% Cr harden when cooled rapidly

**Table 2.—Tensile Properties of Stainless Steels at Elevated Temperatures**

(Average Values)

| | Temp., deg. F. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 100 | 200 | 500 | 1000 | 1200 | 1400 | 1600 | 1800 | 2000 |
| | Cr, 18–20; Ni, 8–10; C, 0.08–0.20. | | | | | | | | |
| Ultimate Strength, 1000 lb. per sq. in. | 89.4 | 82.4 | 74.4 | 70.6 | 64.5 | 50.2 | 30.0 | ..... | ..... |
| Proportional Limit, 1000 lb. per sq. in. | 28.0 | 28.0 | 26.9 | 25.2 | 23.6 | 18.0 | ...... | ..... | ..... |
| Elongation in 2 in., % | 61.0 | 57.4 | 49.2 | 46.2 | 42.0 | 34.1 | 34.6 | ..... | ..... |
| Reduction of Area, % | 75.2 | 75.8 | 71.1 | 70.2 | 69.9 | 54.8 | 55.5 | ..... | ..... |
| | Cr, 22–26; Ni, 11–13; C, 0.20. | | | | | | | | |
| Ultimate Strength, 1000 lb. per sq. in. | 98.8 | 92.4 | 85.0 | 77.0 | 69.4 | 56.4 | 36.0 | 20.9 | 13.4 |
| Elongation in 2 in., % | 59.4 | 57.2 | 49.2 | 46.2 | 35.1 | 24.8 | 17.1 | 22.6 | 33.4 |
| Reduction of Area, % | 68.2 | 68.1 | 67.3 | 57.5 | 42.2 | 30.9 | 24.6 | 39.0 | 46.4 |
| | Cr, 12–15; C, 0.12. | | | | | | | | |
| Ultimate Strength, 1000 lb. per sq. in. | 75.0 | 75.1 | 70.0 | 30.3 | 23.2 | 11.5 | 9.6 | 7.8 | 4.3 |
| Elongation in 2 in., % | 45.8 | 44.9 | 37.0 | 46.0 | 40.0 | 39.0 | 42.0 | 60.0 | 64.0 |
| Reduction of Area, % | 70.6 | 69.7 | 71.9 | 85.4 | 88.3 | 94.1 | 90.4 | 64.9 | 93.1 |
| | Cr, 16–18; C, 0.12. | | | | | | | | |
| Ultimate Strength, 1000 lb. per sq. in. | 84.5 | 82.6 | 76.5 | 47.5 | 30.0 | 18.1 | 9.6 | 4.0 | ..... |
| | Cr, 23–30; C, 0.25. | | | | | | | | |
| Ultimate Strength, 1000 lb. per sq. in. | ..... | 116.0 | 102.2 | 84.5 | 43.1 | 24.3 | 14.4 | 7.3 | 5.5 |

**Table 3.—Physical Properties of Stainless Steels**

(Average Values)

| | Cr, 18–20; Ni, 8–10; C, 0.08–.20 | Cr, 22–26; Ni, 11–13; C, 0.20 | Cr, 12–15; C, 0.12 | Cr, 16–18; C, 0.12 | Cr, 23–30; C, 0.35 |
|---|---|---|---|---|---|
| Specific gravity | 7.97–8.07*<br>7.86–7.94† | 7.86–794 | .......... | 7.65–7.75 | 7.58–7.62 |
| Average weight, lb. per cu. in. | 0.2985*<br>0.285† | 0.285 | .......... | 2.78 | 2.74 |
| Melting point, deg. F. | 2606–2679 | 2552–2597 | 2723 | 2714 | |
| Thermal conductivity, B.t.u. per sq. ft. per 1 in. thick per hr. per 1°F., 68-212°F | 200.04 | 113.22 | 278.69 | 238.05 | 171.28 |
| Specific heat | .......... | 0.142 | 0.159 | .......... | .......... |
| Coefficient of linear expansion, per deg. F. × $10^{-5}$: | | | | | |
| at 54–212° F. | 0.961 | 0.900 | 0.567 | 0.583 | 0.567 |
| at 54–1832° F. | 1.112 | 1.112 | 0.711 | 0.733 | 0.744 |
| Electrical resistivity, microhms per cu. cm.: | | | | | |
| at 400° F. | 92 | 92 | 78 | 83 | 88 |
| at 1200° F. | 117 | 115 | 109 | 115 | 116 |

* Cold-rolled, annealed. † Hot-rolled, annealed.

from above 1500° F. With Cr below 15%, they harden intensely when air-cooled from above 1500° F. Depending on the Cr-C combination, the Brinell hardness may be as high as 600. With Cr from 15 to 18%, the Brinell hardness, in the lower carbon ranges, may be as high as 300, but for the high-carbon steels, *i.e.*, steels containing 0.70 carbon, they may run as high as 600 Brinell. Steels with Cr above 18% harden to a much smaller extent, depending on the carbon content. Because of this air-hardening, these steels, when rapidly cooled from 1500° F. are not highly resistant to shock, and annealing or tempering usually is necessary to improve their shock resistance. Steels with Cr over 20% do not harden when rapidly cooled. Annealing has little effect on their properties. When subjected to long-time heating above 1750° F., they are brittle when cold. As this brittleness is due to the large grain formed, it cannot be corrected by heat treatment. When cooled slowly from above 1200° F. they are brittle when cold, but this brittleness may be removed by heating to above 1200° F. and cooling rapidly by water quenching. While steels of this composition are classified as brittle, and will break if subjected to violent

**Table 4.—Effect of Annealing on Physical Properties of Chromium-nickel-iron Alloys**

Standard 0.505 in. diameter test bars used. Hot-rolled bars were held 1 hr. at the annealing temperature designated

| Temperature, deg. F. | Cr, 18–20; Ni, 8–10; C, 0.08–.20 | | | | | Cr, 22–26; Ni, 11–13; C, 0.20 | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | Elongation in 2 in., per-cent | Reduction of Area, per-cent | Brinell Hardness | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | Elongation in 2 in., per-cent | Reduction of Area, per-cent | Brinell Hardness |
| Hot rolled..... | 110.9 | 73.9 | 43.0 | 65.9 | 223 | 130.5 | 107.3 | 31.0 | 59.8 | 262 |
| 1600.......... | ........ | ....... | ....... | ....... | ...... | 122.9 | 88.4 | 32.0 | 55.3 | 234 |
| 1700.......... | 107.1 | 54.4 | 48.0 | 56.0 | 217 | 120.5 | 89.8 | 34.0 | 56.7 | 229 |
| 1800.......... | 106.8 | 54.9 | 43.0 | 62.8 | 207 | 109.9 | 59.9 | 44.0 | 59.8 | 187 |
| 1900.......... | 93.6 | 30.6 | 58.5 | 67.6 | 165 | 107.3 | 56.6 | 49.0 | 62.2 | 179 |
| 2000.......... | 90.4 | 28.5 | 61.0 | 68.2 | 135 | 100.3 | 49.4 | 58.0 | 70.0 | 149 |
| 2100.......... | ........ | ....... | ....... | ....... | ...... | 95.5 | 45.8 | 62.0 | 69.0 | 143 |

**Table 5.—Effect of Hardening and Tempering on Chromium-iron Alloys**

| Condition | Deg. F. | Cr, 16–18; C, 0.12 | | | | | Cr, 11.5–13; C, max., 12* | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | Elongation in 2 in., per-cent | Reduction of Area, per-cent | Brinell Hardness | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | Elongation in 2 in., per-cent | Reduction of Area, per-cent | Brinell Hardness |
| Oil quench..... | 1800 | ........ | ...... | ...... | ...... | ...... | 178.8 | 161.5 | 15.0 | 51.0 | 364 |
| Hot rolled...... | ...... | 163.4 | 88.0 | 10.0 | 19.6 | 312 | ........ | ...... | ...... | ...... | ..... |
| Tempered...... | 900 | 157.4 | 84.0 | 21.5 | 51.1 | 286 | 178.8 | 161.6 | 18.0 | 51.9 | 364 |
| " ...... | 1000 | 114.5 | 82.1 | 22.0 | 62.9 | 295 | 144.9 | 133.1 | 19.0 | 64.0 | 302 |
| " ...... | 1100 | 108.5 | 81.1 | 24.5 | 61.3 | 228 | 110.6 | 91.1 | 23.5 | 69.6 | 228 |
| " ...... | 1200 | 99.0 | 74.6 | 26.0 | 61.6 | 183 | 98.8 | 69.1 | 25.5 | 73.3 | 207 |
| " ...... | 1300 | 96.7 | 63.4 | 27.5 | 64.5 | 173 | 89.4 | 60.1 | 29.5 | 75.0 | 187 |
| " ...... | 1400 | 84.9 | 53.8 | 31.5 | 71.0 | 170 | 83.9 | 54.9 | 31.5 | 74.2 | 166 |
| " ...... | 1500 | ........ | ...... | ...... | ...... | ...... | 85.0 | 55.4 | 31.0 | 72.7 | 179 |
| Annealed...... | 1500 | ........ | ...... | ...... | ...... | ...... | 73.0 | 40.6 | 37.0 | 78.7 | 153 |
| Slow cool...... | 1600 | 71.6 | 41.8 | 33.5 | 71.0 | 143 | ........ | ...... | ...... | ...... | ..... |

* All samples were annealed before heat treatment; 1 in. diameter hot-rolled test bars used. These data were obtained by drawing back samples which had been quenched from 1800° F. at temperatures indicated.

**Table 6.—Creep Stresses in Chromium-nickel-iron Alloys**

Stress necessary to give an elongation of 1% in 10,000 hr. or 100,000 hr.

| Temperature, deg. F. | Cr, 18–20; Ni, 8–10; C, 0.8–.20 | | Cr, 22–26; Ni, 11–13; C, 0.20 | | Cr, 12–15; C, 0.12 | | Cr, 16–18; C, 0.25 | |
|---|---|---|---|---|---|---|---|---|
| | 10,000 hr. | 100,000 hr. | 10,000 hr. | 100,000 hr. | 10,000 hr. | 100,000 hr. | 10,000 hr. | 100,000 hr. |
| | Stress, 1000 lb. per sq. in. | | | | | | | |
| 800 | 33.4 | 24.0 | 36.9 | 25.8 | ......... | ......... | ......... | ......... |
| 900 | 20.4 | 14.9 | 23.0 | 16.3 | 11.8 | 9.0 | ......... | ......... |
| 1000 | 13.8 | 9.8 | 15.3 | 10.4 | 5.5 | 4.4 | 5.38 | 4.15 |
| 1100 | 9.1 | 6.2 | 10.3 | 7.0 | 2.6 | 2.0 | 2.35 | 1.85 |
| 1200 | 6.0 | 4.0 | 7.0 | 4.5 | 1.4 | 0.95 | 1.25 | 0.92 |
| 1300 | 3.6 | 2.2 | 4.4 | 2.7 | 0.73 | .55 | 0.64 | .39 |
| 1400 | 2.0 | 1.1 | 2.45 | 1.35 | ......... | ......... | 0.39 | .20 |
| 1500 | 0.9 | 0.65 | 1.0 | 0.75 | ......... | ......... | ......... | ......... |

### Table 7.—Fields of Service of the Various Chromium-iron-nickel Alloys

In the following table, the significance of the identifying letters is as follows: A = Cr, 18–20; Ni, 8–10; C, 0.08–0.20. B = Cr, 22–26; Ni, 11–13; C, 0.20. C = Cr, 12–15; C, 0.12. D = Cr, 23–30; C, 0.25. E = Cr, 16–18; C, 0.12.

| Service or Quality | Alloy | Service or Quality | Alloy |
|---|---|---|---|
| Abrasion resistance | C | Glass molds | D |
| Acid mine water | C, D | High temp., oxidation resistance | C, D, E |
| Atmospheric corrosion | A, C, E | High temp., high tensile | B |
| Automotive parts | C | Industrial oven linings | B |
| Blowers | C, E | Internal combustion engine parts | C |
| Boiler baffles | B, D | Kiln linings | B, D |
| Building trim | A | Laundry machinery | A |
| Cannery equipment | A | Low temp. furnace parts | E |
| Chemical plant equipment | A, C, E | Machinability | C, D, E |
| Condensers | C, E | Malleability | B |
| Cooking equipment | A | Nitric acid plants | E |
| Creamery equipment | A | Oil still tube supports | D, E |
| Creep stress | B | Pump parts | B |
| Dairy equipment | A | Pyrometer protection tubes | D |
| Drawing and forming | A, C, E | Rabble arms (roaster furnace) | D |
| Evaporators | C, E | Recuperator parts | D |
| Fans | C, E | Roofing and siding | C, E |
| Firebox sheets | B | Stack dampers | B, D |
| Food mfg. and processing | A | Steam engine parts | C, E |
| Forming operations | C, E | Still tube supports | B, D |
| Furnace dampers | B, D | Tanks | C |
| Furnace parts | B, D | | |
| " " low temp. | E | | |

shock when cold, they still are sufficiently ductile to withstand normal handling operations.

Steels of the third group, when subjected to temperatures between 800° F. and 1500° F., are less resistant to corrosion. Under severe corrosion conditions they gradually will lose ductility and become brittle if the time of heating is sufficiently long. The rate at which this change takes place depends on the composition. When the time of heating is short, as in a welding operation, the selection of steels with a low carbon content will prevent a change sufficient to render the steels susceptible to attack. The same result may be obtained by increasing the chromium or nickel content. If, however, the steels are to be subjected to this temperature range for a long time, the addition of titanium or columbium in sufficient quantity will render them immune to intergranular corrosion. It has not been found possible to deposit titanium in weld metal. Consequently, if the titanium-bearing steels are welded, the weld metal itself will be subject to intergranular corrosion when the steels are heated in the temperature range 800–1500° F. and subjected to severe corrosive conditions. This change, which occurs in these steels after exposure to the temperature range 800–1500° F. for a long period of time, can be corrected by subsequent annealing at about 1900° F., followed by rapid cooling. If maximum ductility is required, an annealing temperature of 2100° F. is better, but large grain is apt to form unless the time of high temperature application is held at a minimum. See articles by C. A. Scharschu, *Metal Progress*, May, June and August, 1931.

**FABRICATION OF STAINLESS STEEL.**—The following notes on fabrication apply to all varieties of the stainless steels excepting as noted. Manufacturers should be consulted concerning special conditions.

**Welding.**—Stainless steels can be welded electrically with the metallic arc, the carbon arc, by the resistance method, the atomic hydrogen method, and can be seam-, butt-, spot- or line-welded. They also can be welded by the oxy-acetylene flame. Electric welding is preferred.

For oxy-acetylene welding, tips one or two sizes smaller than used for steel of the same gage are used. Uncoated rod of the same composition as the parts to be welded is necessary. The flame should be slightly reducing, cover the weld at all times, and be pointed in the direction of the weld. The weld should not be puddled. The parts should be preheated to 300–400° F.

Metallic arc direct-current welding requires that the work be negative and the welding rod positive. The welding rod should be low-carbon coated with a non-carbonaceous flux, and with Cr and Ni high enough to compensate for evaporative losses, so that deposited metal is of the same composition as the work. Gages heavier than No. 14 should be scarfed. A medium short arc should be used. If the weld is to be ground a light bead should be flowed on the side of the work to be ground and a heavy bead on the opposite side. The weld should be in the direction of current flow and should not be puddled.

The oxide should be removed from previously deposited metal before striking the arc.

Metallic arc alternating-current welding is used only on No. 16 gage work and lighter. The technique is the same as for direct-current welding.

Resistance welding requires heavier pressure but lower power input than is necessary with steel of the same gage. The metal must be clean and free from grease and have good contact.

After welding, the bead is removed by rough grinding with a rubber bond wheel. The temperature of the metal must not be raised above 400° F. After grinding the joint should be polished.

**Annealing.**—The procedure varies with the character of the metal. Heating should be done in a muffle or semi-muffle furnace. The Cr-Ni steels should be annealed at from 1850–2000° F. Sections $^1/_4$ to $^1/_2$ in. thick should be held at 1850° F. for 30 min., or at 1950° F. or over from 3 to 5 min., followed by rapid cooling in air. Sections $^1/_4$ in. thick or over may require an air blast or water quench. The work should be cooled below red heat in less than 3 min. The steels of Types (see Table 1) 403, 410, 416, should be heated slowly to 1450–1500° F. Steels of Type 430 should be heated slowly to 1550–1600° F. A slightly reducing atmosphere should be maintained in the furnace. The work should be cooled in the furnace at the rate of 50° F. per hr. If it is air-cooled, the maximum temperature should not exceed 1450° F.

**Pickling.**—For the Ni-Cr steels the following solutions are used. 1. 38° Baumé nitric acid, 25%; hydrofluoric acid, 2%. 2. Nitric acid, 20%; hydrochloric acid, 2%. 3. 20° Baumé muriatic acid, 25%. All by volume. 4. 60° Baumé sulphuric acid, 10%; rock salt, 10% (by weight). Solution No. 1 gives a white pickle. Solution No. 2 gives a high luster finish, but attacks the metal more than No. 1. Solution No. 4 gives the ordinary white finish. Solution should be heated to 140–180° F. Either or both bath and work should be agitated, and the loose oxide scrubbed off. After pickling wash the work in hot water.

The steels without Ni should be pickled in a solution of 10% muriatic acid by volume, heated to 140 to 160° F. Immersing in a 5 to 10% solution of nitric acid at 140–160° F. produces a white finish. The work should be washed in hot water after pickling.

**Shearing.**—About twice the energy necessary for mild steel is required. The knives must be sharp and closely adjusted, or the steel will be drawn over the bottom knife and work-hardened. The Cr-Ni steels must be cut all the way through, while the steels without Ni do not work-harden and will break after being cut through 50–75%.

**Blanking and Punching.**—For the Cr-Ni steels a neat clearance between punch and die is necessary or the metal will work-harden. About twice the power necessary for mild steel is required. For steels without Ni, the same practice as used for mild steel is recommended, but more energy is required.

**Forming and Drawing.**—The best drawing compound is white lead, thinned with linseed oil to about the consistency of 600W engine oil. Slow press speeds are necessary. For the Cr-Ni steels the clearance between punch and die should be greater than for mild steel, and a larger radius on the draw ring is advisable. Rough dies will cause trouble. The work should be annealed after drawing, being first cleaned thoroughly of oil and grease. For cleaning, sodium-metasilicate, at 160–200° F., or the electrolytic process, should be used.

The steels without Ni have about the same ductility as deep drawing steel. They require heavier backing and heavier die sections than steel, but the die clearances and speeds are the same as for steel. Steel of Type 446 is not suited to deep draws.

**Drilling.**—High-speed steel drills, ground slightly flatter than standard, should be used. A triangular-nose center punch should be used, the center punch mark being as shallow as possible to avoid work-hardening. Continuous pressure on the drill is necessary. The work should be backed up with a cast-iron backing plate, so that the drill will cut all the way through. Cutting speed should be about one-half of that for mild steel. The drill should be immersed in water after each hole. For deep holes, a compound of 1 lb. sulphur and 1 gal. of lard oil is advisable.

**Sawing.**—For the Cr-Ni steels, use a high-speed steel blade with wavy teeth. For light gages, use saws with 32 teeth per 1 in., with fewer teeth for heavier gages. The speeds recommended by saw makers should not be exceeded. For the steels without Ni, use the same methods as for 0.30–.40 carbon steel.

**Machining.**—For the Cr-Ni steels, speeds about one-half of those used for mild steel are best. Tool clearances should be about the same as for dead soft brass. The tool must cut at all times or the metal will work-harden. Taps should be ground as for dead soft brass, with slightly more taper and with but two or three full teeth. The best lubricant is water, 40 gal., lard oil, 10 gal., soda ash, 2.5 lb. For automatic machines, metal of 200–240 Brinell is best.

For the steels without Ni, the same practice as above is used with Types 430 and 446. For steels of Types 403, 410, 416, the same design of tool as used for 0.30–.40 carbon steel is used. These steels machine best at 220 Brinell. Slightly slower speeds are used than for steel.

**Spinning.**—For the Cr-Ni steels, speeds about one-half those used for spinning copper are advisable. Much greater power is required and the ordinary spinning lathes usually have insufficient belt pull. The best lubricant is white lead and linseed oil mixed to the consistency of a thick paste. The tool should be hardened steel, of the same design as for copper, except that it should be somewhat flatter. Frequent annealing of the work is necessary. For steels without Ni, the speed should be about two-thirds of those required for mild steel. The work requires annealing about one-third oftener than does steel.

**Riveting.**—Cold riveting is recommended for sizes less than $^3/_{16}$ in. The rivets should be set up in one blow. With the Cr-Ni steels, hot riveting should be done at temperatures of 1900–2100° F., and finished above 1600° F. Work exposed to corrosive conditions should be annealed after riveting. The steels without Ni should be hot riveted at the following temperatures: Types 403, 410, 416, below 1450° F.; type 430, below 1550° F.; Type 446, not over 1650° F.

**Brazing** is not recommended for any type of stainless steel.

**Soldering.**—Soft soldering should be avoided if possible. Silver soldering can be used for attaching steel to copper or steel to steel. For silver soldering the Cr-Ni steels to copper, use a flux of potassium acid fluoride and borax, and direct the flame on the copper. For soldering steel to steel, use the same flux and preheat slightly the areas adjacent to the joint. The flux should be removed immediately after soldering by a high-pressure steam jet. For steels without Ni, a solder should be used whose melting point is such that soldering can be done below 1500° F.

**Forging.**—The precautions usual in forging high-carbon steels should be observed. Forging of the Cr-Ni steels may be conducted at temperatures of 2150° down to 1700° F. These steels are stronger at high temperatures and harder to work. If a finished or semi-finished forging is to be pickled, it first should be annealed, otherwise surface cracks will develop.

The following temperatures are recommended for the steels without Ni: Types 403, 410, 416, 430, 2050–1600° F.; Type 446, 1950–1600° F. They should be slowly and uniformly heated to avoid rupture during heating. Forged bars at room temperature, which have not been annealed, should not be placed in furnaces whose temperatures exceeds 600° F. Special precautions should be taken to insure uniform cooling.

## 2. HIGH-SILICON ALLOYS

**CORROSIRON** is cast iron containing about 14.25% silicon, and other metalloids in small quantities. It melts at about 2400° F., and weighs about 0.263 lb. per cu. in. It is brittle with long flat crystals. A bar 1 × 2 in., 24 in. long, has a transverse strength of about 1200 lb., with the load applied at the center, the deflection being about 0.16 in. Shrinkage is about $^3/_{16}$ in. per ft. In designing, flat surfaces should be avoided, all corners rounded, and sections of metal kept as even as possible. The acid-resisting qualities of corrosiron are shown in Table 8. Corrosiron is used in the construction of pipes, valves, pipe fittings, acid pumps, pans, etc. It is made by the Pacific Foundry Co., San Francisco.

**Table 8.—Acid-resisting Properties of Corrosiron**

| Corrosive Agent | Concentration, percent | Depth of Corrosion, in. per yr. | Corrosive Agent | Concentration percent | Depth of Corrosion, in. per yr. |
|---|---|---|---|---|---|
| Acids | | | Acids (*cont.*) | | |
| Acetic | 90 | 0.0000048 | Phosphoric | 87 | 0.000019 |
| Hydrochloric | 5 | .00145 | “ | 25 | .000012 |
| Nitric | 70 | nil | “ | 10 | .000023 |
| “ | 25 | nil | Sulphuric | 95 | .000021 |
| “ | 10 | nil | “ | 2 | .0000007 |
| Oleic (commercial) | .......... | nil | “ | 10 | .000017 |
| Oxalic | 7 | 0.0000234 | Ammonium chloride | 27 | .000018 |
| “ | 2.1 | .0000306 | Copper sulphate | 25 | .00001 |
| Picric | 9.1 | .000005 | Ferric chloride | 48 | .00414 |

**DURIRON** is a corrosion-resisting material made by the Duriron Co., Dayton, O., consisting of Si, 14.50%; Mn., 0.35%; C, 0.80 to 0.85%; Fe, balance. The following are given as its physical properties: Specific gravity, 7.0; weight per cu. in., 0.253 lb.; melting range, 2275 to 2375° F.; coefficient of expansion, 0.000 015 65 per deg. F., from room temperature to the melting point, and 0.000 004 per deg. F., from room temperature to 212° F.; electrical resistivity, microhms per cu cm. at 32° F., 63.3; thermal conductivity (silver = 1.000), 0.125; Shore scleroscope hardness, 49–51; contraction allowance in casting, 1/4 in. per ft. The makers give the tensile strength as 15,000–20,000 lb. per sq. in., and the compressive strength as 95,000 lb. per sq. in. Duriron will not soften or materially alter its shape at temperatures below 2000° F., nor does it oxidize at temperatures below 1500° F. It is highly resistant to erosion. Its corrosion resistance properties are given in Table 9, supplied by the makers.

**Table 9.—Corrosion-resisting Properties of Duriron**

| Corrosive Agent | Concentration, percent | Temperature, deg. F. | Depth of Corrosion, in. per yr. | Corrosive Agent | Concentration, percent | Temperature, deg. F. | Depth of Corrosion, in. per yr. |
|---|---|---|---|---|---|---|---|
| Acids | | | | Acids (*cont.*) | | | |
| Acetic | 10 | 68 | 0.002 | Sulphuric (*cont.*) | conc. | 176 | nil |
| " | Glacial | 176 | .003 | " | 30 | Boiling | 0.070 |
| Benzoic | 100 | 68 | .003 | " | 35 | " | .080 |
| Butyric | 98 | 68 | .002 | " | 40 | " | .060 |
| Formic | 85 | 68 | nil | " | 77.7 | " | nil |
| Hydrochloric | 10 | 68 | 0.007 | " | 90 | 437 | nil |
| " | 25 | 68 | .027 | " | 30 Fuming | 68 | 0.130 |
| " | conc. | 68 | .036 | " + 1 1/2 % ferric sulphate | 4 | 176 | .021 |
| " | 5 | 178 | .079 | " + 1 1/2 % ferric sulphate | 10 | 68 | .001 |
| " | 10 | 176 | .162 | Tartaric | 55 | 68 | .001 |
| " | 15 | 176 | .267 | Ammonium carbonate | 20 | 68 | .005 |
| Hydrocyanic | 2 | 68 | .001 | " chloride | 43.5 | 203 | .003 |
| Hydrofluoric | 10 | 68 | 1.521 | " hydroxide | 17. | 68 | .002 |
| Lactic | 98 | 68 | nil | Calcium chloride | 75 | 212 | nil |
| Nitric | ........ | 68 | nil | " sulphite | 11 | 68 | nil |
| Oxalic | 30 | 185 | 0.013 | Coal mine water | .......... | 176 | 0.013 |
| Phosphoric (crude) | ........ | 203 | .004 | Copper sulphate | 17 | 68 | nil |
| Sulphuric | 10 | 176 | .030 | Ferric chloride | 10 | 68 | 0.001 |
| " | 25 | 176 | .038 | Mono-calcium phosphate slurry | .......... | 203 | .004 |
| " | 50 | 176 | .040 | Sodium chloride | 16 | 176 | nil |
| " | 60 | 176 | .014 | " hydroxide | 26 | 203 | 0.005 |
| " | 77.7 | 176 | nil | | | | |

## 3. NON-FERROUS ALLOYS

**COPPER-ALUMINUM-IRON BRONZE.**—Tests on copper-aluminum-iron bronze as an acid-resisting material, made at the Univ. of Wisconsin, were reported by O. L. Kowalke (*Chem. & Met. Engg.*, Jan. 7, 1920). Two bronzes of the following composition were tested:

| | Copper | Aluminum | Iron Alloy | Specific Gravity |
|---|---|---|---|---|
| Bronze A | 87.0 | 9.8 | 3.14 | 7.71 |
| Bronze B | 85.6 | 10.81 | 3.57 | 7.58 |

Bronze A is resistant particularly to sulphuric, lactic, phosphoric, tartaric and acetic acids and to caustic soda. Nitric acid attacks it readily, and hydrochloric acid and aqua ammonia both attack it too readily for commercial purposes. Bronze B is more resistant to sulphuric acid than Bronze A. It is less resistant to hydrochloric acid and more resistant to aqua ammonia than Bronze A, and also is attacked readily by nitric acid. The loss in weight, in grams per sq. in. at the end of 28 days of immersion was as follows:

| | BRONZE A | | | | BRONZE B | | |
|---|---|---|---|---|---|---|---|
| Temp. deg. F. | 72 | 72 | 72 | 194 | 85 | 85 | 194 |
| Strength of acid | 50% | 35% | 10% | 35% | 50% | 35% | 10% |
| Sulphuric acid | 0.0246 | 0.0177 | 0.0365 | 0.2520 | 0.0024 | 0.0044 | 0.0120 |
| Hydrochloric acid | ...... | 0.781* | 1.90 | ...... | ...... | ...... | ...... |
| Ammonia | 0.700 | 0.850 | 0.413* | ...... | 0.3810 | 0.3406 | 0.2237* |
| Caustic soda | 0.0007 | 0.0016 | 0.0055 | 0.0020† | ...... | 0.0022 | 0.0072 |
| Lactic acid | 0.0111 | 0.0132 | 0.0168 | 0.1550 | ...... | ...... | ...... |
| Phosphoric acid | 0.0114 | 0.0181 | 0.0281 | 0.0395 | ...... | ...... | ...... |
| Tartaric acid | 0.0028 | 0.0046 | 0.0077 | 0.0155 | ...... | ...... | ...... |
| Acetic acid | 0.0386 | 0.0312 | 0.0221 | 0.0713 | 0.0689 | 0.0442 | 0.0334 |

* At end of 21 days. † Gain due to formation of a film.

The corrosion of forged specimens of Bronze A in 35% sulphuric acid was only about one-half that of cast or rolled specimens.

**ILLIUM** is an alloy of nickel, chromium, and copper, developed by the Burgess-Parr Co., with the following average physical characteristics. Tensile strength, 60,000 lb. per sq. in.; elastic limit, 50,000 lb. per sq. in.; Brinell hardness, 170–200; melting point 2372° F.; specific gravity, 8.3; weight per cu. in. 0.3 lb.; specific heat 0.105 cal. per deg. C.; coefficient of expansion per deg. C. (20–300°), 0.0000135; electrical resistance 121.9 microhms per $cm.^3$; contraction in casting, $^5/_{16}$ in. per ft. Illium is readily machined, being slightly harder than cast steel, and may be welded to itself or other metals by acetylene or electric arc. It is resistant to the following: *Acids.*—Acetic, butyric, citric, formic, hydrocyanic, lactic, nitric, oleic, oxalic, phosphoric, picric, sulphuric. *Salts.*—Ammonium chlorate and sulphate, copper sulphate, ferric chloride, mercuric chloride, potassium alum, silver nitrate, sodium chloride, and hydroxide, hypochlorite and nitrite. *Organic Substances.*—Acid mine water, apple vinegar, carbolic acid, formaldehyde, fruit juices, malt vinegar. *Miscellaneous.*—Bleaching powder, bromide solution, carbonated water, hydrogen sulphide, sulphur dioxide. See *Product Engg.*, Nov. 1932.

# Section 4

# NON-FERROUS METALS AND ALLOYS

## GENERAL PROPERTIES OF METALS AND ALLOYS

By Paul D. Merica
Revised by Norman B. Pilling

## COPPER

By W. H. Bassett

## ZINC AND ITS ALLOYS

By W. M. Peirce

## BRASS AND BRONZE

## SAND-CAST, COPPER-BASE ALLOYS

By W. M. Corse

## WROUGHT BRASSES, BRONZES AND NICKEL-SILVER ALLOYS

By W. H. Bassett

## ALUMINUM AND ITS ALLOYS

By Zay Jeffries

## NICKEL AND ITS NON-FERROUS ALLOYS

By Robert Worthington

## MAGNESIUM AND ITS ALLOYS

By John A. Gann

## TUNGSTEN AND MOLYBDENUM

By Zay Jeffries

## DIE-CASTING ALLOYS

By J. C. Fox

## BEARING METALS

By Christopher H. Bierbaum

## SOLDERS

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# NON-FERROUS METALS AND ALLOYS

## GENERAL PROPERTIES OF METALS AND ALLOYS

By Paul D. Merica

Revised by Norman B. Pilling

**References.**—*General:* Introduction to Physical Metallurgy, W. Rosenhain, D. Van Nostrand Co.; Metallography (parts 1 and 2), S. L. Hoyt, McGraw-Hill Co.; Textbook of Metallography, G. Tammann, Chemical Catalog Co.; Metallography, C. H. Desch, Longmans, Green and Co.; Engineering Non-ferrous Metals and Alloys, L. Aitchison and W. R. Barclay, Henry Frowde; Science of Metals, Zay Jeffries and R. S. Archer, McGraw-Hill; *Specialized Journal*, British Inst. of Metals, 1909 to date; *Trans.* A.I.M. & M.E.; *Proc.* A.S.T.M.; National Metals Handbook, A.S.M. Publications of the U. S. Bureau of Standards (Circulars, Technologic Papers, Scientific Papers, Journal); *Equilibrium diagrams:* Landolt-Bornstein Tables (Physikalisch-Chemische Tabelen), vol. I, p. 527; Guertler's Handbüch der Metallographie; International Critical Tables, vol. II, p. 400.

**Structure Metallography.**—All metals and alloys are composed of an aggregate of grains, or crystals, the latter being in general *allotriomorphic*; *i.e.*, of internal crystalline nature but bounded by surfaces having no necessary relation to the crystalline structure. These grains vary greatly in size in different materials and under different treatments; grains have been observed as large as 1 in. and as small as 0.0001 in. in mean diameter. The grain structure of a metal, or alloy, may be revealed by polishing and etching with acid or other reagent, or in its fractured surface, and usually is observed under the microscope. It is now generally considered that these grains are bound or cemented together at the boundaries by extremely thin films of so-called amorphous metal; *i.e.*, of the same composition as the grains themselves but not crystallized or *anisotropic.*

The metal comprising the grain consists of atoms arranged upon a regular pattern known as *space lattice.* Most of the workable metals have *cubic* or *hexagonal* lattices; hard and brittle metallic crystals frequently have lattices of great complexity. Many unique properties of metals originate from this symmetrical arrangement of atoms in planes. Atomic spacings in metals, or *lattice constants*, range from 10 to 22 billionths of an inch. This *fine structure* of metals is revealed and studied only with x-rays or other short radiations.

**PHASE RULE.**—The metals or chemical metallic elements consist of grains of one kind only; namely, those of the metal. Alloys may consist of one kind, or of a mixture of several kinds, of grains of different chemical composition and crystalline structure. *The phase rule* is a physical-chemical law which governs absolutely the number of simultaneously existent different kinds of crystal in alloys (when in equilibrium) and its relation to the chemical composition and temperature of the alloy. This rule as applied to alloys is:

$$C - P + 1 = F,$$

where $C$ is the number of constituent metals, $P$ is the number of constituent *phases*, or kinds of grain or crystal (liquid phases also are recognized), and $F$ is the number of *degrees of freedom* in variation of composition or temperature.

**THE EQUILIBRIUM DIAGRAM** of an alloy series describes graphically the equilibrium condition of the alloy of any composition and at any temperature; it is a temperature-composition graph based on observed facts, which, however, are related according to the phase rule. Figs. 1 and 2 show two types of equilibrium diagrams commonly encountered in binary alloy series; *i.e.*, alloys composed of two metals only. From these diagrams the course of solidification from the liquid, or molten state, or of other crystallization may accurately be followed. Two metals may be completely soluble in each other, *i.e.*, in all proportions, either in the molten or liquid state or in the solid state, or they may have limited solubility. In the former case only one *liquid phase* or solid phase is formed, no matter what the proportions of the constituent metals; in the latter case an excess in relative amount of either metal over that which is soluble in the other will give rise to the formation of two phases or kinds of crystal which will coexist in equilibrium. The addition of one metal to another in the liquid state depresses the melting or solidification temperature, if it is soluble in the liquid state and not (appreciably) soluble in the solid state; if soluble in the liquid and solid state the temperature of solidification of the resulting alloy may either be depressed or raised.

On the equilibrium diagram the *liquidus* curve is the locus of temperatures at which solidification normally begins upon cooling the molten alloy. The *solidus* curve is the locus of temperatures at which melting normally begins upon heating the solid alloy. These curves are conjugate ones and pairs of points on the two curves connected by a horizontal line of equal temperature represent respectively the compositions of liquid phase and solid crystal in equilibrium. (See *l* and *s*, respectively, Fig. 1.) The composition of the crystal first formed upon solidification from a molten alloy is not, in general, the same as that of the liquid from which it crystallizes. It is due to this fact that *cored* or *dendritic* structures always are observed in cast alloys, not annealed or otherwise treated. Upon cooling and solidification the crystals first separating are richer in the constituent metal whose melting point is higher than that of the liquid. The process of setting or solidifying is too rapid to permit of equalization of composition through diffusion; consequently the last portion of the liquid to solidify is enriched in the constituent metal of lower melting point. A skeleton, or tree-like structure of the first crystals results, within the interstices, or arms, of which have solidified the last portions of the alloy which are much leaner in the metal of high melting point.

The mechanical mixture of two crystals forming simultaneously at the junction of two liquidus curves is called a *eutectic*, and the constant temperature at which it completely crystallizes is called the *eutectic temperature.*

Other relations than those discussed above are possible. *Allotropic transformations* at invariant temperatures may occur in either, or both, of the constituent metals, resulting in corresponding crystal and phase changes in the alloys. The temperature, or temperature range, at or within which phase changes resulting from allotropic transformations in metals or compounds, or intermediate crystal phases, occur on heating or cooling an alloy, is called *critical temperature* or *critical range*. These phase changes are *reversible* if they occur at the same temperature on heating as on cooling; a *temperature lag*, which in some cases may be as great as several hundred degrees Centigrade, occurs in the case of *irreversible* transformations. *Intermetallic compounds* may be found between metals and behave, and are represented in the equilibrium diagram, like metals.

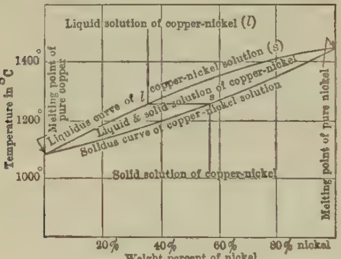

Fig. 1

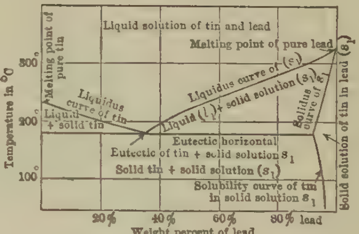

Fig. 2

**GRAIN SIZE AND GROWTH.**—Cast metals, as a rule, are coarser grained than metals that have been worked and annealed. The grain size of wrought metals is set by conditions of working and annealing. Cold-worked metal generates new grains when heated, beginning at a characteristic *recrystallization temperature*, above which the grain size developed increases with rise in annealing temperature. Heavy cold-working tends to lower the recrystallization temperature and to produce finer and more uniform recrystallized grains. A considerable increase in duration of annealing increases grain size less than a moderate rise in annealing temperature. Grain growth is not a reversible process; coarser grains can be developed readily by re-annealing at a higher temperature, but finer grains cannot be produced by re-annealing at lower temperatures, except through intervening cold work. An exception is metals having allotropic transformations, as steel.

Grain size in ingots and castings is determined by conditions during solidification, slow freezing promoting coarseness of grain. Thus, high casting temperature, heavy sections and sand molds tend to produce large grains, while chilling, thin sections and agitation promote fine grains. Centrifugally-cast shapes are fine grained. Heavy ingots are apt to have a coarse structure, which is broken up by working only with difficulty. Coarse, column-shaped grains result from large temperature gradients during freezing.

Some properties of metals are closely related to grain size. Those which are considerably affected are tensile and compressive strengths, ductility, impact toughness, hardness, magnetic hysteresis and permeability; on the other hand, elastic modulus, density, electrical properties, magnetic saturation, induction, thermal conductivity and expansivity are very slightly affected, if at all. Fineness and uniformity of grain are sought when high and uniform strength, toughness and ductility, or good deep drawing properties are desired; coarse grain is a desirable condition in some high-permeability magnetic alloys, and in metals from which strength at the highest temperatures is required.

Grain size customarily is expressed as *average grain diameter*, *i.e.*, $\sqrt{A/N}$, where $A$ = area of some plane section, and $N$ = the total number of grains visible upon it; it alternatively may be expressed as the number of grains per square inch, and usually is measured with a microscope. Grain sizes in metal range from 0.000 002 in. in hardened steel to 1 in. or more in some transformer steels or in castings; annealed non-ferrous metals with good mechanical properties usually range between 0.0005 in. and 0.005 in. average diameter. Sheets or strip for deep drawing often develop a pebbled or *orange peel surface* after drawing, if the grain size is over about 0.004 in. diameter. Standard grain size classifications for steel (A.S.T.M. Standard E 19) are:

| No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| Grains per sq. in. at 100 × | Up to 1.5 | 1.5–3 | 3–6 | 6–12 | 12–24 | 24–48 | 48–96 | Over 96 |

**PHYSICAL PROPERTIES AND STRUCTURE, COMPOSITION AND TREATMENT.**—The mechanical and physical properties of metals and alloys primarily are determined by the chemical composition, the structure and the thermal and mechanical treatment to which they have been subjected, and may be controlled with some precision or varied through wide or narrow ranges by suitable variations in these factors.

The mechanical properties; *i.e.*, hardness, ductility, tensile and compressive properties of alloys depend quite markedly on their chemical composition, although no precise laws are known by which the relation of these qualities to the composition may be described. In general, the addition of one or more metals, in relatively small quantities, to another, causes an increase in hardness and tensile strength and a decrease in ductility. If the added metals go into solid solution in the original metal, the ductility may increase as well (the brasses and cupro-nickel). Ductility seems uniformly to be impaired by the addition of elements which result in the formation of two or more constituents in the alloy. For this reason, commercially useful alloys generally are composed predominantely of one metal, as the alloys in which both or all metals are present in large proportions are apt to be too brittle to be serviceable. The *hardening power* of different metals in conjunction with any other is, roughly, inversely proportional to the percentage of it which is necessary to exceed its solubility in the solid solution formed, or to form the first eutectic. The mechanical properties also are markedly affected by thermal and mechanical treatment.

**HARDENING OF METALS.**—A characteristic of metals is their hardening capacity, or ability to be changed in hardness by treatment to suit requirements. The term *hardness* describes a property defined as *resistance to permanent deformation*, and usually is measured with indentation instruments. It often is used in special senses, connoting difficult degrees of machinability, resistance to abrasion, or simply strength. Metals, including alloys, owe their hardness ultimately to forces between atoms composing the space lattice, and have exceedingly great potential strengths which are not obtained in practice due to the lack of complete continuity in structure and tendencies to rupture by degrees. Thus, wires and filaments generally have greater tensile strengths than metals in bulk. A result of the crystalline structure of metals is low shearing strength along lattice planes, and metals begin to distort by slippage without fracture within grains. Anything which obstructs slippage increases hardness and strength, and the various hardening treatments resolve into this ultimate principle. Cold-working causes slip and fragmentation which by mutual interference hinder further slip. Small hard particles, *e.g.*, of another crystal or foreign substance, scattered within the grain, act mechanically as keys to hinder slip. This action improves as the particles become finer; the most effective size or *critical dispersion* is about one ten-millionth inch. The *dispersion theory* of hardening considers the quench-hardening of steel, the spontaneous aging of duralumin and the temper hardening of certain non-ferrous alloys as examples of this, and heat treatment response as the dissolving or throwing out of solution of these keying particles by changes in temperature.

**EFFECT OF TEMPERATURE.**—Two effects of temperature on metals are distinguished: temporary, in which properties characteristic of the temperature are acquired; and permanent, in which changes occur which persist upon return of the metal to normal temperatures. The art of heat treatment is built upon the latter. At low, sub-atmospheric temperatures many steels and iron alloys become quite brittle, while certain nickel and copper alloys remain tough. At high temperatures, not only are metals subject to atmospheric attack, or *scaling*, but take on characteristics which are not readily disclosed in brief tests. *Flow* under sustained loading, which is absent in elastic materials at ordinary temperatures, becomes the dominant characteristic at temperatures beginning considerably below the softening range, and fracture may finally occur with stresses as little as $^1/_{10}$ the breaking stress found in ordinarily rapid tension tests. Within or below the softening range, flow diminishes with time, due to accumulation of work-hardening effects, and *creep stresses* are designated (European terminology differs) with reference to temperature and rates of stretching, *e.g.*, 1% elongation in 100,000 hours. For data on high and low temperature properties consult A.S.T.M. Symposium on Effects of Temperature on Metals, 1931.

**TREATMENT OF METALS.**—When a metal or alloy is deformed in the cold, as by rolling, hammering, swaging or drawing, its hardness and tensile strength increase and its ductility decreases. These changes are roughly proportional to the amount of deformation, usually expressed as percentage reduction in cross-sectional area, or reduction in thickness in the case of rolling. The hardness and tensile strength of ductile alloys thus may be increased two, and even three, times with, however, relatively much greater loss in ductility (to $^1/_{30}$ or $^1/_{50}$). Very heavy working develops fiber both as to included particles and grain structure, which results in difference in properties in different directions, *e.g.*, longitudinal *vs.* transverse. Once developed, this fiber structure may persist after annealing. Overworking results in edge cracking, internal bursts and other permanent impairments of quality. *Hot-working*, as forging, rolling, or piercing does not alter the mechanical properties of the material to the extent that *cold-working* does, as it is accompanied by simultaneous annealing and restoration of original properties. In general, a metal is cold-worked below the range of annealing temperatures, and hot-worked within and above it; for example, zinc may be hot-worked at temperatures which would result in cold-working of steel. *Extrusion* is a form of hot working in which the metal is heated to a plastic condition and squirted through a die under heavy pressure.

Upon *annealing*, cold-worked metals recrystallize and grain growth occurs, accompanied by restoration of the original mechanical properties. It is found that there exists for each metal or alloy, temperature ranges including usually about 200° C. (392° F.), above which annealing is completed rapidly, *i.e.*, the ductility is restored and the hardness is reduced to normal value. Within this range the annealing is completed slowly, and below it the metal cannot be altered so markedly by heating. This is called the *softening range* for the alloy in question; softening ranges for common materials are given in Table 1.

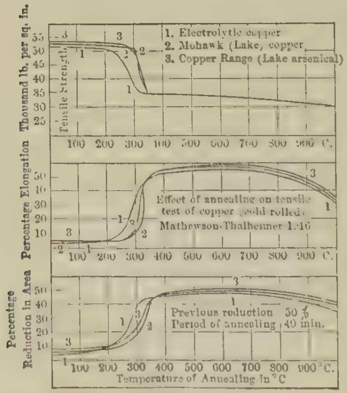

FIG. 3

Fig. 3 illustrates typically the effect of cold-working and annealing on the mechanical properties of metals; in this case, copper. In simple alloys, maximum softness results from annealing at the highest temperatures, while good combinations of mechanical properties, *i.e.*, fine grain, strength and ductility, result from annealing close to the softening range. The terms annealing and normalizing, while used more or less interchangeably to designate softening heat treatments for non-ferrous metals, have taken on specialized meanings in the case of steel. Annealing of steel consists in heating to a temperature above the critical range and cooling slowly, as in a large furnace; normalizing consists in cooling from that temperature freely in the open air. *Stress relief* annealing of cold-worked metal consists in heating at temperatures far below the softening range, *e.g.*, 200° C. (392° F.) for brass, 300–400° C. (572–752° F.) for Monel metal and steel; the metal remains hard but is improved in elastic properties and toughness, and internal stresses are dissipated.

Certain non-ferrous alloys with constituents of variable solid solubility are thermally hardenable. Such dispersion-hardening alloys are placed in a soft and workable condi-

Table 1.—Softening Ranges of Temperature of Various Metals and Alloys

| | Deg. C. | Deg. F. | | Deg. C. | Deg. F. |
|---|---|---|---|---|---|
| Zinc | 60–150 | 140–302 | Nickel-silver | 350–650 | 662–1202 |
| Aluminum | 200–400 | 392–752 | Iron and low carbon steel | 600–800 | 1112–1472 |
| Copper | 200–400 | 392–752 | Nickel | 600–800 | 1112–1472 |
| High Brass | 300–600 | 572–1112 | Monel metal | 600–800 | 1112–1472 |

tion by quenching from an appropriate temperature, and they then may be hardened by reheating for a considerable time, usually a number of hours, at some lower temperature. Table 2 gives heat treatment ranges for a few more important alloys.

**SURFACE TREATMENT** of metals to produce a hard or chemically resistant skin is accomplished by prolonged heating in contact with the impregnating substance or an unstable compound, and the surface alloy is formed by *diffusion* in the solid state; such are carburizing treatments of steel with carbonaceous solids or gases; nitriding with ammonia; zinc (sherardizing), aluminum (calorizing), chromium, as well as others.

**PHYSICAL PROPERTIES.**—The effect of composition and structure of alloys upon their other physical properties may be briefly considered. The *specific volume*, or reciprocal of the *density*, of alloys is approximately a linear function of the percentage composition, regardless of the type of alloy. This is true also of the *specific heat*. The *electrical conductivity* of binary alloys generally is linearly dependent on the volume percentage composition of the constituents when they are insoluble in the solid state, but is markedly less than this theoretical amount in the case of solid solutions. For this reason alloys for *electrical resistance purposes*, *i.e.*, copper-nickel, nickel-chromium, and iron-nickel, are chosen from among the latter type. The *temperature-coefficient* of electrical resistivity generally also is small in the case of alloys of the solid solution type having high values of the restivity. In general the *thermal conductivity* follows the electrical conductivity, being low in the case of solid-solution alloys.

There are given below some approximate relations between the various physical properties of metallic materials which may be useful for a rough orientation regarding the value of any particular one from a knowledge of others.

*Grüneisens relation:*

$$\frac{\alpha}{c_p} = n \quad \left\{ \begin{array}{l} \alpha = \text{coefficient of thermal expansion;} \\ c_p = \text{specific heat at constant pressure;} \\ n = \text{constant for each material independent of temperature.} \end{array} \right.$$

*Wiedemann-Franz law:*

$$\frac{k}{x} = nT \quad \left\{ \begin{array}{l} \left. \begin{array}{l} k = \text{thermal conductivity} \\ x = \text{electrical conductivity} \end{array} \right\} \text{in C.G.S. units;} \\ n = \text{constant varying from 5000 in the case of pure metals and} \\ \qquad \text{alloys without solid solutions to 7000 in the case of solid} \\ \qquad \text{solutions;} \\ T = \text{absolute temperature.} \end{array} \right.$$

*Richards relation:*

$$L = 2.1 \times m.p. \quad \left\{ \begin{array}{l} L = \text{latent heat per gram atom or molecule;} \\ m.p. = \text{melting point, absolute.} \end{array} \right.$$

The physical properties, as a rule, are not affected markedly by thermal and mechanical treatment. Electrical resistivity is increased and the temperature coefficient of resistance is diminished by cold-working; density, specific heat, elasticity, and thermal conductivity are only slightly altered by the most severe mechanical and thermal treatments. An exception is magnetic properties, which in ferro-magnetic materials often are highly sensitive to strain and heat treatment, and the development of exceedingly high magnetic quality requires painstaking attention to these details.

**TESTING OF METALS AND ALLOYS.**—For definitions of the physical and mechanical properties of metallic materials which usually are determined, see p. 7–03. For descriptions of the methods of test see p. 7–51. See also A.S.T.M. Standards, and A.S.M. Recommended Practice Committee (National Metals Handbook).

The engineer should take great pains to ascertain the exact relation of test results, or quoted information on metallic materials, to the problem of design or construction before him. In the case of rolled, forged, or drawn products not heat-treated, this may consist chiefly only in assuring himself of the accuracy of the data and in the consideration of the necessary factors of safety; the properties of the material going into his design or construc-

**Table 2.—Heat Treatment Ranges of Various Alloys**

| | Softening Quench | | Hardening Draw | |
|---|---|---|---|---|
| | Deg. C. | Deg. F. | Deg. C. | Deg. F. |
| Duralumin | 500–515 | 932–959 | Room temp. | Room temp. |
| "25 S" | 510–530 | 950–986 | 120–160 | 248–302 |
| Nickel bronze | 700–900 | 1292–1652 | 300–500 | 572–932 |
| Copper-beryllium | 750–850 | 1382–1562 | 250–350 | 482–662 |
| Copper-nickel-silicon | 750–900 | 1382–1652 | 350–550 | 662–1022 |

tion should be identical with those obtained in the ordinary tests. This is not true in the case of castings.

The mechanical properties of castings depend to a great extent on their mass and cross-section. It is the usual practice in obtaining tests on castings to pour either separately or in the same flask with the castings, from the same heat or melt, a test coupon of standard shape and section. The shape and section of this coupon varies with different materials and often unfortunately in different foundries for the same material. The properties determined from this coupon will be characteristic of the melt of metal from which it was poured, but characteristic only of the section and mass of the coupon, *not of the casting itself.* Generally speaking, hardness and tensile strength and elongation in the tensile test increase as mass and section of the casting decrease. For this variation allowance must be made, particularly in the design of castings of large mass and section.

**Non-destructive Inspections** often can detect internal flaws, as seams, cracks, holes or porous sections; for example, by disturbance in potential drop or of flux distribution when a standardized current or magnetic flux is passed through the object. x-ray radiographs reveal defects involving changes in density, as blowholes or cracks in welds, etc. Steel sections up to 4 in. thick are accessible to this method. A similar method utilizes the *gamma radiation* from radium or radium emanation.

Great caution should be used in the interpretation of corrosion tests, as slight differences in arrangement often develop great differences in result; tests carefully simulating all details of service frequently are necessary.

**FAILURES AND DEFECTS IN METALS AND ALLOYS.**—Commercial shapes, as rods, sheet, tubes, plate, etc., often are subjected to inspection for surface defects such as laps, seams, splits, cracks, blisters, or scale, which are due to a variety of causes extending throughout the manufacturing process and which vary with different materials. A commercially good shape is determined by practice and usage in each case, and general specifications for the presence, or absence, of such surface defects cannot be laid down.

**Pipe** in the ingot, or casting, often is the cause of defects in manufactured shapes. This is a shrinkage cavity formed by insufficient feeding of metal while pouring the casting, and usually is found in the upper central part of the casting. It often is accompanied by *segregation*, that is, the non-uniform distribution of impurities or alloying elements throughout the casting or shape made from it. In such cases there generally is a greater concentration of such impurities in the center of the ingot, casting, or resulting shape, than in the outer portions. It is due to selective freezing, whereas *liquation* is an actual separation taking place in the liquid or semi-liquid state and not necessarily directly related to the freezing. Lead may liquate in copper, carbon segregates in steel.

**Blow-holes or Gas-holes** in castings are formed by evolution of gas in the metal just prior to solidification in the mold.

**Contraction Cracks or Mold Pulls** in casting are the result of interference by parts of the mold, or core, with the natural contraction of the solid alloy, particularly through its temperature zone of weakness. Misplaced or shifted cores, washing of sand, and dirt in castings, blows, caused by wet sand or gas in metal, porosity, draws caused by shrinkage at changes in section, mis-runs due to lack of temperature, poor match of cope and drag, cold shuts also should be looked for in commercial castings.

**The Failure of Metals and Alloys** generally connotes mechanical failure by fracture, yielding, or cracking. *Yielding* of a metal or alloy indicates only that the applied stress was greater than its yield point or elastic limit. Examination usually is made of a fractured or cracked surface to determine the cause of failure. A detail fracture can be seen to originate at some point in the fractured section, generally a material defect of some sort. Mechanical failure from shear or torsional stress usually is observed readily from the character of the fractured surface. Fracture from direct static tension stress (that is, not changing or only slowly changing) should not occur in sound and ductile metals or alloys without appreciable elongation on both sides of the fracture. When a brittle or short fracture is observed in such materials, without appreciable elongation, the possibility should be considered of the failure having occurred by reason of the application of alternating or repeated stress, or as the result of a sharp blow or impact, and particularly if sharp offsets, shoulders, or grooves in the failed article tend to localize the stress and produce a notch effect. All metals if subjected to vibratory, repeated or reversed stresses ultimately will fail (after perhaps millions of such cycles) at intensities of stress much below that required to break in tension under steady loading. Such *fatigue failures* sometimes occur in shafts and other machine parts subject to rapidly changing loads. Under cyclic stressing, many metals show true *endurance limits*, *i.e.*, limiting stresses below which they will escape breakage indefinitely. The endurance limits of common metals and alloys range from 35 to 55% of their tensile strength. Some forms of corrosion accelerate fatigue failure. The oft-quoted explanation of mysterious failures of metallic materials—

that of their "crystallization" is utterly fallacious. The usual materials, iron, steel, brass, bronze, copper, etc., always are crystalline. When fractured in impact or under the action of repeated stress they often display a crystalline fractured surface, solely by reason of the fact that the path of break in such cases relates itself more definitely to the crystalline structure of the material.

**Season or Corrosion-cracking**, of certain metals and alloys often occurs without external applied stress, but as the result of internal stress in the material, combined with surface corrosion. Stress-relief annealing is the preventive measure. Factors controlling the corrosion of metals are too complicated for discussion here. Consult U. R. Evans, The Corrosion of Metals, Edward Arnold & Co.; see also pp. 3–03 to 3–19.

**ENGINEERING ALLOYS.**—The alloys of engineering importance are few compared with the almost limitless number of possible mixtures, and the tendency is toward greater standardization. A number of the engineering metals now are defined by specifications. Besides these, there are upwards of 1000 proprietary alloys, generally bearing names which do not suggest their composition, but similar to standard commerical alloys in properties. In addition, there are many obsolete alloys which have been described and have established a foothold in handbooks, but which no longer have distinctive qualities. The number is too great to list here. *Metal Industry*, London, publishes a handbook of Metals and Alloys which lists many of these by name. Prof. W. Campbell has published, through A.S.T.M. (1930) A List of Alloys. A list of corrosion-resisting alloys is given in *Chem. and Met. Engg.*, Sept., 1932 (p. 497). See also pp. 4–19 to 4–34.

**AMERICAN NON-FERROUS METAL SPECIFICATIONS.**—The principal standardizing agencies in the U. S. for metals are the American Society for Testing Materials, Philadelphia; the Society of Automotive Engineers, New York; the Federal Specifications Board, Bureau of Standards, Washington. D. C. The American Engineering Standards Committee publishes standard specifications for non-ferrous metals which are identical to those published by A.S.T.M. In addition, many manufacturing plants and railroads issue their own purchase specifications, copies of which often may be obtained on request. Many other standard specifications, dealing with metal manufactures and finished products, are indexed in the *National Directory of Commodity Specifications* (Miscellaneous Publication No. 130), obtainable from the Superintendent of Documents, Washington, D. C.

# COPPER

By W. H. Bassett

**PHYSICAL PROPERTIES OF COPPER** are as follows:

### General Properties

| | | | |
|---|---|---|---|
| Density | 0.322 lb. per cu. in. | Specific heat at 77° F. | 0.0917 |
| Melting point | 1981 deg. F. | Electrical resistivity at 20° C. | 1.27 microhm-cm. |
| Coefficient of expansion per deg. F. | 77–100° F. 0.000 009 332<br>77–572° F. 0.000 009 888 | Temperature coefficient of resistivity at 20° C. | 0.0039 per deg. C. |
| Pattern shrinkage | 1/4 in. per ft. | Magnetic properties | Diamagnetic |
| Thermal conductivity at 77° F., B.t.u. per hr. per sq. ft. per inch thickness per deg. F. | .2679 | Optical properties | Selectively reflecting |
| | | Young's modulus | 17,300,000 |

### Tensile Properties

| | Annealed | Cold Rolled or Drawn | Cast |
|---|---|---|---|
| Tensile strength, lb. per sq. in. | 30,000–40,000 | 50,000–70,000 | 20,000–30,000 |
| Elongation in 2 in. | 40–60% | 2–4% | 25–45% |
| Reduction of area | 40–60% | 2–4% | ............ |

### Hardness

| | Annealed | Cold Rolled or Drawn |
|---|---|---|
| Brinell (500 Kg.) | 30–40 | 60–100 |
| Scleroscope (Universal Hammer) | 6–8 | 20– 25 |

**COMPOSITION OF COMMERCIAL COPPER.**—Specifications for copper, generally accepted by industry, are the A.S.T.M. standard specifications. These cover lake copper, electrolytic copper, and fire-refined copper. Lake copper, which must originate on the upper peninsula of Michigan, may be low-resistance or high-resistance lake copper.

**Low-resistance Lake Copper** is used for electrical purposes, and may be electrolytically or fire refined. It is required to have a content of (copper + silver) of not less than 99.900%. Maximum permissible resistivities in international ohms (meter, gram) are as follows: Copper wire bars, 0.15436; ingots and ingot bars, 0.15694.

**High-resistance Lake Copper** is lake copper of resistivity exceeding 0.15694 international ohm (meter, gram) at 20° C. Its (copper + silver + arsenic) content is not less than 99.900%.

**Electrolytic Copper** specifications for purity and resistivity are the same as those of low-resistance lake copper.

**Fire-refined Copper,** other than lake, is used for rolling into sheets and shapes. It is not acceptable for electrical uses or for wrought alloys. A.S.T.M. Specifications call for the following analysis: Cu + Ag, 99.700%; As, max., 0.1000%; Sb, max., 0.0120%; Bi, max., 0.0020%; Fe, max., 0.0100%; Pb, max., 0.0100%; Ni, max., 0.1000%; O, max., 0.0750%; Se, max., 0.0400%; Te, max., 0.0140%; Sn, max., 0.0500%.

**HARDENED COPPER.**—There are only two methods for hardening copper; namely by cold working it or by alloying with it small amounts of other elements such as silicon, tin, zinc, aluminum, etc. Both of these methods are in commercial use. When copper is hardened with tin, silicon, or aluminum it generally is called bronze; when hardened with zinc, brass.

**CORROSION.**—Copper is resistant to the action of sea-water, and to atmospheric corrosion. It is not resistant to the common acids, and is unsatisfactory in service with ammonia and with most compounds of sulphur. Manufacturers should be consulted in regard to its use under corrosive conditions.

**OXYGEN-FREE HIGH-CONDUCTIVITY COPPER** is a highly ductile material, made under conditions that prevent the entrance of oxygen and formation of copper oxide. It is utilized in deep drawing, spinning, and edge bending operations, and in welding, brazing, and other hot-working operations where embrittlement must be avoided. It has the same conductivity and tensile properties as tough pitch electrolytic copper.

**FABRICATION.**—Copper may be hot or cold forged, hot or cold rolled, hot extruded, hot pierced, and drawn, stamped, or spun cold. It can be silver-soldered, brazed, and welded. For welding by the oxy-actylene torch or electric arc, the oxidized copper will give more satisfactory welds than electrolytic or lake copper. Copper cannot be cut by the oxygen jet because of its high heat conductivity. Copper is annealed from 480 to 1400° F., depending on the properties desired, and in a slightly oxidizing atmosphere. Ordinary commercial annealing is done in the neighborhood of 1100° F. A reducing atmosphere will make the copper brittle and unfit for use. Copper may be electro-deposited from the alkaline cyanide solution, or from the acid sulphate solution.

**COMMERCIAL FORMS OF COPPER.**—Copper is produced in practically all commercial forms, *viz.*, wire, sheets, strips, bars, plates, seamless tubes and castings, and in most fabricated forms as screen, wire cloth, bolts, nuts, etc. For dimensions, see pp. 6–08 to 6–12 and 6–19 to 6–22. The following is a list of the specifications of copper for various purposes given in the 1936 A.S.T.M. standards:

Designation B4–27, Lake Copper Wire Bars, Cakes, Slabs, Billets, Ingots, and Ingot Bars; Designation B5–27, Electrolytic Copper Wire Bars, Cakes, Slabs, Billets, Ingots, and Ingot Bars; Designation B72–33, Fire-refined Copper, other than Lake; Designation B1–27, Hard-drawn Copper Wire; Designation B2–27, Medium Hard-drawn Copper Wire; Designation B3–27, Soft or Annealed Copper Wire; Designation B33–21, Tinned, Soft or Annealed Copper Wire for Rubber Insulation; Designation B8–36, Bare Stranded Copper Cable, Hard, Medium-hard, or Soft; Designation B11–33, Copper Plates for Locomotive Fireboxes; Designation B12–33, Copper Bars for Locomotive Staybolts; Designation B13–33, Seamless Copper Boiler Tubes; Designation B42–33, Copper Pipe, Standard Sizes; Designation B48–27, Soft Rectangular Copper Wire; Designation B49–26, Hot-rolled Copper Rods for Wire Drawing; Designation B68–33, Seamless Copper Tubing, Bright Annealed; Designation B75-30, Seamless Copper Tubes; Designation B88–33, Copper Water Tube.

# ZINC AND ITS ALLOYS

By W. M. Peirce

**References.**—U. S. Bureau of Standards Circular No. 395, Zinc and Its Alloys (1931), which includes a complete bibliography. Valuable bulletins on various subjects relating to zinc and its uses may be obtained from the principal producers of zinc and from the American Zinc Institute.

## 1. PROPERTIES OF ZINC

**CHEMICAL CHARACTERISTICS.**—Zinc is a chemically active metal. Its chemical properties are the reason for its use in primary cells, for cyaniding and in the production of nascent hydrogen. It forms both basic and acid radicals, and is soluble both in acids and strong alkalies.

**Atmospheric Corrosion.**—In pure distilled water zinc forms a loosely adherent film of hydroxide which affords little protection. This film, dried in the presence of oxygen and the traces of carbon dioxide always present in the atmosphere, forms a protective layer or film believed to be a zinc oxide-hydroxide-carbonate mixture. This is quite insoluble in rainwater, affords effective protection against further attack, and makes zinc a satisfactory roofing and flashing material. The presence of sulphur gases in industrial atmospheres increases the solubility of this film, accounting for the higher rate of weathering in industrial as compared to rural atmospheres.

**Effect of Organic Acids.**—Certain organic acids dissolve the atmospheric corrosion film on zinc. Acids formed by decomposition of red cedar and white cedar at ordinary temperatures, and of other woods at higher temperatures (as in steam) have this effect. Hence zinc cannot be safely used in contact with some moist woods or drainage from them.

**Resistance to Organic Substances.**—Zinc is inert to such common organic substances as gasoline, kerosene, oil, and grease free from water. Soap and soap paste do not attack it, though in dilute soap solutions there is a tendency to form zinc soaps. Zinc is entirely inert toward anhydrous alcohol, but is oxidized by mixtures of alcohol and water. Sugar solutions made up with hard water do not attack zinc when totally immersed at room temperature, but distilled-water solutions at any temperature or hard-water solutions at elevated temperatures attack it.

**Effect of Carbon Dioxide and Steam.**—Very high carbon dioxide concentrations appear to aid the solution of the corrosion film. Increased temperature sometimes may have the same effect. Live steam destroys the protective film and also attacks the metallic zinc at an increased rate. Hence unalloyed zinc is unsuitable for use when exposed to live steam.

**Effect of Oxygen Concentration.**—Severe corrosion of zinc occasionally occurs in the presence of a water film without free access of oxygen to all portions of it. An example is a zinc roof, laid over an absorbent sheathing and building paper, on a building having high humidity inside. If the roof is unprotected from condensation on the under side, rapid corrosion of the zinc may occur. The preventive is building paper, saturated and coated on both sides with asphalt, under the metal roofing.

**Toxicity.**—Excepting the salts of strong acids, which are corrosive, or zinc compounds containing other poisonous elements or radicals, the compounds of zinc are not strongly toxic. Zinc is a normal constituent in many foods and considerable excesses of zinc taken into the body are readily eliminated without ill effects. Excess zinc content in foods or beverages may cause unpleasant but not serious symptoms. Drinking water should not contain more than 40 mg. per liter.

**PHYSICAL PROPERTIES.**—The more important physical constants of zinc are given in Table 1.

**METALLOGRAPHY.**—The common impurities in zinc are lead, cadmium, and iron. Their importance varies individually with the use to which the metal is put. These metals, as well as copper, magnesium, and aluminum, which are the more common elements used in zinc alloys, are discussed below.

**Lead,** which does not enter into solid solution, has no great effect in unalloyed zinc. Its effect on the properties of certain zinc alloys, however, is enormous.

**Cadmium,** in the amounts present in commercial zinc, is completely in solid solution, and exerts an important hardening effect. It also raises the recrystallization temperature,

**Table 1.—Physical Properties of Zinc**

(Compiled from various sources)

| | |
|---|---|
| Atomic weight | 65.38 |
| Density (rolled), (see Note 1) | 445.8 lb. per cu. ft. |
| Melting point | 787° F. |
| Boiling point | 1661 ± 4° F. |
| Mean specific heat (68°–212° F.) | 0.0931 |
| Latent heat of fusion | 47.88 B.t.u. per lb. |
| Latent heat of vaporization | 768.2 B.t.u. per lb. |
| Coefficient of linear expansion | |
| Cast | $21.9 \times 10^{-6}$ Per deg. F. |
| Rolled, with grain | $17.8 \times 10^{-6}$ |
| Rolled, across grain | $12.8 \times 10^{-6}$ |
| Thermal conductivity | 784 B.t.u. per sq. ft. per hr. per 1 in. thickness per deg. F. |
| Electrical resistivity (see Note 2) | 2.33 microhm-in. |
| Temperature coefficient of resistance (0°–100° C.) | 0.00233 per deg. F. |
| Resonance | See Note 3 |

NOTES. 1. Solid zinc (rolled) is about 3% heavier than liquid zinc at the melting point.

2. Based on soft rolled zinc strip 99.99% pure. Resistivity of strip zinc may vary 3%, depending on rolling treatment, and in commercial grades about the same amount, depending on purity.

3. Rolled zinc has extremely low resonance, which permits its use in various sound apparatus.

and whereas pure zinc recrystallizes at so little above room temperature that it cannot be appreciably work hardened, zinc containing cadmium can be given a considerable degree of work hardening by cold rolling.

**Iron** has a limited solid solubility of the order of 0.001 to 0.003%. Attention must be given to the iron content of zinc for rolling.

**Copper** is the alloying element most frequently added to zinc. It enters into solid solution in zinc to about the same extent (approximately 1%), and with about the same effect, as cadmium, except that the copper-zinc alloys are more ductile and easier to roll. See pp. 4–18 to 4–33.

**Magnesium** has a solid solubility of 0.01 or 0.02% in zinc, but even these amounts exert a remarkable influence on its properties. When magnesium is added in the presence of copper, its effect is even more striking. Some of the most important zinc alloys contain both copper and magnesium.

**Aluminum** is limited entirely to zinc casting alloys, as usually zinc alloys containing aluminum are unstable when rolled. In the cast form, however, several percent of aluminum can be used to improve greatly the physical properties, particularly the tensile and impact strength of zinc. The alloys, if properly formulated, are stable.

## 2. SLAB ZINC

**COMPOSITION OF SLAB ZINC.**—A.S.T.M. Standard Specifications B6–33 for slab zinc (Spelter) gives the values of Table 2 for the various grades of slab zinc.

### Uses of Slab Zinc

**GALVANIZING** is the process of coating iron or steel with zinc as a protection against corrosion. The product is known as galvanized iron or steel. Zinc coatings are desirable not only because the zinc corrodes more slowly than iron but also because at cut edges or similar discontinuities in the coating, the zinc, being anodic to the iron, affords galvanic protection and the iron is not dissolved until the zinc has been completely removed by corrosion.

The protection afforded depends largely on the thickness of the coating. For protection indoors, a few tenths of an ounce is sufficient. For outdoor exposure a coating of at least $^3/_4$ oz. per sq. ft. of surface, and preferably 1 oz. or more is necessary.

The less pure grades of zinc commonly are used for galvanizing, but the coating so produced will not withstand much deformation without cracking and flaking. High-grade zinc nearly free from cadmium is used for galvanizing bridge wire, telegraph wire and other products that are subject to deformation such as twisting, etc., and for sheets that are to be fabricated after galvanizing.

The life of zinc coatings, regardless of the method of application, has been shown to be directly proportional to the thickness or weight of the coating. The proportion of pure zinc to iron-zinc alloys in the coating does not have any significant effect.

**Specifications and Tests for Galvanized Coatings.**—Galvanized coatings usually are specified as a given weight of zinc per unit area, *i.e.*, ounces of zinc per square foot of surface. For details of methods of determining the weight of the coating see Standard Methods of Determining Weight of Coating on Zinc Coated Articles, A.S.T.M. Specification A–90–33.

Specifications for most commercial products covering weight of coating and bend tests when required are available in A.S.T.M. Standards, 1936, Part I, Metals, and A.S.T.M. Tentative Standards, 1936.

On sheets, coating weights are given as weight per square foot of sheet, *i.e.*, 2 sq. ft. of coated surface. Ordinary coatings on sheets range from 0.75 to 2.75 oz. per sq. ft. of sheet coated on both sides. The weight of zinc per single covered surface would be one-half this amount. Zinc coatings on pipe, structural shapes, hardware, etc., generally

**Table 2.—Standard Specifications of Slab Zinc**

| | Grade | Maximum Allowable Percentages of | | | Lead + Cadmium +Iron not to Exceed |
|---|---|---|---|---|---|
| | | Lead | Cadmium | Iron | |
| 1a | Special high grade *† | 0.010 | 0.005 | 0.005 | 0.010 |
| 1 | High grade* | .07 | .07 | .03 | .10 |
| 2 | Intermediate* | .20 | .50 | .03 | .50 |
| 3 | Brass special* | .60 | .50 | .03 | 1.00 |
| 4 | Selected* | .80 | .75 | .04 | 1.25 |
| 5 | Prime western | 1.60 | ........ | .08 | .......... |

* All grades except prime western must be free from aluminum. †Tentative standard.

range between 1.15 and 3.5 oz. per sq. ft. of covered surface. The weight of coating on wire varies with the size of wire and process of coating, and, except in fine sizes, will range from 0.15 to 1.25 oz. per sq. ft. of covered surface.

**METHODS OF APPLICATION.**—Galvanized coatings are applied: (*a*) by dipping the article in molten zinc, (*b*) by electroplating, (*c*) by heating in contact with zinc dust.

**Hot Dip Galvanizing.**—The iron or steel is cleaned by pickling, usually in hot sulphuric acid. A rinse and a dip in cold hydrochloric acid usually follow, but better practice involves removal of iron salts from the pickled material by vigorous rinsing and scrubbing if necessary, followed by dipping in slightly acidulated 25% zinc chloride. After partial drying, the material enters the bath of molten zinc, sometimes through a flux of zinc ammonium chloride. A solid layer of iron-zinc alloy forms on the iron surface in a few seconds. The longer the immersion, the thicker the alloy layer. The material is withdrawn from the zinc bath through rolls kept clean by zinc ammonium chloride flux in the case of sheet, or through a charcoal sand or asbestos wipe in the case of wire. Speed of withdrawal and character of wipe influence the thickness of the layer of molten zinc adhering to the work. Tin may be added to the zinc bath to produce a sheet with bright spangle. Aluminum may be added in hand dip galvanizing of pipe, shapes, and hardware to produce a smooth coating. The vapor of burning sulphur or ammonium chloride dust sometimes is blown over the surface of sheets to improve brightness.

**Electrogalvanizing.**—Zinc coatings may be applied to wire, conduit, and hardware by electroplating. The only distinguishing feature of this process, as compared to other electroplating processes, lies in the attention given to methods of rapidly depositing heavy coatings (*e.g.*, the patented Tainton Process), since the serviceability of the coatings depends on their thickness and the thickness ordinarily desirable is greater than in purely decorative plating. This method of zinc coating avoids changes in the base steel due to heat and does not produce excessive deposits in recesses, such as the base of threads, which would affect dimensional tolerances.

**SHERARDIZING.**—Zinc and iron alloy readily, and coatings therefore can be produced by heating iron or steel with zinc dust in a closed vessel for several hours. The vapor pressure of the zinc is sufficient at the temperatures used, 600–700° F., to cause a reaction of zinc vapor with the iron. It has been demonstrated that alloying will occur by diffusion at only slightly elevated temperatures. The resulting coating, which is quite uniform, is almost entirely iron-zinc alloy, richer in iron near the base metal, with some mechanically held zinc dust in the outer layers. See also p. 3–14.

Sherardizing may result in some heat treatment of the base metal. Sherardized coatings, like coatings of iron-zinc alloy produced by other methods, afford an excellent base for paints or enamels. The coating itself is a matte gray and can be polished. Like other coatings consisting almost entirely of iron-zinc alloys, sherardized coatings dust, rather than flake, on bending. Also in common with such coatings the protection afforded by a given thickness is, under the average normal service conditions, equal to the protection afforded by an equal thickness of pure zinc. The zinc dust used may be the by-product blue powder from spelter furnaces, or it may be expressly manufactured for the purpose, attention being given to a suitable particle size and metallic zinc content.

**BRASS.**—See p. 4–18. The second largest use of zinc is in the manufacture of brass. Brass alloys are discussed in detail on pp. 4–18 to 4–33. Since lead is objectionable in brass used for drawing, grades of zinc low in lead are largely used in brass manufacture.

**DIE-CASTING.**—Suitable alloys for die-casting require zinc of exceptional purity, and the Special High-grade specification listed above is drawn primarily to define a suitable grade for use in zinc die-casting alloys. Freedom from lead, tin, and cadmium is the primary requirement. By alloying with about 4% aluminum, a few hundredths of 1% magnesium, and in some cases 1 to 3% copper, alloys amenable to die casting, and having in the die-cast condition radically improved properties as compared with unalloyed zinc, are obtained. See also p. 4–60.

**SLUSH CASTING.**—High grade slab zinc is used in slush castings, largely used for lighting fixtures and other ornamental parts. High grade unalloyed zinc is used, and also alloyed zinc containing small percentages of aluminum, and sometimes of copper.

## 3. ROLLED ZINC

**CLASSIFICATION OF ROLLED ZINC.**—Rolled or wrought zinc is classified as follows (see A.S.T.M. Standard Specifications for Rolled Zinc, B 69–29):

Type A. Ribbon zinc and sheets or strips cut from ribbon zinc which has been rolled from a single bar in one continuous direction.

Type B. Sheet zinc or strips cut from sheet zinc that has been rolled by the pack rolling method.

Type C. Boiler and hull plates, which may be rolled either from a single bar or by the pack rolling method.

**Strip Zinc** may be obtained in coils or flat lengths in thicknesses from 0.006 to 0.050 in. and in flat lengths up to 0.10 in. Depending on the gage, maximum width varies from 12 to 20 in., but widths less than 12 in. are obtainable in any gage. Lengths of coils may vary from 100 to 250 ft. in heavy and medium gages, and up to 1600 ft. in the lightest gage. Dimensions of sheets range as follows: Thickness, from 0.006 to 0.10 in.; width, 18 to 60 in.; length, 72 to 144 in.

**Boiler Plates.**—Plates of rolled zinc, usually about 1/2 in. thick, termed boiler plates or hull plates, are attached to the inside of boilers or to steel hulls to give galvanic protection against corrosion. The steel is protected at the expense of the zinc, which dissolves.

**ZINC GAGE.**—Table 3 compares the standard zinc gage and the American or Brown & Sharpe gage and the U. S. Standard gage and gives the tolerances permitted. The use of this special zinc gage is discouraged and, to avoid errors, the actual thickness, expressed in thousands of an inch, is preferable.

**Table 3.—Comparison of Zinc Gage and Other Gages**

| Brown & Sharpe Gage, Approximate Thickness | | U. S. Standard Gage, Approximate Thickness | | Rolled Zinc Gage | | | Permissible Variation in Thickness of Rolled Zinc* | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Type A, in. ± | | | | Type B, Variation, % by Wt. |
| No. | In. | No. | In. | No. | Wt. of 1 sq. ft., lb. | Thickness, in. | 0–6 in. Wide | 6–9 in. Wide | 9–14 in. Wide | 14–20 in. Wide | |
| 34 | 0.0063 | 38 | 0.0062 | 3 | 0.22 | 0.006 | 0.0009 | 0.0010 | 0.0011 | 0.0012 | ±16 |
| .. | ........ | 37 | .0066 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 33 | .0070 | 36 | .0070 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 32 | .0079 | 35 | .0078 | 4 | .30 | .008 | .0010 | .0011 | .0012 | .0013 | ±13 |
| 31 | .0089 | 34 | .0086 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| .. | ........ | 33 | .0093 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 30 | .0100 | 32 | .0101 | 5 | .37 | .010 | .0011 | .0012 | .0013 | .0014 | ±10 |
| 29 | .0112 | 31 | .0109 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 28 | .0126 | 30 | .0125 | 6 | .45 | .012 | .0011 | .0012 | .0013 | .0014 | ± 9 |
| 27 | .0141 | 29 | .0140 | 7 | .52 | .014 | .0013 | .0014 | .0015 | .0016 | ± 8 |
| 26 | .0159 | 28 | .0156 | 8 | .60 | .016 | .0014 | .0015 | .0016 | .0017 | ± 6 |
| 25 | .0179 | 27 | .0171 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 24 | .0201 | 26 | .0187 | 9 | .67 | .018 | .0015 | .0016 | .0017 | .0018 | ± 6 |
| 23 | .0225 | 25 | .0218 | 10 | .75 | .020 | .0016 | .0017 | .0018 | .0019 | ± 6 |
| 22 | .0253 | 24 | .0250 | 11 | .90 | .024 | .0018 | .0019 | .0020 | .0021 | ± 6 |
| 21 | .0284 | 23 | .0281 | 12 | 1.05 | .028 | .0019 | .0020 | .0021 | .0022 | ± 6 |
| 20 | .0319 | 22 | .0312 | 13 | 1.20 | .032 | .0020 | .0021 | .0023 | .0024 | ± 6 |
| 19 | .0353 | 21 | .0343 | 14 | 1.35 | .036 | .0021 | .0023 | .0025 | .0026 | ± 6 |
| 18 | .0403 | 20 | .0375 | 15 | 1.50 | .040 | .0022 | .0024 | .0026 | .0028 | ± 6 |
| 17 | .0452 | 19 | .0437 | 16 | 1.68 | .045 | .0023 | .0025 | .0027 | .0029 | ± 6 |
| 16 | .0508 | 18 | .0500 | 17 | 1.87 | .050 | .0024 | .0026 | .0028 | .0030 | ± 6 |
| 15 | .0570 | 17 | .0562 | 18 | 2.06 | .055 | .0025 | .0027 | .0029 | .0031 | ± 6 |
| 14 | .0640 | 16 | .0625 | 19 | 2.25 | .060 | .0026 | .0028 | .0030 | .0032 | ± 6 |
| 13 | .0719 | 15 | .0703 | 20 | 2.62 | .070 | .0027 | .0029 | .0031 | .0033 | ± 6 |
| 12 | .0808 | 14 | .0781 | 21 | 3.00 | .080 | .0028 | .0030 | .0032 | .0034 | ± 6 |
| 11 | .0907 | 13 | .0937 | 22 | 3.37 | .090 | .0029 | .0031 | .0033 | .0035 | ± 6 |
| 10 | .1018 | 12 | .1093 | 23 | 3.75 | .100 | .0030 | .0032 | .0034 | .0036 | ± 6 |
| 9 | .1144 | 11 | .1250 | 24 | 4.70 | .125 | .0032 | .0033 | .0035 | .0037 | ........ |
| 8 | .1284 | 10 | .1406 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 7 | .1442 | 9 | .1562 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 6 | .1620 | 8 | .1718 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 5 | .1819 | 7 | .1875 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 4 | .2043 | 6 | .2031 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 3 | .2294 | 5 | .2187 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| .. | ........ | 4 | .2343 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 2 | .2576 | 3 | .2500 | 25 | 9.40 | .250 | ........ | ........ | ........ | ........ | ........ |
| .. | ........ | 2 | .2656 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 1 | .2893 | 1 | .2812 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 0 | .3249 | 0 | .3125 | .. | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 3/0 | .4096 | 3/0 | .3750 | 26 | 14.00 | .375 | ........ | ........ | ........ | ........ | ........ |
| .. | ........ | 7/0 | .5000 | 27 | 18.75 | .500 | ........ | ........ | ........ | ........ | ........ |
| .. | ........ | .. | ........ | 28 | 37.50 | 1.000 | ........ | ........ | ........ | ........ | ........ |

* Permissible variations in thickness of Type C rolled zinc are the same as those for either Type A or Type B, depending on whether the material is cut from a ribbon or a sheet.

**TESTING METHODS.**—The properties of zinc depend largely on the rate of loading. It is important that tests be made at rates of deformation that approach those which occur in service. Thus a cupping test, to indicate the suitability of strip for drawing, must be made at the speed of a draw press operation. At the other extreme, corrugated roofing sheets, which must continuously support static loads, must be tested under static load for long periods.

A.S.T.M. Specification B 69–29 calls for temper and dynamic ductility (cupping) tests. These are considered the only ones sufficiently reproducible and valuable for test purposes.

**Temper** is measured by a testing machine which wraps a specimen tightly around a mandrel to which one end of the specimen is clamped. The degree to which the free end springs away from the mandrel is measured on an arbitrary scale, ranging from 0, representing no spring whatever, to 100, representing material that returns to its original straight condition.

The three types of rolled zinc are classified according to temper as follows: Type A: dead soft, soft, half hard, and hard; Type B: commercial; Type C: no classification. It should be noted that the terms soft, dead soft, etc., serve to designate certain specific ranges of properties regardless of the combination of composition and rolling treatment by which the properties are obtained. Table 4 gives the temper requirements of Type A and Type B rolled zinc.

**Dynamic Ductility** is determined by forming a series of cups by forcing a spherical-ended plunger into a test strip, the plunger being adjustably mounted in a drawing press to permit varying the depth of the cup. The maximum depth of the cup, in inches, which can be formed before visible "necking down" occurs is the dynamic ductility. See Table 4 for dynamic ductility requirements of the various types of rolled zinc.

**Tensile Strength,** as ordinarily measured, is of secondary importance in the application of rolled zinc. Other properties, as ductility and temper, must be controlled, which control fixes the tensile strength. Tensile strengths are not used generally in product control and rarely are specified. Percentage of elongation in the tensile test is of little significance, as the rate of loading, 0.25 in. per min., is far below the rate of deformation in commercial operations and is no index to the ductility in such work.

However, where tensile tests are required the A.S.T.M. standard tensile specimen for

**Table 4.—Properties of Rolled Zinc***

| Am. Zinc. Gage No. | Thickness, in. | Temper Requirements, Type A | | | | | | | | Temper Requirements, Type B | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Dead Soft | | Soft | | Half Hard | | Hard | | | |
| | | Dynamic Ductility, min. | Temper, max. | Dynamic Ductility, min. | Temper, max. | Dynamic Ductility, min. | Temper, max. | Dynamic Ductility, min. | Temper, max. | Dynamic Ductility, min. | Temper, max. |
| 3 | 0.006 | 0.275 | 70 | 0.255 | 76 | 0.215 | 80 | 0.165 | 86 | 0.165 | 83 |
| 4 | .008 | .280 | 65 | .260 | 72 | .220 | 75 | .170 | 83 | .170 | 77.5 |
| 5 | .010 | .285 | 60 | .265 | 66 | .225 | 70 | .175 | 80 | .175 | 73 |
| 6 | .012 | .290 | 55.5 | .270 | 62 | .230 | 65 | .180 | 77 | .180 | 69 |
| 7 | .014 | .295 | 52 | .275 | 58 | .235 | 61 | .185 | 74 | .185 | 65 |
| 8 | .016 | .300 | 48.5 | .280 | 54 | .240 | 58 | .190 | 72 | .190 | 62 |
| 9 | .018 | .305 | 46 | .285 | 51 | .245 | 54 | .195 | 69 | .195 | 58.5 |
| 10 | .020 | .310 | 43.5 | .290 | 48 | .250 | 51 | .200 | 68 | .200 | 57 |
| .. | .022 | .315 | 41.5 | .295 | 45 | .255 | 48 | .205 | 65 | .205 | 55.5 |
| 11 | .024 | .320 | 40 | .300 | 43 | .260 | 45 | .210 | 63 | .210 | 54 |
| .. | .026 | .325 | 39 | .305 | 40 | .265 | 43.5 | .215 | 61 | .215 | 52.5 |
| 12 | .028 | .330 | 37.5 | .310 | 37 | .270 | 42.5 | .220 | 59 | .220 | 51 |
| .. | .030 | .335 | 36.5 | .315 | 37 | .275 | 42 | .225 | 58 | .225 | 49 |
| 13 | .032 | .340 | 36 | .320 | 36.5 | .280 | 41 | .230 | 57 | .230 | 48 |
| .. | .034 | .345 | ....... | .324 | 36 | .285 | 40.5 | .235 | 56 | .235 | 47 |
| 14 | .036 | .347 | ....... | .326 | ....... | .290 | ....... | .240 | ....... | .240 | ....... |
| .. | .038 | .349 | ....... | .328 | ....... | .295 | ....... | .245 | ....... | .245 | ....... |
| 15 | .040 | .350 | ....... | .329 | ....... | .300 | ....... | .250 | ....... | .250 | ....... |
| .. | .042 | .352 | ....... | .330 | ....... | .300 | ....... | .250 | ....... | .250 | ....... |
| .. | .044 | .353 | ....... | .331 | ....... | .305 | ....... | .255 | ....... | .255 | ....... |
| .. | .046 | .354 | ....... | .332 | ....... | .305 | ....... | .255 | ....... | .255 | ....... |
| .. | .048 | .355 | ....... | .333 | ....... | .310 | ....... | .260 | ....... | .260 | ....... |
| 17 | .050 | .356 | ....... | .334 | ....... | .310 | ....... | .260 | ....... | .260 | ....... |

* These properties are the minimum requirements for acceptable shipment when no special requirements have been specified.

thin sheet metal is entirely suitable for zinc. A pulling speed as near to 0.25 in. per min. as can be obtained on the testing machine should be used. Under these conditions, rolled zinc will have a "with-grain"* tensile strength varying from 18,000 lb. per sq. in. for soft temper high-grade zinc to 35,000 lb. per sq. in. for hard-temper prime western zinc. The elongation will vary from 5% for the hardest to 65% for the softest metal. The "across-grain" strength in every case will be a few thousand pounds higher and the elongation somewhat lower.

**Static Tensile Tests.**—Of much greater importance than the ordinary tensile test, from a practical standpoint, is the static tensile or creep test which determines the resistance of various grades of zinc to elongation or flow at different temperatures under static tensile load. This information has been correlated with actual loading tests on corrugated zinc roofing sheets. See Tables 5 to 7.

**Modulus of Elasticity** of zinc lies between 10,000,000 and 20,000,000, depending on the rolling treatment. Approximately 12,000,000 is a representative value for soft-rolled high-grade zinc.

**Shearing Strength** of high-grade zinc plates ranges from 17,000 to 19,000 lb. per sq. in.

**FABRICATION.**—The ductility of zinc falls off rapidly at temperatures below 70° F. All drawing and bending operations, therefore, should be performed at temperatures above 70°. Frequently an operation which is just too severe to be successful at temperatures near the lower limit may be successfully accomplished by warming the zinc and tools, for example to 100° F. In any type of extrusion, it is necessary to use an elevated temperature in order to operate with reasonable pressures. For certain ornamental sheet metal work heavy gage zinc is hand formed at temperatures as high as 480° F.

**Lubrication.**—In all operations where a lubricant is required, excepting those operations involving temperatures well above room temperature, soapy water is the cheapest lubricant, and is quite satisfactory. A neutral soap should be used to avoid staining. If finish is important, it may be necessary to avoid spotting by washing the work and then drying it in sawdust.

**Table 5.—Safe Span for Corrugated Zinc Roofing Sheets at 70 deg. F.**

Corrugations 7/8 in. deep × 2 1/2 in.

| | Center to Center of Purlins | |
|---|---|---|
| | 30 lb. per sq. ft. Load | 40 lb. per sq. ft. Load |
| 13 gage 0.032 in. thick | 39.5 in. | 36 in. |
| 14 gage 0.036 in. thick | 41 | 38 |
| 15 gage 0.040 in. thick | 42 | 39 |
| 16 gage 0.045 in. thick | 43.5 | 40 |

**Table 6.—Safe Span for Zilloy † Roofing Sheets at 70 deg. F.**

Under a Uniform Load of 40 lb. per sq. ft.

| Corrugations 1 in. deep × 2 1/2 in. | Safe Span, Center to Center of Purlins | Corrugations 7/8 in. deep × 2 1/2 in. | Safe Span, Center to Center of Purlins |
|---|---|---|---|
| 12 gage 0.028 in. thick | 50.5 in. | 13 gage 0.032 in. thick | 48 in. |
| 13 gage 0.032 in. thick | 54 | 14 gage 0.036 in. thick | 51 |
| 14 gage 0.036 in. thick | 57 | 15 gage 0.040 in. thick | 54 |
| 15 gage 0.040 in. thick | 60 | 16 gage 0.045 in. thick | 57 |
| 16 gage 0.045 in. thick | 64 | | |

**Table 7.—Safe Loads for Zilloy † Roofing Sheets**

Corrugations 1 in. deep × 2 1/2 in.

| | Span Center to Center of Purlins | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 39 in. | 42 in. | 45 in. | 48 in. | 51 in. | 54 in. | 57 in. | 60 in. | 63 in. | |
| | Safe Load, lb. per sq. ft. | | | | | | | | | |
| 12 gage 0.028 in. | 67 | 58 | 50 | 44 | 39 | Rigidity when | | | | |
| 13 gage 0.032 in. | 76 | 65 | 57 | 50 | 44 | 40 | roof is walked | | | |
| 14 gage 0.036 in. | 86 | 74 | 64 | 56 | 50 | 45 | 40 | on question- | | |
| 15 gage 0.040 in. | 96 | 83 | 72 | 63 | 56 | 50 | 45 | 40 | able beyond | |
| 16 gage 0.045 in. | 108 | 93 | 81 | 71 | 63 | 56 | 50 | 45 | 41 | this line |

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* The grain in rolled zinc is considered as running parallel with the direction of rolling.

† Trade Mark, Reg. U. S. Patent Office. A zinc-copper-magnesium alloy patented and sold by The New Jersey Zinc Co.

**Bending** usually is performed either on an ordinary brake or in forming rolls. When using a brake, the edges of the bending and clamping jaws should be rounded to the extent of at least removing the knife edge. If possible, bends in rolled zinc should be at right angles to the grain or rolling direction. It then is easy to obtain proper bending properties without the sacrifice of hardness and stiffness.

A *with-grain bend* is a bend, the edge of which is parallel to the grain. A with-grain temper test is one in which the grain is parallel to the length of the specimen. The bending then occurs across the short dimension.

Bends frequently must be made with the grain because of width limitations in strip zinc, or to take advantage of automatic machinery using coils of ribbon zinc. Zinc manufacturers should be consulted in regard to the best combination of high cold-bend value and high temper for the particular conditions involved. If no drawing operation is contemplated, the omission of dynamic ductility from the zinc specifications may be of advantage.

Temperature is most important in bending zinc, as in other forming operations. A slight warming of the metal sometimes makes possible an otherwise impossible bending operation.

**Drawing.**—Operations in which cups are formed from flat blanks, without any general reduction in gage, are termed drawing. Shop methods in drawing zinc vary considerably from one plant to another, but in general, brass practice is followed, with departures in detail depending on the particular operation. It is unnecessary to anneal pure zinc between operations, and the work-hardening alloys require fewer annealings than brass. Reductions in diameter from blank to the first cup should not exceed 45%. A somewhat lower reduction usually is preferable. It is possible to reduce the wall thickness 25 to 33% in redrawing operations in which the cup diameter is reduced 20%.

**Extrusion.**—Zinc is successfully extruded into any of the forms commonly produced by this process. Tubing and rod have been produced, but in the U. S. development has been limited principally to the production of cups, buttons, nails, etc. Blanks for the extrusion of small articles usually are punched from rolled metal, although cast billets have been used for rods. Because of the directional properties of rolled zinc, some trouble due to non-uniform flow is encountered in some classes of work. A specially-rolled zinc can be supplied to minimize this difficulty.

**Welding** of zinc requires some care, because of the low melting point and the tenacious character of the oxide. The best welding is accomplished with a gas flame, but zinc also may be spot-welded electrically. Zinc cannot be welded without seriously weakening areas adjacent to the weld by annealing. If strength is important, some cold-working operation to refine the grain should follow welding.

Zinc tubing is made commercially by welding, followed by a drawing or swaging to a smaller diameter. It then has a uniform strength that approximates that of the rolled zinc from which the tubing is made.

**Soldering.**—Zinc is easily soldered. A low soldering temperature is necessary to avoid excessive grain growth in areas adjacent to the joint. This is best obtained by using a large iron, hot enough to accomplish the soldering with a single pass. Half-and-half solder generally is used, although a low-melting-point solder also is of advantage. A solder composed of Pb 55%, Sn 30% and Cd 15% is suitable for zinc. It has a low melting point and high tensile strength. Acidulated zinc chloride or killed hydrochloric acid makes an excellent flux, although much zinc is soldered without a flux.

**Hot Stamping.**—Certain complicated and deep draws, such as occur in ornamental sheet metal work, are made in heavy gage zinc by preheating the metal to about 480° F. before stamping. This work requires some knowledge of the working qualities of zinc at this temperature, and should be done by skilled workmen. For some uses, as in the manufacture of organ pipe, the zinc is given a thorough anneal prior to working. A 30-min. anneal at 480° F., while removing much of the desirable bending and drawing qualities, will reduce the metal to the very soft condition required for this use.

**Machining.**—In machining zinc it is common practice to use tools with a greater rake than normally is used, and to use soapy water as a lubricant.

**Plating.**—Articles produced both from die-cast and wrought zinc may be plated readily with any of the common plated finishes. Certain metals are absorbed by zinc, or present some difficulty in direct deposition. Nickel, therefore, commonly is used as a first coat and chromium or any other desired finish plated over it. A thickness of at least 0.0003 in. of nickel is considered necessary to good durability of plating for outdoor exposures.

**Paint and Baked Enamel Finishes.**—Sand blasting generally is recommended for preparing the surface of cast or rolled zinc for the application of any organic finish. Any finish used should be one properly formulated by the manufacturer to give good adherence on zinc. Baking enamels developed for use on zinc are designed to employ a low baking temperature.

# BRASS AND BRONZE

## 1. SAND-CAST, COPPER-BASE ALLOYS

**By W. M. Corse**

**References.**—*The Foundry*, Cleveland; *The Brass World*, New York; *Metal Industry*, New York; *Metal Industry*, London; *Metals and Alloys*, New York; *Chemical and Metallurgical Engineering*, New York; *Revue de Metallurgie*, Paris; *Zeitschrift fur Metallkunde*, Berlin. Also the following proceedings of technical societies: *Trans.* A.F.A., Chicago; *Journal*, Inst. of Metals, London; *Proc.* A.S.T.M., Philadelphia. Also the following books: Metals and Their Alloys, Chas. Vickers; Practical Alloying, J. F. Buchanan; Metallic Alloys, G. H. Gulliver; Mixed Metals, A. H. Hiorns, Technical Analysis of Brass, Price and Meade; Gun Metal and Brass Founding, H. S. and J. S. Primrose; Alloys, A. H. Sexton; Alloys and Their Industrial Application, E. F. Law; Foundry Practice, R. H. Palmer; Introduction to Physical Metallurgy, Walter Rosenhain; Metallography, C. H. Desch; The Science of Metals, Z. Jeffries and R. S. Archer; The Metallurgy of Bronzes; H. C. Dews; Cast Metals Handbook of A.F.A.

The alloys used by engineers and architects in the form of sand castings of bronze, red brass, yellow brass and white alloys containing more than 50% copper given in Tables 1 and 2, have been compiled from about twelve authorities. The values given are average and should not be used for specification limits. These results have been checked by Mr. Lewis H. Fawcett, Metallurgical Testing Laboratory, U. S. Naval Gun Factory, Washington, D. C.

**SELECTION OF ALLOY.**—The alloy which is suitable for any specific purpose is given in Table 1. The following notes cover the items which should be considered by the engineer in selecting and using cast copper-base alloys. Where corrosion resistance is of importance, the manufacturer of the alloy should be consulted as to its suitability under the proposed conditions, or tests made on samples of the alloy in question. The relative value of these alloys for mechanical purposes varies with temperature, and information on this point should be obtained before subjecting them to high-temperature use. The tensile properties of four copper-base alloys at high temperatures are given in Fig. 1.

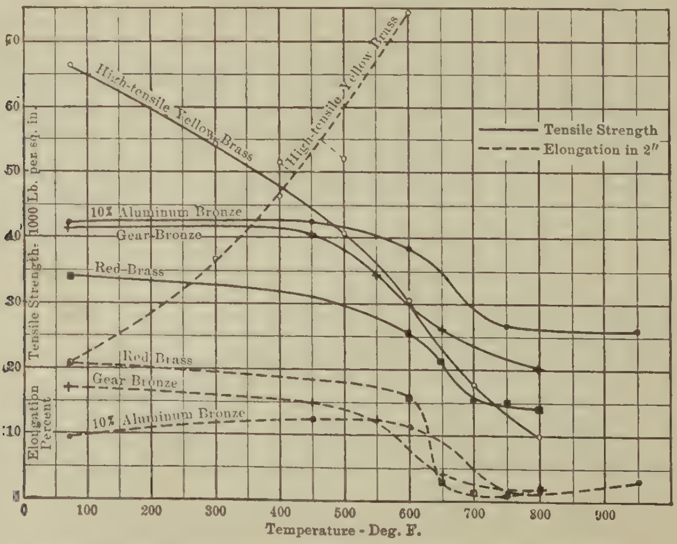

FIG. 1. Tensile Properties of Copper-base Alloys at High Temperature

**Table 1.—Types of Cast Brass or Bronze Alloys for Various Services**

(Refer to Table 2 for specifications of the various alloys listed)

| Quality or Use | Type of Alloy | Alloy No. | Quality or Use | Type of Alloy | Alloy No. |
|---|---|---|---|---|---|
| Abrasion Resisting | Gear Bronze | C12 | Machine Parts | Bronze | C11 |
| | Nickel Bronze | C9 | Miscellaneous Work | Naval Brass | C23 |
| Acid Resisting | See *Corrosion Resisting* | | Non-corrosive | See *Corrosion Resisting* | |
| Air Fittings | Semi-Red Brass | C19 | Ornamental Work | Yellow Brass | C21 |
| Architectural Trim | Ambrac | C24 | | Ambrac | C24 |
| | Benedict Metal | C25 | | Benedict Metal | C25 |
| Bathroom Fittings | Benedict Metal | C25 | Patterns, Metal | Red Brass | C15 |
| Bearing Backs | Red Brass | C15 | Pressure Resisting | Al-Bronze | C2 |
| Bearings | See p. 4–62 | | | Bronze | C11, C14 |
| Bell Metal | High-Tin Bronze | C17 | Propellers | Al-Bronze | C2, C3 |
| Builders' Hardware | Benedict Metal | C25 | | Mn-Bronze | C7 |
| | Yellow Brass | C21 | Pipe Fittings | See *Fittings, Air, Chemical, Gas, Steam, Water* | |
| Castings, Chill | Al-Bronze | C1, C2, C3 | | | |
| Common | Yellow Brass | C22 | | | |
| Light | Yellow Brass | C21 | Pump Bodies | Red Brass | C15 |
| Strong & Tough | Mn-Bronze | C6, C7 | Pumps | | |
| Corrosion Resisting | | | Lubricating Oil | Yellow Brass | C22 |
| Resistant to: | | | Water | Red Brass | C16 |
| Acids | Al-Bronze | C1, C2, C3 | Radiator Valves | Red Brass | C18 |
| | Ambrac | C24 | Resistance to Wear | Al-Bronze | C2 |
| | Everdur | C4 | | Nickel-Bronze | C9 |
| | Mn-Bronze | C7 | Resistance to Corrosion | See *Corrosion Resisting* | |
| Alkalies | Everdur | C4 | | | |
| Atmosphere | Semi-Red Brass | C16 | | | |
| | | | Resistance to Shock | Al-Bronze | C2 |
| Bleaching Solution | Ambrac | C24 | Severe Duty | Bronze | C11 |
| Phenol | Everdur | C4 | Steam Fittings | Bronze | C10, C13 |
| Sea Water | Al-Bronze | C1, C2, C3 | | Red Brass | C15 |
| | Everdur | C4 | Steam Metal | Bronze | C13 |
| | Mn-Bronze | C7 | Strength | Al-Bronze | C1, C2, C3 |
| | Ambrac | C24 | | Mn-Bronze | C6, C7 |
| Water | Bronze | C11 | | Naval Brass | C23 |
| | | | Structural Parts | Mn-Bronze | C7 |
| Electrical Equipment, Overhead | Semi-Red Brass | C16 | Temperature, High | Al-Bronze | C1, C2, C3 |
| Electrical Fittings | Pure Copper | 0 | Toughness | Al-Bronze | C1 |
| Fittings, Air | Semi-Red Brass | C19 | | Mn-Bronze | C6, C7 |
| Chemical | See *Corrosion Resisting* | | Under-Water Fittings | Mn-Bronze | C7 |
| Electrical | Pure Copper | 0 | Valve Bodies, | | |
| Gas | Bronze | C10 | Chemical | Al-Bronze | C2, C3 |
| | Semi-Red Brass | C19 | High Pressure | Bronze | C14 |
| Steam | Bronze | C10, C13 | Valve Stems | Mn-Bronze | C6 |
| | Red Brass | C15 | Valve Facings | Nickel-Bronze | C9 |
| Valve | Red Brass | C18 | Valves | Bronze | C10 |
| Water | Semi-Red Brass | C19 | | Red Brass | C15 |
| Gas Fittings | Semi-Red Brass | C19 | Low Pressure | Red Brass | C18 |
| Gears | Al-Bronze | C2, C3 | Radiator | Red Brass | C18 |
| | Gear Bronze | C12 | Water Fittings | Semi-Red Brass | C19 |
| Hardness | Al-Bronze | C2, C3 | Water Tightness | Bronze | C11 |
| | Silicon-Brass | C5 | Wear Resistance | Al-Bronze | C2 |
| | Mn-Bronze | C6 | | Nickel-Bronze | C9 |
| | Nickel-Bronze | C9 | Worm Wheels | Al-Bronze | C2, C3 |
| | Bell Metal | C17 | | Gear Bronze | C12 |

**FABRICATION.**—Nearly all of the sand-cast copper-base alloys listed are readily machined. Some of the stronger alloys, such as aluminum-bronze and manganese-bronze are tough and difficult to machine, but with proper tool angles and machine conditions it is entirely possible to machine even these. These alloys as a class are not readily welded by any of the commercial methods. Their high heat conductivity dissipates the heat so rapidly that it is difficult to make a proper weld. The classes that can most readily be welded are the yellow brasses or manganese-bronze. The higher zinc content here seems to make welding commercially possible under many conditions.

**INSPECTION AND DEFECTS.**—The principal defects in castings, from an engineering standpoint, are shrinkage, blow holes, sand holes, and oxide inclusions. Shrinkage holes are usually found at a point where the heavy and light sections join. Blow holes are due to a faulty molding and melting condition, and sand holes are caused by imperfect

(*Continued on p.* 4–23)

**Table 2.—Sand-cast Copper-base Alloys**

| Alloy No. | Name | Analysis, percent | | | | | | Tension† | | | Reduction in Area, percent | Compression | | Brinell Hardness | Impact Strength, Izod., ft. lb. | Remarks and Uses |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | 1000 lb. per sq. in. | | Elongation in 2 in., percent | | Deformation Limit, 1000 lb. per sq. in.* | Reduction in Height, % Load = 100,000 lb. per sq. in. | | | |
| | | Cu | Al | Zn | Si | Mn | Fe | Yield Point | Ultimate Strength | | | | | | | |
| | | | | | | | | CAST COPPER | | | | | | | | |
| 0 | Pure copper | 99.6–99.9 | ...... | ...... | ...... | ...... | ...... | ....... | 20–25 | 20–25 | 75 | ....... | ....... | 40 | ........ | Deoxidized with Si, $B_2C_3$ or carbon-free Mn. Electrical uses, if pure. |
| | | | | | | | | HIGH TENSILE BRONZES AND BRASSES | | | | | | | | |
| C1 | Aluminum bronze | 90.6 | 9.4 | ...... | ...... | ...... | ...... | 21.7<br>46.6‡<br>24.0*s* | 67<br>85‡<br>52.5*s* | 36<br>9.5‡<br>17.3*s* | 47<br>20‡<br>32*s* | ....... | ....... | ....... | ........ | These alloys should not be slowly cooled from a high temperature if they are to be stressed at ordinary temperatures. Resistant to cold dilute $H_2SO_4$, cold weak HCl, sea water, etc. Pipe fittings valves and other equipment for chemical industry. |
| C2 | Aluminum bronze | 89 | 10 | ...... | ...... | ...... | 1 | 20–30<br>54‡ | 60–75<br>90‡ | 24<br>10‡ | 27<br>12*t* | 18–20 | 13–15 | 92–120<br>175‡ | ........ | |
| C3 | Aluminum bronze | 88 | 9 | ...... | ...... | ...... | 3 | 30–45<br>40‡ | 70–80<br>84‡ | 25–45<br>30‡ | 20–40<br>25‡ | 14–18 | ....... | 95–120<br>125‡ | ........ | |
| C4 | Everdur | 94.4 | ...... | ...... | 4.5 | 1.1 | ...... | 20–30 | 50–60 | 10–25 | 15–22 | ....... | ....... | 92 | 22 | Resistant to $H_2SO_4$, HCl (in absence of air), sea water, caustic soda, phenol, etc. |
| C5 | Silicon brass | 81.5 | ...... | 14 | 4.5 | ...... | ...... | ....... | 65–75 | 9–10 | ....... | ....... | ....... | 130–150 | 50 | For hardness. |
| C6 | Manganese bronze | 68.5 | 4 | 22 | ...... | 3.0 | 2.5 | 40–45 | 85–100 | 15–20 | 20–30 | 60–70 | ....... | 150–180 | ........ | For strength, toughness, and hardness. |
| C7 | Manganese bronze | 58 | ...... | 39 | ...... | 0–1 | 0.75*t* | 30–45 | 65–75 | 20–40 | ....... | 12–16 | ....... | 90–110 | ........ | For propellers, engine frames, parts requiring strength and toughness. Resistant to sea water and acids. |

| | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| C8 | Silicon brass | 59.7 | ...... | 39.7 | 0.6 | ...... | ...... | ...... | 50–55 | 12–15 | ...... | ...... | ...... | ...... | 25 | |
| C9 | Nickel bronze | 60 | 8 | ...... | ...... | 30 | 2 | ...... | 67 | 0 | ...... | ...... | ...... | 240 | ...... | Valve facing for severe conditions. Abrasion resistant. For hardness. |

BRONZE AND RED BRASS FOR GENERAL ENGINEERING WORK (See note *tt*)

| | | Cu | Sn | Pb | Zn | Ni | P | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| C10 | Bronze | 89 | 8 | 2 | 3 | ...... | 0.03*t* | 16–18 | 30–40 | 15–30 | 15–35 | 12.5–13 | 33–34 | 45–60 | ...... | For valves and fittings for steam, gas, etc. |
| C11 | Bronze | 88 | 8 | 1.0*t* | 4 | 0.75*t* | 0.05*t* | 20–25 | 30–45 | 15–40 | 25–35 | ...... | ...... | 55–75 | ...... | For water-tight castings, underwater fittings, machine parts. Non-corrosive. |
| C12 | Gear bronze | 89 | 11 | ...... | ...... | ...... | 0.3*t* | 18–20 | 30–35 | 5–15 | 6–8 | 16 | 25–27 | 65–85 | ...... | Very resistant to abrasion. For heavy-duty gears and worm wheels. |
| C13 | Bronze | 86 | 8 | 3 | 3 | ...... | 0.03*t* | ...... | 30–35 | 25–30 | 25–30 | ...... | ...... | 55–65 | ...... | Steam fittings. Machines well. |
| C14 | Bronze | 86 | 13 | 1 | ...... | ...... | Trace | 20–35 | 30–45 | 1–8 | ...... | 16–22 | ...... | 85–120 | ...... | Valve bodies subject to high pressure and vibration. |
| C15 | Red brass | 85 | 5 | 5 | 5 | ...... | ...... | 15–19 | 27–33 | 16–20 | 15–20 | 11.5–12 | 36 | 50–60 | ...... | For pump bodies, valves, steam fittings, bearing backs, and metal patterns. |
| C16 | Semi-red brass | 85 | 3 | 3 | 9 | ...... | 0.03*t* | ...... | 30–37 | 20–35 | 20–30 | ...... | ...... | 50–60 | ...... | Very resistant to atmospheric corrosion. For overhead electrical fittings and for oil and water pumps. |
| C17 | Bell metal | 78 | 22 | ...... | ...... | ...... | ...... | 26–32 | 26–32 | 0 | ...... | 24–30 | ...... | 130–160 | ...... | For hardness. Difficult to machine. |
| C18 | Red brass | 75 | 5 | 10 | 10 | ...... | ...... | ...... | 25–30 | 10–15 | 10–15 | 19 | 32 | 50–55 | ...... | For low-pressure valves and fittings. (Radiator valves.) Machines easily. |

*(See footnotes at end of table.)*

*(Table continued on following page)*

**Table 2.—Sand-cast Copper-base Alloys—*Continued***

| Alloy No. | Name | Analysis, percent | | | | | | Tension† | | | Reduction in Area, percent | Compression | | Brinell Hardness | Impact Strength, Izod., ft. lb. | Remarks and Uses |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cu | Sn | Pb | Zn | Ni | Fe | 1000 lb. per sq. in. Yield Point | 1000 lb. per sq. in. Ultimate Strength | Elongation in 2 in., percent | | Deformation Limit, 1000 lb. per sq. in.* | Reduction in Height, % Load = 100,000 lb. per sq. in. | | | |
| | **BRONZE AND BRASS FOR GENERAL ENGINEERING WORK** (*Continued.* See note *tt*) | | | | | | | | | | | | | | | |
| C19 | Semi-red brass | 76 | 2 | 6 | 16 | ...... | ...... | ...... | 25–30 | 20–30 | 20–30 | ...... | ...... | 43–48 | ...... | For air, gas, and water fittings. Machines very easily. |
| | **YELLOW BRASS** | | | | | | | | | | | | | | | |
| C20 | Yellow brass | 77 | ...... | 0.2*t* | 22 | ...... | ...... | 7–11 | 22–28 | 23–35 | 15–25 | ...... | ...... | 40–60 | ...... | For oil feed cylinder top cover, etc., for automobiles. |
| C21 | Yellow brass | 69 | 1 | ...... | 30 | ...... | ...... | 10–20 | 30–40 | 20–40 | 25–35 | 6 | ...... | 45–60 | ...... | For light castings, builders' hardware, ornamental work. |
| C22 | Yellow brass | 64 | 1 | 2 | 33 | ...... | 0.50 | 15 | 30–35 | 25–35 | 20–30 | ...... | ...... | 40–50 | ...... | For common castings where cheapness and good machining properties are main considerations. |
| C23 | Naval brass | 62 | 1 | ...... | 37 | ...... | ...... | 25–30 | 35–50 | 15–30 | 20–30 | ...... | ...... | ...... | ...... | Miscellaneous work for strength. |
| | **WHITE ALLOYS** | | | | | | | | | | | | | | | |
| C24 | Ambrac | 64.5 | 3.5 | 5.0 | 7.5 | 20.0 | ...... | 20–25 | 40–45 | 15–20 | ...... | ...... | ...... | ...... | ...... | Resistant to salt spray, dilute $H_2SO_4$, bleaching solution. Suitable for architectural trim. |
| C25 | Benedict metal | 55.0 | ...... | 12.5 | 20.0 | 12.5 | ...... | 15–20 | 30–35 | 15–20 | ...... | ...... | ...... | ...... | ...... | Resistant to atmospheric corrosion. For architectural trim, bathroom and kitchen fittings, builders' hardware. |

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* Load in pounds per square inch producing 0.001 in. compression in a specimen 1 in. long and 1 sq. in. in cross-section. † Proof stress is frequently taken as 75 percent of yield point. ‡ Quenched from 1472 °F. *s* Slowly cooled from 1472 °F. *t* Maximum. *tt* Should not contain over 0.25 percent Fe.

molding. Oxide inclusions may be caused by bad material but also, in some alloys, by excessive agitation during the pouring operation either inside or outside the mold.

**CORROSION.**—Certain alloys resist corrosion better than others, and in Table 2, under the heading "Remarks," this condition is noted in some cases. As a class these alloys are considered non-corrosive with reference to atmospheric conditions, but they vary widely when exposed to chemical agents. It is difficult to predict resistance to corrosion. In nearly every instance a trial is necessary to be certain.

**DESIGN OF CASTINGS.**—The design of castings is extremely important. It is well known that castings can be designed that are impossible to make in a foundry. The ideal casting is one which has a fairly uniform metal section with easy gradations from light to heavy parts. If engineers would keep this in mind in designing, much trouble would be saved the factory and the user. The variation in thickness in different parts of the same casting causes variations in strength, sometimes amounting to 50%. It is, therefore, impossible to predict exactly the strength of metal at any particular point in a casting without actually cutting a test piece from that point. As this usually is not practicable, standard test pieces are used as indicators. These test pieces are made under uniform conditions and show the properties of the alloy when cast in the best manner. From the results of mechanical tests of these test pieces specifications are drawn. If these specification values are used for comparison with results from test pieces cut from the castings of variable design a discrepancy usually will result. Frequently, engineers draw the conclusion that castings do not meet specification values. The error made here is in judging these results as comparable. The standard test piece indicates the quality of the material used, but the results from it should not be used to check values in the casting itself because of the variations noted above.

## 2. WROUGHT BRASSES, BRONZES AND NICKEL-SILVER ALLOYS

By W. H. Bassett *

The nomenclature used in the non-ferrous metals trade is confusing to those who are not familiar with the business, but through long usage it has become so thoroughly rooted that it is practically impossible to change it. The trade names have an actual value to the producers of various materials, and they would, therefore, naturally resist changes.

All questions of properties of engineering material are tied up with factors of safety, factors of reliability, conditions of exposure (which are always local) and similar matters. The requirements of special alloys for specific purposes may be considered from the viewpoint of the physical requirements or resistance to corrosion or both. The physical requirements and properties may be studied in the laboratory. The question of corrosion is less easily solved, due to the difficulty of reproducing in the laboratory the conditions to which the metal may be exposed in service.

The joint use of "dissimilar" alloys has caused much trouble. For instance, the use of one of the so-called "stronger brasses" (one of the brasses sometimes called bronzes which contains a considerable percentage of zinc together with tin or iron or a combination of these elements) in connection with a straight copper-tin bronze in an electrolyte such as sea water will result in the weakening of the alloy containing the larger proportion of zinc. Where brass strainers have been held in position with copper or gun-metal bolts and where brass screens have been placed in position with copper rivets, galvanic action has occurred, destroying the alloy containing the greater percentage of zinc.

Before proceeding with design, engineers not thoroughly familiar with the composition and properties of the alloys with which they propose to deal should consult reputable metallurgists, giving a thorough review of the conditions under which the material is to be used. The question of conditions of use is very important and should be carefully considered.

The so-called "season cracking" of brasses and bronzes has given much trouble in certain places. This phenomenon is due to the internal stress in the alloy brought about by cold working, such as rolling and drawing, and corrosive action. Copper and the copper-zinc alloys containing 20% or less of zinc are not subject to this trouble. The stronger copper-zinc alloys, as well as those containing tin and other elements, may be so prepared as to prevent this difficulty if the conditions of exposure are understood beforehand.

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* This material was prepared originally by Mr. Bassett. After his death, Table 4 was revised by his associates, Messrs. H. G. Jennison and W. R. Hibbard.

## Table 3.—Type of Wrought Brass or Bronze Alloys for Various Services

| Use or Quality | Alloy | Alloy No. |
|---|---|---|
| Architectural shapes | Extruded | B66 |
| Architectural work | Red brass 85% | B6 |
| Bearings | Beryllium copper | B67 |
| Bolts | Muntz metal | B16 |
| " | Naval brass | B28 |
| " | Tobin bronze | B29 |
| Bolt stock (high ductility) | Everdur (B) | B61 |
| Brazed articles | Brazing brass | B8 |
| Bullet jackets | 15% Cupro-nickel | B41 |
| Bushings, low duty | Beryllium copper | B67 |
| Caps | Cap gilding | B17 |
| Cartridges | Cartridge brass | B10, B11 |
| " | Eyelet brass | B12 |
| " | Drawing or spinning brass | B13 |
| Chain, bronze | Phosphor bronze A | B33 |
| " " | Phosphor bronze D | B36 |
| " window weight | Phosphor bronze A | B33 |
| Clock parts | Clock brass | B23 |
| Color | Red brass 85% | B6 |
| " | Red brass 80% | B7 |
| " | 5% Aluminum bronze | B52 |
| Commutator segments | Hitenso BB | B63 |
| Condenser tubes | Admiralty | B27 |
| " " | Super-nickel | B39 |
| Contact fingers | Phosphor bronze | B35 |
| " " | Hitenso BB | B63 |
| Corrosion resistance | Commercial bronze 90% | B5 |
| " " | Red brass 85% | B6 |
| " " | Red brass 80% | B7 |
| " " | 20% Cupro-nickel | B40 |
| " " | Ambrac A | B44 |
| " " | 8% Aluminum bronze | B53 |
| " " | 8% Aluminum bronze with iron | B54 |
| " " | 10% Auminum bronze | B55 |
| " " | Calsun bronze | B57 |
| " " | Manganese bronze | B58, B59 |
| " " | Everdur A | B60 |
| " " | Everdur B | B61 |
| " " | Tempaloy 917 | B65 |
| Corrosion resistance, atmospheric | Commercial bronze, 90% | B5 |
| do | Red brass 80% | B7 |
| do | Ambrac A | B44 |
| Decorations | 15% Nickel-silver | B48 |
| Diaphragms | Phosphor bronze C | B35 |
| " | Beryllium copper | B67 |
| " to withstand pressure | 5% Aluminum bronze | B52 |
| do | 8% Aluminum bronze | B53 |
| Drawing | 15% Nickel-silver | B47 |
| " deep | Drawing or spinning brass | B13 |
| Electrical instruments | 30% Nickel-silver | B42 |
| " parts | Cap gilding | B17 |
| " parts, high-conductivity | Copper | B1 |
| " purposes | Signal bronze | B32 |
| " " | High-strength bronze | B38 |
| " " | Calsun bronze | B57 |
| " resistance | 30% Nickel-silver | B42 |
| " wire and cable, etc. | Hitenso A | B62 |
| do | Hitenso BB | B63 |
| do | Hitenso C | B64 |
| Elevated temperatures | Avialite | B56 |
| Erosion resistance | 20% Cupro-nickel | B40 |

| Use or Quality | Alloy | Alloy No. |
|---|---|---|
| Eyelets | Eyelet brass | B12 |
| Fatigue resistance | Beryllium copper | B67 |
| Free cutting | Leaded commercial bronze | B20 |
| " " | Leaded red brass 80% | B21 |
| " " | Leaded nickel-silver | B49 |
| Free milling | Clock brass | B23 |
| Forging, hot | Forging brass | B24 |
| Hardware | Red brass 85% | B6 |
| " | Leaded commercial bronze | B20 |
| Heading operations | Brass wire | B15 |
| Hinges | Butt brass | B19 |
| Hose, flexible metal | Special bronze | B31 |
| Jewelry trade | Bronze, commercial, 95% | B4 |
| " " | Leaded commercial bronze | B20 |
| Lamp fixtures | Commercial high brass | B14 |
| Machine work, automatic | Free-cutting rod | B25 |
| Machining | Forging brass | B24 |
| " | Oreide | B26 |
| " | Phosphor bronze B | B34 |
| " | Free-cutting phosphor bronze | B37 |
| Meter parts | Clock brass | B23 |
| Nuts | Muntz metal | B16 |
| " | Naval brass | B28 |
| " | Tobin bronze | B29 |
| Ornamental work | Commercial high brass | B14 |
| " " | Ambrac A | B44 |
| Pins | Brass wire | B15 |
| Pipe | Deoxidized copper | B2 |
| " | Red brass 85% | B6 |
| " | Tube brass | B18 |
| Piston rods | Naval brass | B28 |
| " " | Tobin bronze | B29 |
| Plates | Naval brass | B28 |
| " | Tobin bronze | B29 |
| Propeller shafts | Naval brass | B28 |
| " " | Tobin bronze | B29 |
| Plumbing | 15% Nickel-silver | B48 |
| Primers | Cartridge brass | B10 |
| Primer gilding | Primer gilding | B3 |
| Radiator, automobile | Commercial bronze 90% | B5 |
| " " | Commercial high brass | B14 |
| Radio parts | Phosphor bronze C | B35 |
| Rivets | Brass wire | B15 |
| " | Leaded red brass 80% | B21 |
| Rod, brass | Spring brass | B9 |
| " copper | Copper | B1 |
| " welding | See Welding rod | |
| Rods, piston | Naval brass | B28 |
| " " | Tobin bronze | B29 |
| Screws | Brass wire | B15 |
| Sea-water, resistance to | Admiralty | B27 |
| Shapes, special | Leaded brass | B22 |
| Sheathing | Muntz metal | B16 |
| Sheet metal (high ductility) | Everdur (B) | B61 |
| Sheets | Copper | B1 |
| " | 15% Nickel-silver | B48 |
| Shells, drawn | Eyelet brass | B12 |
| Shot shells | Cartridge brass | B10 |
| Silver-plated ware | 15% Nickel-silver | B47 |
| " " " | 10% Nickel-silver | B50 |
| " " " | 5% Nickel-silver | B51 |
| Silver-soldered articles | Brazing brass | B8 |
| Spinning | 15% Nickel-silver | B47 |

**Table 3.—Type of Wrought Brass or Bronze Alloys for Various Services**—*Continued*

| Use or Quality | Alloy | Alloy No. |
|---|---|---|
| Springs | Phosphor bronze A | B33 |
| " | Phosphor bronze D | B36 |
| " | Beryllium copper | B67 |
| Spring-stock | Spring brass | B9 |
| Strength | Manganese bronze | B58, B59 |
| " | Everdur (A) | B60 |
| " | Everdur (B) | B61 |
| " | Tempaloy 917 | B65 |
| Structural work | Manganese bronze | B58, B59 |
| Switches, electric | Phosphor bronze A | B33 |
| " " | Phosphor bronze C | B35 |
| Tableware, plated and unplated | 25% Nickel-silver | B43 |
| do | 18% Nickel-silver | B45 |
| Threading and similar operations | Butt brass | B19 |
| Tubes | Copper | B1 |
| " | Deoxidized copper | B2 |
| " | Tube brass | B18 |
| " Bourdon | Beryllium copper | B67 |
| " condenser | Admiralty | B27 |
| " " | Super-nickel | B39 |
| Tubes, seamless | Cartridge brass | B10 |
| " white metal | 15% Nickel-silver | B48 |
| Turbine blades | Spring brass | B9 |
| " " | 20% Cupro-nickel | B40 |
| Valve seats (airplane engines) | Avialite | B56 |
| Watch parts, etc. | Leaded Nickel-silver | B49 |
| Wear resistance | 8% Aluminum bronze | B53 |
| " " | 8% Aluminum bronze with iron | B54 |
| " " | 10% Aluminum bronze | B55 |
| Welding rod | Deoxidized copper | B2 |
| " " | Naval brass | B28 |
| " " | Tobin bronze | B29 |
| " " | Everdur (A) | B60 |
| Wire | Copper | B1 |
| " | 15% Nickel-silver | B48 |
| " Fourdrinier | Red brass 80% | B7 |
| " " | Fourdrinier wire | B30 |
| " pin | Muntz metal | B16 |
| " window screen | Commercial bronze 90% | B5 |

Table 4 (p. 4–26) gives an idea of what may be expected in the way of physical properties from some of the more common combinations of metals, but should be used as a general guide only. Table 3 is an index to the alloys of Table 4.

Table 4 gives the more common wrought alloys with their ordinary physical properties, both "Hard Rolled" or "Drawn" and "Annealed" or "Soft." The figures are averages that ordinarily may be expected in practice but may be modified by "Extra Hard," "Half Hard" or "Quarter Hard" rolling. In other words, almost any physical properties between the figures for "Hard" and "Soft" may be obtained by appropriate working, and higher figures for tensile strength, or hardness, may be obtained by a greater amount of working. The alloys have been chosen to represent the more commonly used copper-zinc, copper-zinc-nickel, copper-tin and copper-aluminum mixtures. The alloys intermediate in composition between those chosen have physical properties lying between those given for the nearest compositions.

# ALUMINUM AND ITS ALLOYS

**By Zay Jeffries**

## 1. PROPERTIES OF ALUMINUM

Aluminum is one of the lightest of the metals used for structural purposes. The principal source of aluminum is bauxite, a hydrated oxide of aluminum, which is reduced to pure alumina, $Al_2O_3$, by treatment with caustic soda, from which the metal is obtained by electrolytic process. Impurities in the alumina, as silicon and iron oxide, will appear in the metal. Commercially pure aluminum contains from 99 to 99.5% aluminum, but grades containing 99.8% or more of aluminum are produced.

**CORROSION-RESISTANT CHARACTERISTICS.**—Aluminum is resistant to most atmospheric influences, especially inland industrial atmospheres. Paint or other protection, however, generally is advisable for permanent structures exposed to seacoast atmospheres. Examples of the resistance of unpainted aluminum and its alloys to atmospheric conditions are the relatively high-purity aluminum transmission lines and outside bus bar installations; aluminum-silicon alloy castings, used in large buildings as spandrels and other ornamental parts; wrought products, as window frames, store fronts and other parts of buildings. Aluminum is resistant to distilled water, many food products, fruit juices, milk and beer, and atmospheric nitric and glacial acetic acids. It is readily dissolved by hydrochloric and hydrofluoric acids, and caustic solutions.

The high corrosion resistance of aluminum is due to the protective effect of the oxide coating. In general, the highest purity aluminum has the highest corrosion resistance. Nearly all other metals present, either as impurities or alloying elements, tend to lower corrosion resistance. Manganese, chromium, and magnesium produce the most corrosion-resistant alloys. Silicon affects the corrosion resistance slightly, but it conveys other

(*Continued on p.* 4–34)

**Table 4.—Chemical and Physical Properties**

(Variations must be

| | Material | Form§ | Approximate Composition, percent | | | | | Tensile Strength, lb. per sq. in. | | Elongation in 2 in., percent | | Yield Point, lb. per sq. in. | | Johnson's Elastic Limit, lb. per sq. in. | | Modulus of Elasticity, lb. per sq. in. ×$10^{-6}$, Hard* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Copper | Zinc | Lead | Tin | | Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | |
| B 1 | Copper | S | 99.90+ | ..... | ..... | ..... | ..... | 51,000 | 32,500 | 4 | 37 | 46,000[l] | 11,000[l] | ....... | 5,000 | 16.0 |
| | | W | 99.90+ | ..... | ..... | ..... | ..... | 60,000 | 38,000 | 3[e] | 36[e] | 37,000[l] | ....... | 36,500 | ...... | ..... |
| | | R | 99.90+ | ..... | ..... | ..... | ..... | 50,000 | 32,000 | 18 | 38 | 44,000[l] | 14,000[l] | 35,000 | ...... | ..... |
| B 2 | Deoxidized Copper | T | 99.90+ | ..... | ..... | ..... | Phosphorus. present | 50,000 | 35,000 | 10 | 35 | 46,000[l] | ....... | ....... | ...... | ..... |
| | | S | 99.90+ | ..... | ..... | ..... | present | 55,000 | 35,000 | 5 | 35 | 42,000[l] | 15,000[l] | ....... | ...... | ..... |
| | | R | 99.90+ | ..... | ..... | ..... | present | 58,000 | 35,000 | 5 | 38 | ....... | 14,000[l] | ....... | ...... | 16.0 |
| | | W | 99.90+ | ..... | ..... | ..... | present | 60,000 | 35,000 | 2.6[e] | 35 | ....... | ....... | ....... | ...... | ..... |
| B 3 | Primer Gilding | S | 97.00 | 3.00 | ..... | ..... | ..... | 55,000 | 35,000 | 5 | 37 | 38,000[g] | ....... | ....... | ...... | ..... |
| B 4 | Commercial Bronze, 95% | S | 95.00 | 5.00 | ..... | ..... | ..... | 55,000 | 35,000 | 5 | 38 | 34,000[l] | 11,000[l] | ....... | ...... | 15.0 |
| B 5 | Commercial Bronze, 90% | S | 90.00 | 10.00 | ..... | ..... | ..... | 67,000 | 37,000 | 3 | 40 | 47,000[l] | 11,000[l] | ....... | ...... | 15.0 |
| B 6 | Red Brass, 85% | S | 85.00 | 15.00 | ..... | ..... | ..... | 75,000 | 42,000 | 4 | 43 | 61,000[l] | 17,000[l] | ....... | ...... | 15.0 |
| | | T | 85.00 | 15.00 | ..... | ..... | ..... | 68,000 | 42,000 | 6 | 42 | 56,000[l] | 18,000[l] | 49,000 | ...... | ..... |
| B 7 | Red Brass, 80% | S | 80.00 | 20.00 | ..... | ..... | ..... | 85,000 | 43,000 | 4 | 50 | ....... | ....... | ....... | ...... | 15.0 |
| | | W | 80.00 | 20.00 | ..... | ..... | ..... | 125,000 | 49,000 | 2[e] | 43[e] | ....... | ....... | ....... | ...... | ..... |
| B 8 | Brazing Brass | S | 75.00 | 25.00 | ..... | ..... | ..... | 80,000 | 47,000 | 5 | 45 | ....... | ....... | ....... | ...... | ..... |
| B 9 | Spring Brass | S | 72.00 | 28.00 | ..... | ..... | ..... | 76,000 | 47,000 | 4 | 55 | 35,000[l] | ....... | ....... | ...... | 14.0 |
| B10 | Cartridge Brass | S | 70.00 | 30.00 | ..... | ..... | ..... | 86,000 | 45,000 | 4 | 50 | ....... | ....... | ....... | ...... | ..... |
| B11 | Cartridge Brass | S | 69.00 | 31.00 | ..... | ..... | ..... | 85,000 | 46,000 | 4 | 58 | ....... | ....... | ....... | ...... | ..... |
| B12 | Eyelet Brass | S | 68.00 | 32.00 | ..... | ..... | ..... | 78,000 | ...... | 5 | 58 | 50,000[l] | ....... | ....... | ...... | ..... |
| B13 | Drawing or spinning brass | S | 66.67 | 33.33 | ..... | ..... | ..... | 76,000 | 46,000 | 5 | 52 | ....... | ....... | ....... | ...... | ..... |
| B14 | Commercial high brass | S | 65.00 | 35.00 | ..... | ..... | ..... | 76,000 | 45,000 | 5 | 60 | ....... | ....... | 48,000 | 7,500 | 14.0 |
| | | R | 65.00 | 35.00 | ..... | ..... | ..... | 70,000 | 45,000 | 15 | 50 | ....... | 12,000[l] | ....... | ...... | ..... |
| B15 | Brass wire | R | 63.00 | 37.00 | ..... | ..... | ..... | 70,000 | 50,000 | 12 | 50 | ....... | ....... | ....... | ...... | 14.0 |
| | | S | 63.00 | 37.00 | ..... | ..... | ..... | 84,000 | 48,000 | 4 | 50 | ....... | ....... | ....... | ...... | ..... |
| | | W | 63.00 | 37.00 | ..... | ..... | ..... | 125,000 | 50,000 | 4[e] | 50[e] | ....... | ....... | ....... | ...... | ..... |

* For some alloys the figures are for a temper slightly different from that commonly known as "hard."

† Compared to water at 4° C.

‡ Soft.

§R = Rod. S = Sheet. T = Tube. W = Wire.

[a] Temper not known.

[b] Determination.

[c] Circular 63, U. S. Bureau of Standards.

[d] Scientific Paper 410, U. S. Bureau of Standards.

[e] Elongation of wire, percent in 10 in.

[f] Corning Glass Works.

[g] Yield point taken as the load producing an extension under stress of 0.75 percent.

[h] Jenkins and Hanson constitution diagram.

[j] Average linear coefficient per deg. C. from 25 to 300 deg. C. Tests on rod. Scientific Paper 410, U. S. Bureau of Standards.

**of Various Wrought Copper-base Alloys**

expected in practice)

| Shearing Strength, lb. per sq. in. | | Brinell Hardness No. 10-mm. Ball, 500-kg. Load | | Rockwell Hardness No. B 1/16-in. Ball, 100-kg. | | Bend Test, deg. | | Melting point, deg. F. | Specific Gravity † | Density, lb. per cu. in. | Coefficient of Expansion $\times 10^7$ (*j*) | Electrical Properties at 20° C. Resistivity, ohms (mil-ft.) | Electrical Properties at 20° C. Conductivity, percent I.A.C.S. | Thermal Conductivity (*u*) | Uses—Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | | | | | | | | |
| ....... | 21,000 | 103 | 42 | 58 | Too Soft | 180 | 180 | 1981*c* | 8.89 | 0.3217 | 98 | 10.371 | 100.0 | 2678.02 | For electrical parts requiring high conductivity; also for rods, sheets, tubes and wire for commercial purposes. |
| ....... | ...... | ... | ... | ... | ..... | 180 | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | ... | ..... | 180 | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | 58 | Too Soft | 180 | 180 | 1981*b* | 8.93*b* | 0.323 | 98 | ...... | ...... | ....... | Copper pipe and tube. Tougher than electrolytic copper and has better corrosion resistance. |
| ....... | ...... | ... | ... | 61 | ..... | 180 | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | ... | ..... | 180 | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | Welding rod. May also be deoxidized with silicon, manganese, calcium boride, and other deoxidizers. |
| ....... | ...... | ... | ... | ... | ..... | 180 | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | 107 | 42 | 65 | Too Soft | 180 | 180 | 1967*x* | 8.890*d* | 0.321 | 98 | 15.62 | 66.40 | 1712.77 | Primer gilding. |
| ....... | ...... | 110 | 43 | 68 | ..... | 180 | 180 | 1949*x* | 8.866*d* | 0.320 | 101 | 18.98 | 54.6 | 1672.13 | For jewelry trade and manufacturing where soft, pliable metal is required. |
| ....... | 25,000 | 115 | 50 | 75 | 1 | 180 | 180 | 1913*x* | 8.804*d* | 0.318 | 101 | 25.36 | 40.90 | 1294.74 | For window screen wire and automobile radiators on account of resistance to corrosion and atmospheric action. |
| ....... | ...... | 135 | 52 | 82 | 10 | 180 | 180 | 1868*x* | 8.745*b* | 0.316 | 104 | 28.03 | 37.0 | 1103.14 | For architectural work on account of color and resistance to corrosion; also for hardware and pipe. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| 43,000*r* | 27,000 | 150 | 53 | 86 | 11 | 180 | 180 | 1832*x* | 8.667*d* | 0.313 | 106 | 31.95‡ | 32.5‡ | 972.51 | For its color and resistance to corrosion and atmospheric action. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | 36.90*y* | 28.1*y* | ....... | Fourdrinier wire. |
| ....... | ...... | 157 | 53 | 87 | ..... | ... | 180 | 1796*x* | 8.594*d* | 0.310 | 109 | 34.6‡ | 30.0‡ | 899.93 | For parts or articles to be brazed or silver soldered, etc. |
| ....... | ...... | 158 | 53 | 88 | 20 | ... | 180 | 1769*x* | 8.553*d* | 0.309 | 110 | 36.3 | 28.60 | 856.39 | For turbine blades, best spring stock, brass rod, etc. |
| ....... | ...... | 157 | 53 | 87 | ..... | ... | 180 | 1751*x* | 8.528*d* | 0.308 | 111 | 37.61 | 27.58 | 841.87 | For primers, shot shells, cartridges, seamless tubes, etc. |
| ....... | ...... | 156 | 53 | 87 | 22 | ... | 180 | 1742*x* | ...... | ...... | .... | 37.55 | 27.60 | 841.87 | For cartridges, etc. |
| ....... | ...... | 156 | 53 | 87 | 22 | ... | 180 | 1733*x* | 8.50*b* | 0.307 | .... | 37.95 | 27.30 | 838.97 | For eyelets, cartridges, drawn shells, etc. |
| ....... | 30,000 | 153 | 52 | 86 | 20 | ... | 180 | 1720*x* | 8.476*d* | 0.306 | 112 | 40.12 | 25.85 | 833.16 | For deep drawing, cartridges, etc. |
| ....... | ...... | 153 | 52 | 85 | 30 | ... | 180 | 1706*x* | 8.460*d* | 0.306 | 112 | 38.68 | 26.8 | 827.36 | For a large variety of articles, lamp fixtures, automobile radiators, and for ornamental purposes. Not good for exposure to weather. |
| ....... | 28,000 | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | ... | ..... | ... | 180 | 1688*x* | 8.437*d* | 0.305 | 114 | 39.97 | 25.95 | 827.36 | For rivets, pins, screws, and other heading operations. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |

*k* At 18.1 deg. C.

*l* Yield point taken as the load producing an extension under stress of 0.50%. Yield point up to 1934 was taken as the load producing an extension under stress of 0.75%. At that time, however, the A.S.T.M. and Federal Specifications Board adopted the figure of 0.50%.

*m* Cold worked and heat-treated.

*n* Guertler-Tammann constitution diagram.

*p* Annealed, quenched and heat-treated.

*r* Smith constitution diagram.

*t* Stockdale constitution diagram.

*u* B.t.u. per sq. ft. per hr. per 1 in. thick per 1° F., at 68° F.

*v* Tafel constitution diagram.

*x* Bauer and Hansen constitution diagram.

*y* Hard at 77° F.

*z* Heycock-Neville constitution diagram.

(*Table continued on following page*)

**Table 4.—Chemical and Physical Properties of**

(Variations must be

| | Material | Form§ | Approximate Composition, percent | | | | | Tensile Strength, lb. per sq. in. | | Elongation in 2 in., percent | | Yield Point, lb. per sq. in. | | Johnson's Elastic Limit, lb. per sq. in. | | Modulus of Elasticity, lb. per sq. in.×10⁻⁶, Hard* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Copper | Zinc | Lead | Tin | | Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | |
| B16 | Muntz metal | S | 60.00 | 40.00 | ..... | ..... | ..... | 80,000 | 57,000 | 9.5 | 48 | ....... | 19,000[l] | ....... | ...... | 12.8 |
| B17 | Cap gilding | S | 90.00 | 9.60 | 0.40 | ..... | ..... | 65,000 | 39,000 | 4 | 35 | ....... | ....... | ....... | ...... | ..... |
| B18 | Tube brass...... | S | 67.50 | 32.00 | 0.50 | ..... | ..... | 85,000 | 43,000 | 3 | 46 | ....... | ....... | ....... | ...... | ..... |
| | | T | 67.50 | 32.00 | 0.50 | ..... | ..... | 50,000 | 44,000 | 5 | 45 | ....... | 16,000[l] | ....... | ...... | 14.0[‡] |
| B19 | Butt brass | S | 64.00 | 35.00 | 1.00 | ..... | ..... | 80,000 | 45,000 | 5 | 60 | ....... | ....... | ....... | ...... | ..... |
| B20 | Leaded commercial bronze | R | 88.50 | 10.00 | 1.50 | ..... | ..... | 60,000 | 35,000 | 3 | 30 | ....... | ....... | ....... | ...... | 15.0 |
| B21 | Leaded red brass, 80% | R | 78.50 | 20.00 | 1.50 | ..... | ..... | 60,000 | 40,000 | 10 | 35 | ....... | ....... | ....... | ...... | ..... |
| B22 | Leaded brass | R | 69.00 | 29.50 | 1.50 | ..... | ..... | 65,000 | 45,000 | 10 | 34 | 33,000[l] | ....... | ....... | ...... | ..... |
| B23 | Clock brass | S | 61.50 | 37.00 | 1.50 | ..... | ..... | 80,000 | 45,000 | 4 | 40 | ....... | ....... | ....... | ...... | ..... |
| B24 | Forging brass | R | 60.00 | 38.00 | 2.00 | ..... | ..... | 70,000 | 50,000 | 10 | 45 | 30,000[l] | 21,000[l] | ....... | ...... | ..... |
| B25 | Free-cutting yellow brass | R | 62.00 | 35.00 | 3.00 | ..... | ..... | 62,000 | 47,000 | 20 | 60 | 45,000[l] | 27,000[l] | ....... | ...... | 15.0 |
| B26 | Oreide | S | 87.25 | 11.50 | .... | 1.25 | ..... | 80,000 | 45,000 | 4 | 40 | ....... | ....... | ....... | ...... | ..... |
| B27 | Admiralty | S | 70.00 | 29 00 | ..... | 1.00 | ..... | 95,000 | 45,000 | 5 | 60 | ....... | ....... | ....... | ...... | ..... |
| | | W | 70.00 | 29.00 | ..... | 1.00 | ..... | 125,000 | 57,000 | 2[e] | 42[e] | ....... | ....... | ....... | ...... | ..... |
| | | T | 70.00 | 29.00 | ..... | 1.00 | .... | ....... | 55,000 | .... | 60 | ....... | ....... | ....... | ...... | ..... |
| B28 | Naval brass | R | 60.00 | 39.25 | ..... | 0.75 | ..... | 62,000 | 54,000 | 25 | 40 | ....... | ....... | ....... | ...... | ..... |
| B29 | Tobin bronze | R | 60.00 | 39.25 | ..... | 0.75 | ..... | 75,000 | 54,000 | 25 | 50 | 55,000[l] | 24,000[l] | ....... | ...... | 15.0 |
| | | S | 60.00 | 39.25 | ..... | 0.75 | ..... | 90,000 | 54,000 | 4 | 40 | ....... | 24,000[l] | 72,500 | 16,000 | ..... |
| B30 | Fourdrinier wire | W | 81.00 | 18.75 | ..... | 0.25 | ..... | ....... | 49,000 | .... | 43[e] | ....... | ....... | ....... | ...... | ..... |
| B31 | Special bronze | S | 98.75 | ..... | ..... | 1.25 | ..... | 65,000 | 40,000 | 4 | 48 | ....... | ....... | ....... | ...... | ..... |
| B32 | Signal bronze | W | 98 25 | ..... | ..... | 1.75 | ..... | 100,000 | 50,000 | 3[e] | 33[e] | ....... | ....... | 73,000 | ...... | ..... |
| B33 | Phosphor bronze, A | S | 96.00 | ..... | ..... | 3.75 | P<br>0.25 | 90,000 | 45,000 | 4 | 50 | ....... | 18,000[l] | 68,500 | 11,100 | 15.0 |
| | | S | 95.00 | ..... | ..... | 5.00 | ..... | 100,000 | 50,000 | 3 | 55 | 75,000[l] | 20,000[l] | 74,000 | ...... | 15.0 |
| B34 | Leaded phosphor bronze, B | R | 94.00 | ..... | 1.00 | 5.00 | ..... | ....... | 50,000 | .... | 40 | ....... | 17,000[l] | ....... | ...... | ..... |
| B35 | Phosphor bronze, C | S | 92.00 | ..... | ..... | 8.00 | ..... | 110,000 | 55,000 | 3 | 70 | 70,000[l] | 22,000[l] | ....... | ...... | 14.0 |
| B36 | Phosphor bronze, D | S | 89.50 | ..... | ..... | 10.50 | ..... | 115,000 | 60,000 | 5 | 65 | 75,000[l] | 37,000[l] | 91,000 | ...... | ..... |
| B37 | Free-cutting phosphor bronze | R | 88.00 | 4.00 | 4.00 | 4.00 | ..... | 60,000 | ....... | 20 | .... | 46,000[l] | ....... | ....... | ...... | 15.0 |
| B38 | High-strength bronze | W | 97.25 | ..... | ..... | 2.00 | Si<br>0.75 | 120,000 | 45,000 | 3[e] | 36 | ....... | ....... | ....... | ...... | ..... |

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* For some alloys the figures are for a temper slightly different from that commonly known as "hard."

† Compared to water at 4° C.

‡ Soft.

§ R = Rod. S = Sheet. T = Tube. W = Wire.

[a] Temper not known.

[b] Determination.

[c] Circular 63, U. S. Bureau of Standards.

[d] Scientific Paper 410, U. S. Bureau of Standards.

[e] Elongation of wire, percent in 10 in.

[f] Corning Glass Works.

[g] Yield point taken as the load producing an extension under stress of 0.75 percent.

[h] Jenkins and Hanson constitution diagram.

[j] Average linear coefficient per deg. C. from 25 to 300 deg. C. Tests on rod. Scientific Paper 410, U. S. Bureau of Standards.

**Various Wrought Copper-base Alloys**—(*Continued*)

expected in practice)

| Shearing Strength, lb. per sq. in. | | Brinell Hardness No. 10-mm. Ball, 500-kg. Load | | Rockwell Hardness No. B 1/16-in. Ball, 100-kg. | | Bend Test, deg. | | Melting point, deg. F. | Specific Gravity † | Density, lb. per cu. in. | Coefficient of Expansion × $10^7$ (*j*) | Electrical Properties at 20° C. | | Thermal Conductivity (*u*) | Uses—Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | | | | | Resistivity, ohms (mil-ft.) | Conductivity, percent I.A.C.S. | | |
| ....... | ...... | 155 | 80 | 87 | 42 | ... | 180 | 1661*x* | 8.396 | 0.303 | 116 | 36.25 | 28.60 | 870.90 | For bolts, nuts, sheathing, pin wire, etc. |
| ....... | ...... | ... | ... | .. | ..... | ... | ... | ....... | ...... | ...... | .... | 24.63 | 42.10 | ....... | For caps and electrical parts. |
| ....... | ...... | 158 | 52 | 90 | 15 | ... | ... | ....... | 8.495*b* | 0.307 | .... | 38.68 | 26.8 | ....... | Pipe and tube. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | 85 | 15 | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | For hinges, etc. Also for threading and similar operations. |
| 35,000 | ...... | 104 | ... | 58 | ..... | ... | ... | ....... | 8.830*d* | 0.319 | 102 | 25.61*y* | 40.50*y* | 1254.10 | For hardware, jewelry, etc. Free cutting. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | 8.698*d* | 0.314 | 107 | 35.87*y* | 28.91*y* | ....... | For rivets, etc. Free cutting. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | 8.562*d* | 0.309 | 111 | 37.65 | 27.55 | ....... | For special shapes where high lead is detrimental to the bending or working of the stock. |
| ....... | ...... | 142 | 54 | 87 | 13 | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | For clock and meter parts, pinions and other articles where free milling is required. |
| ....... | ...... | 122 | 82 | ... | ..... | ... | 120 | ....... | 8.44*d* | 0.305 | .... | 39.10‡ | 26.5‡ | 748.97 | For hot forging. Machines easily. |
| 36,000 | 28,000 | 120 | 54 | 77 | 16 | 120 | 180 | 1625*b* | 8.489*d* | 0.307 | 113 | 41.46 | 25.0 | 748.97 | For automatic machine work. |
| ....... | ...... | ... | ... | ... | ..... | ... | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | Drills and turns easily. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | 1715*b* | 8.535*b* | 0.308 | 112 | 42.07 | 24.65 | 763.49 | For condenser tubes. Resists action of sea water. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .. . | ...... | ...... | ....... | |
| 37,000 | 33,000 | ... | ... | ... | ..... | ... | 180 | ....... | ...... | ...... | 119 | ...... | ...... | ....... | ...................... |
| 45,000 | 33,000 | 100 | 89 | 75 | ..... | ... | ... | 1625*b* | 8.404*b* | 0.304 | 117 | 41.00 | 24.93 | 809.94 | For piston rods, propeller shafts, nuts, bolts, plates, etc. Welding rod. |
| ....... | 33,000 | 165 | 90 | 93 | 55 | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ....... | ...... | ... | ... | ... | ..... | ... | 180 | ....... | 8.712*d* | 0.315 | .... | 32.4 | 32.20 | 989.92 | For Fourdrinier wire. |
| ....... | ...... | ... | ... | 71 | ..... | ... | 180 | 1967*z* | 8.89*h* | 0.321 | .... | 24.1 | 43.0 | 1509.56 | For flexible metal hose. |
| ....... | ...... | ... | ... | ... | ..... | ... | 180 | 1958*z* | 8.89*b* | 0.321 | .... | 29.62 | 35.0 | 1016.05 | For electrical purposes. |
| 54,000 | 33,000 | 175 | 60 | 90 | 30 | ... | 180 | 1922*z* | 8.88*b* | 0.321 | 106 | 82.18‡ | 12.62‡ | 435.45 | For springs, electric switches, etc. |
| ....... | ...... | 190 | 60 | 96 | 30 | ... | ... | 1922*z* | 8.87*b* | 0.320 | 99 | 56.46 | 18.37 | 566.09 | For window weight chain. Bronze chain in general. |
| ....... | ...... | ... | ... | ... | 75 | ... | 180 | ....... | 8.929*b* | 0.322 | .... | 56.46 | 18.37 | 577.70 | Phosphor bronze with good machining properties. |
| 60,000 | ...... | 200 | 70 | 99 | 38 | ... | 180 | 1877*z* | 8.815*b* | 0.318 | 101 | 79.8 | 13.0 | 435.45 | For electric switches, contact fingers, diaphragms, radio parts, etc. |
| 64,000 | ...... | 200 | 74 | 100 | 52 | ... | 180 | 1832*z* | 8.78*b* | 0.317 | 102 | 98.10 | 10.6 | 351.26 | For very stiff resilient springs (flat or coiled). Also for chains. |
| 35,000 | ...... | 120 | ... | 75 | ..... | ... | ... | ....... | 8.86*b* | 0.320 | .... | 85 | 12.2 | 386.10 | For good machining properties. |
| ....... | ...... | ... | ... | ... | ..... | ... | 180 | 1872*b* | ...... | ...... | .... | 86.43 | 12.0 | ....... | For electrical purposes. |

*k* At 18.1 deg. C.

*l* Yield point taken as the load producing an extension under stress of 0.50%. Yield point up to 1934 was taken as the load producing an extension under stress of 0.75%. At that time, however, the A.S.T.M. and Federal Specifications Board adopted the figure of 0.50%.

*m* Cold worked and heat-treated.

*n* Guertler-Tammann constitution diagram.

*p* Annealed, quenched and heat-treated.

*r* Smith constitution diagram.

*t* Stockdale constitution diagram.

*u* B.t.u. per sq. ft. per hr. per 1 in. thick per 1° F., at 68° F.

*v* Tafel constitution diagram.

*x* Bauer and Hansen constitution diagram.

*y* Hard at 77° F.

*z* Heycock-Neville constitution diagram.

(*Table continued on following page*)

**Table 4.—Chemical and Physical Properties of**

(Variations must be

| | Material | Form§ | Approximate Composition, percent | | | | | | Tensile Strength, lb. per sq. in. | | Elongation in 2 in., percent | | Yield Point, lb. per sq. in. | | Johnson's Elastic Limit, lb. per sq. in. | | Modulus of Elasticity, lb. per sq. in.×10⁻⁶, Hard* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Copper | Zinc | Nickel | | | | Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | |
| B39 | Super-nickel | T | 70.00 | ..... | 30.00 | .... | .... | .... | ....... | 65,000 | .... | 30 | ....... | ....... | ....... | ....... | ..... |
| B40 | 20% cupro-nickel | S | 80.00 | ..... | 20.00 | .... | .... | .... | 85,000 | 50,000 | 2 | 30 | ....... | ....... | ....... | ....... | ..... |
| B41 | 15% cupro-nickel | S | 85.00 | ..... | 15.00 | .... | .... | .... | 70,000 | 45,000 | 3 | 30 | ....... | ....... | ....... | ....... | ..... |
| B42 | 30% nickel silver | S | 47.00 | 23 00 | 30.00 | .... | .... | .... | 130,000 | 72,000 | 2 | 35 | ....... | ....... | ....... | ....... | ..... |
| | | W | 47.00 | 23.00 | 30.00 | .... | .... | .... | 160,000 | 75,000 | 1*e* | 35*e* | ....... | ....... | ....... | ....... | ..... |
| B43 | 25% nickel silver | S | 55.00 | 20.00 | 25.00 | .... | .... | .... | 110,000 | 72,000 | 4 | 30 | ....... | ....... | ....... | ....... | ..... |
| B44 | Ambrac A | S | 75.00 | 5.00 | 20.00 | .... | .... | .... | 85,000 | 50,000 | 5 | 35 | 67,000*l* | 21,000*l* | 57,000 | 10,000 | ..... |
| | | R | 75.00 | 5.00 | 20.00 | .... | .... | .... | 80,000 | 55,000 | 10 | 50 | 60,000*l* | 17,000*l* | 52,000 | 13,000 | 19.0‡ |
| | | W | 75.00 | 5.00 | 20.00 | .... | .... | .... | 115,000 | 55,000 | 2*e* | 30*e* | ....... | ....... | ....... | ....... | ..... |
| B45 | 18% nickel silver | S | 65.00 | 17.00 | 18.00 | .... | .... | .... | 90,000 | 58,000 | 3 | 40 | 73,000*l* | ....... | 72,000 | ....... | 18.0 |
| B46 | 18% nickel silver | S | 55.00 | 27.00 | 18.00 | .... | .... | .... | 100,000 | 60,000 | 2 | 40 | ....... | ....... | ....... | ....... | ..... |
| | | W | 56 00 | 26 00 | 18.00 | .... | .... | .... | 143,000 | 60,000 | 1*e* | 40*e* | ....... | ....... | 103,000 | ....... | 14.1 |
| B47 | 15% nickel silver | S | 64.00 | 21.00 | 15.00 | .... | .... | .... | 93,000 | 58,000 | 5.5 | 40 | ....... | ....... | 75,000 | 22,000 | ..... |
| B48 | 15% nickel silver | S | 57.00 | 28.00 | 15.00 | .... | .... | .... | 95,000 | 55,000 | 2 | 35 | ....... | ....... | ....... | ....... | ..... |
| B49 | Leaded nickel silver | S | 61.00 | 25.00 | 12.50 | Lead<br>1.50 | .... | .... | 90,000 | ....... | 5 | .... | ....... | ....... | ....... | ....... | ..... |
| B50 | 10% nickel silver | S | 65 00 | 25.00 | 10.00 | .... | .... | .... | 90,000 | 50,000 | 3 | 45 | ....... | 11,000*l* | ....... | ....... | 17.5‡ |
| B51 | 5% nickel silver | W | 63.00 | 32.00 | 5.00 | .... | .... | ... | 135,000 | ....... | 2*e* | .... | ....... | ....... | ....... | ....... | ..... |
| B52 | 5% aluminum bronze | S | 95.00 | ..... | ..... | Al<br>5.00 | .... | .... | 105,000 | 52,000 | 5 | 70 | ....... | ....... | ....... | ....... | ..... |
| B53 | 8% aluminum bronze | S | 92.00 | ..... | ..... | 8.00 | .... | .... | 120,000 | 60,000 | 4 | 60 | 50,000*l* | ....... | ....... | ....... | 15.0 |
| | | R | 92.00 | ..... | ..... | 8 00 | ... | .. | 100,000 | 60,000 | 4 | 60 | ....... | ....... | ....... | ....... | ..... |
| B54 | 8% aluminum bronze with iron | R | 89 50 | ..... | ..... | 8.00 | Iron<br>2.50 | .... | 125,000 | 72,000 | 5 | 50 | 70,000*l* | 25,000*l* | ....... | ....... | ..... |
| B55 | 10% aluminum bronze | R | 90.00 | ..... | ..... | 10.00 | .... | .... | 125,000*m* | 78,000 | 5*m* | 36 | 60,000*l* | 30,000*l* | ....... | ....... | ..... |
| B56 | Avialite | R | 90.00 | ..... | ..... | 9.50 | 0.50 | .... | 88,000 | ....... | 35 | .... | 39,000*l* | ....... | ....... | ....... | ..... |

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* For some alloys the figures are for a temper slightly different from that commonly known as "hard."

† Compared to water at 4° C.

‡ Soft.

§R = Rod. S = Sheet. T = Tube. W = Wire.

*a* Temper not known.

*b* Determination.

*c* Circular 63, U. S. Bureau of Standards.

*d* Scientific Paper 410, U. S. Bureau of Standards.

*e* Elongation of wire, percent in 10 in.

*f* Corning Glass Works.

*g* Yield point taken as the load producing an extension under stress of 0.75 percent.

*h* Jenkins and Hanson constitution diagram.

*j* Average linear coefficient per deg. C. from 25 to 300 deg. C. Tests on rod. Scientific Paper 410, U. S. Bureau of Standards.

**Various Wrought Copper-base Alloys**—(*Continued*)
expected in practice)

| Shearing Strength, lb. per sq. in. |  | Brinell Hardness No. 10-mm. Ball, 500-kg. Load |  | Rockwell Hardness No. B 1/16-in. Ball, 100-kg. |  | Bend Test, deg. |  | Melting point, deg. F. | Specific Gravity † | Density, lb. per cu. in. | Coefficient of Expansion × 10⁷ (*j*) | Electrical Properties at 20° C. |  | Thermal Conductivity (*u*) | Uses—Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft |  |  |  |  | Resistivity, ohms (mil-ft.) | Conductivity, percent I.A.C.S. |  |  |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | 2237*n* | 8.950 | 0.323 | 90*f* | 218.4 | 4.75 | 200.31 | For condenser tubes. |
| ....... | ...... | ... | ... | 85 | 37.5 | ... | 180 | 2192*n* | ...... | ...... | .... | 160.2 | 6.47 | 252.56 | For turbine blades and parts where resistance to corrosion and erosion is required. |
| ....... | ...... | ... | ... | ... | ..... | ... | 180 | 2147*n* | 8.95*b* | 0.323 | .... | 127.0 | 8.17 | 325.14 | For bullet jackets. |
| ....... | ...... | ... | ... | ... | 61 | ... | 180 | 2084*v* | 8.74*b* | 0.316 | .... | 290.0 | 3.58 | ....... | Has comparatively high electrical resistance. Used in electrical instruments. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... |  |
| ....... | ...... | 208 | 89 | ... | 60 | ... | 180 | 2075*v* | 8.72*b* | 0.315 | .... | 259.0*a* | 4.00*a* | ....... | For tableware, plated and unplated. |
| ....... | ...... | 160 | 58 | 88 | 25 | ... | 180 | 2102*b* | 8.860 | 0.320 | 91*f* | 172.0 | 6.2 | 260.08 | For resistance to corrosion and atmospheric action; also for ornamental purposes. |
| 50,000 | 33,000 | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... |  |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... |  |
| ....... | ...... | 170 | 70 | 91 | 40 | ... | 180 | 2030*v* | 8.752*b* | 0.316 | .... | 175.0 | 5.91 | 232.24 | For silver-plated forks, spoons, knives, hollow ware, etc. |
| 65,000 | ...... | 190 | 70 | 95 | 40 | ... | ... | 2111*v* | 8.68*b* | 0.314 | .... | 186.5‡ | 5.56‡ | 206.11 | Similar to 30% nickel silver, but of lower resistance. |
| ....... | ...... | ... | ... | ... | ..... | 180 | 180 | ....... | ...... | ...... | .... | 189 5 | 5 49 | 206.11 |  |
| ....... | ...... | ... | 73 | 92 | 33 | ... | 180 | 1967*v* | 8.691*b* | 0.314 | .... | 165.6 | 6.26 | 235.14 | For silver-plated ware, spinning, drawing, and for work where a low percentage of nickel is required. |
| ....... | ...... | ... | 74 | ... | ..... | ... | 180 | 1886*b* | 8.631*b* | 0.312 | .... | ...... | ...... | ....... | For white metal tubes, sheets, wire, etc. Also in plumbing and decorations. |
| ....... | ...... | 166 | ... | 88 | ..... | ... | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | For watch parts, etc. Free cutting. |
| ....... | 34,000 | ... | ... | 82 | 32 | ... | ... | 1850 | 8.675*b* | 0.313 | .... | 125.5 | 8 27 | 319.33 | For cheaper grades of silver-plated ware. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | 1760*v* | ...... | ...... | .... | 86.5 | 11.99 | 406.42 |  |
| ....... | ...... | 176 | 67 | 93 | 20 | ... | ... | 1940*t* | 8.176*b* | 0.295 | .... | 58.61 | 17.69 | 574.79 | For diaphragms to withstand pressure; also for its color. |
| ....... | ...... | 185 | 60 | 99 | 30 | ... | ... | 1904*t* | 7.80*b* | 0.281 | 99 | 70.08*k* | 14.80*k* | 502.22 | For diaphragms to withstand pressure; also for resistance to ordinary corrosion and wear. |
| ....... | ...... | ... | ... | ... | ..... | ... | ... | ....... | ...... | ...... | .... | ..... | ..... | ....... |  |
| ....... | ...... | 190 | 70 | 100 | 52 | ... | .. | ....... | 7.74*b* | 0.280 | .... | 95.10 | 10.9 | ....... | For strength and resistance to ordinary corrosion and wear. |
| ....... | ...... | 190 | 100 | 100 | 65 | ... | ... | 1904*t* | 7.57*b* | 0.273 | .... | 76.80 | 13.5 | 455.77 | For strength and resistance to ordinary corrosion and wear. |
| 45,000 | ...... | 140 | ... | ... | ..... | ... | ... | 1908*b* | 7.585*b* | 0.274 | 94 | 82.25 | 12.61 | 418.03 | For valve seats in airplane engines and at elevated temperatures. |

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*k* At 18.1 deg. C.

*l* Yield point taken as the load producing an extension under stress of 0.50%. Yield point up to 1934 was taken as the load producing an extension under stress of 0.75%. At that time, however, the A.S.T.M. and Federal Specifications Board adopted the figure of 0.50%.

*m* Cold worked and heat-treated.

*n* Guertler-Tammann constitution diagram.

*p* Annealed, quenched and heat-treated.

*r* Smith constitution diagram.

*t* Stockdale constitution diagram.

*u* B.t.u. per sq. ft. per hr. per 1 in. thick per 1° F., at 68° F.

*v* Tafel constitution diagram.

*x* Bauer and Hanson constitution diagram.

*y* Hard at 77° F.

*z* Heycock-Neville constitution diagram.

(*Table continued on following page*)

**Table 4.—Chemical and Physical Properties of**

(Variations must be

| | Material | Form§ | Approximate Composition, percent | | | | | | Tensile Strength, lb. per sq. in. | | Elongation in 2 in., percent | | Yield Point, lb. per sq. in. | | Johnson's Elastic Limit, lb. per sq. in. | | Modulus of Elasticity, lb. per sq. in.$\times 10^{-6}$, Hard* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Copper | Zinc | Nickel | | | | Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | |
| | | | | | Manganese | Al | Iron | Tin | | | | | | | | | |
| B57 | Calsun bronze | W | 95.50 | ..... | ..... | 2.50 | .... | 2.00 | 135,000 | 50,000 | 4*e* | 35*e* | ....... | ....... | 31,000 | ...... | ..... |
| B58 | Manganese bronze | R | 57.00 | 40.00 | 0.10 | .... | 1.45 | 1.45 | 90,000 | 65,000 | 15 | 45 | ....... | ....... | ....... | ...... | ..... |
| B59 | Manganese bronze | R | 59.00 | 39.00 | 0.50 | .... | 0.80 | 0.70 | 85,000 | 60,000 | 20 | 45 | ....... | ....... | ....... | ...... | ..... |
| | | | | | | Si | | | | | | | | | | | |
| B60 | Everdur (A) 1010 | S | 96.00 | ..... | 1 00 | 3 00 | .... | .... | 113,000 | 55,000 | 5 | 48 | 65,000*l* | 20,000*l* | 72,000 | 9,300 | ..... |
| | | R | 96.00 | ..... | 1.00 | 3 00 | .... | .... | 95,000 | 55,000 | 15 | 85 | 65,000*l* | 20,000*l* | 67,000 | ...... | 15.0 |
| | | W | 96.00 | ..... | 1.00 | 3.00 | .... | .... | 145,000 | 59,000 | 5*e* | 50*e* | 75,000*l* | 20,000*l* | 90,000 | ...... | ..... |
| B61 | Everdur (B) 1015 | T | 98.25 | ..... | 0.25 | 1.50 | .... | .... | 65,000 | 40,000 | 15 | 60 | 48,000*l* | 10,000*l* | ....... | ...... | ..... |
| | | R | 98.25 | ..... | 0.25 | 1.50 | .... | .... | 70,000 | 40,000 | 6 | 60 | ....... | 10,000*l* | ....... | ...... | ..... |
| | | S | 98.25 | ..... | 0.25 | 1.50 | .... | .... | 70,000 | 40,000 | 6 | 46 | 53,000*l* | 10,000*l* | 38,000 | ...... | ..... |
| | | | | | | Cadmium | | | | | | | | | | | |
| B62 | Hitenso A | W | 99.35 | ..... | ..... | 0.65 | .... | .... | 75,000 | ....... | 3*e* | .... | 40,000*l* | ....... | 45,000 | ...... | 15.6 |
| | | S | 99 35 | ..... | .... | 0 65 | .... | .... | 54,000 | ....... | 5 | .... | ....... | ....... | ....... | ...... | ..... |
| B63 | Hitenso BB | S | 99.00 | ..... | ..... | 1.00 | .... | .... | 60,000 | 35,000 | 3 | 50 | ....... | ....... | 36,000 | ...... | ..... |
| | | W | 99.00 | ..... | ..... | 1.00 | .... | .... | 92,000 | 35,000 | 3*e* | 50*e* | ....... | ....... | 55,000 | ...... | ..... |
| B64 | Hitenso C | S | 98.60 | ..... | ..... | 0.80 | .... | 0.60 | ....... | 36,000 | .... | 50 | ....... | 14,000*l* | ....... | ...... | ..... |
| | | W | 98 60 | ..... | ..... | 0 80 | .... | 0.60 | 99,000 | 40,000 | 4*e* | 45*e* | ....... | ....... | 59,000 | ...... | ..... |
| | | | | Iron | Ni | Al | Mn | | | | | | | | | | |
| B65 | Tempaloy 917 | R | 81.90 | 2.50 | 5.00 | 9.60 | 1.00 | .... | 100,000 | ....... | 10 | .... | 50,000*g* | ....... | ....... | ...... | ..... |
| | | | | Zinc | Lead | Tin | Iron | | | | | | | | | | |
| B66 | Extruded architectural bronze shapes | .. | 57.00 | 40 00 | 2 50 | 0.34 | 0.16 | .... | 70,000 | 50,000 | 10 | 20 | ....... | ....... | ....... | ...... | ..... |
| | | | | | Ni | Be | | | | | | | | | | | |
| B67 | Beryllium copper | S | 97.40 | ..... | 0.35 | 2.25 | .... | .... | 118,000 | 70,000 | 4.3 | 45 | 90,000*l* | 27,000*l* | 79,000 | 18,000 | 17.2 |
| | | S | 97.40 | ..... | 0.35 | 2.25 | .... | .... | 193,000<br>*m* | 175,000<br>*p* | 2<br>*m* | 6.3<br>*p* | 97,000<br>*lm* | 95,000<br>*pl* | 130,000<br>*m* | 87,000<br>*p* | 18.4<br>*m* |

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* For some alloys the figures are for a temper slightly different from that commonly known as "hard."

† Compared to wate: at 4° C.

‡ Soft.

§ R = Rod. S = Sheet. T = Tube. W = Wire.

*a* Temper not known.

*b* Determination.

*c* Circular 63, U. S. Bureau of Standards.

*d* Scientific Paper 410, U. S. Bureau of Standards.

*e* Elongation of wire, percent in 10 in.

*f* Corning Glass Works.

*g* Yield point taken as the load producing an extension under stress of 0.75 percent.

*h* Jenkins and Hanson constitution diagram.

*j* Average linear coefficient per deg. C. from 25 to 300 deg. C. Tests on rod. Scientific Paper 410, U. S. Bureau of Standards.

**Various Wrought Copper-base Alloys**—(*Continued*)
expected in practice)

| Shearing Strength, lb. per sq. in. | | Brinell Hardness No. 10-mm. Ball, 500-kg. Load | | Rockwell Hardness No. B 1/16-in. Ball, 100-kg. | | Bend Test, deg. | | Melting point, deg. F. | Specific Gravity† | Density, lb. per cu. in. | Coefficient of Expansion × 10⁷ (*j*) | Electrical Properties at 20° C. | | Thermal Conductivity (*u*) | Uses—Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Hard* | Soft | Hard* | Soft | Hard* | Soft | Hard* | Soft | | | | | Resistivity, ohms (mil-ft.) | Conductivity, percent I.A.C.S. | | |
| ...... | ...... | ... | ... | ... | ...... | ... | ... | 1929*b* | 8.540*b* | 0.308 | .... | 61.0 | 17.0 | ....... | For electrical uses where corrosion resistance is important. |
| ...... | ...... | 170 | ... | ... | ...... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | For structural work, because of strength and resistance to corrosion. |
| ...... | ...... | ... | ... | 90 | ...... | ... | ... | ....... | 8.370*b* | 0.302 | .... | 42.15 | 24.6 | 699.62 | For structural work because of strength and resistance to corrosion. |
| ...... | ...... | 200 | 70 | 95 | 40 | ... | 180 | 1866*b* | 8.539*b* | 0.308 | 100 | 155. | 6.7 | 226.43 | For strength and resistance to corrosion. Has strength of mild steel and corrosion resistance of copper. Welding rod. |
| 56,000 | 33,000 | 190 | 60 | ... | ...... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| 75,000 | 35,000 | ... | ... | ... | ...... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ...... | ...... | ... | ... | 75 | 20 | ... | 180 | 2111*r* | 8.740*b* | 0.316 | .... | ...... | ...... | ....... | For strength and resistance to corrosion; bolt stock, and sheet metal requiring high ductility. |
| ...... | ...... | ... | ... | ... | ...... | ... | 180 | ....... | ...... | ...... | .... | 86.4 | 12.0 | 374.49 | |
| ...... | ...... | ... | ... | 80 | 3 | ... | 180 | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ...... | ...... | ... | ... | ... | ...... | ... | 180 | 1976*h* | 8.89*b* | 0.3212 | .... | 12.20 | 85.0 | ....... | For electrical wire and cable, etc. |
| ...... | ...... | 95 | ... | 62 | ...... | ... | ... | ....... | ...... | ...... | .... | ...... | ...... | ....... | |
| ...... | ...... | 98 | ... | 65 | ...... | ... | 180 | 1969*h* | 8.89*b* | 0.3212 | .... | 12.95*y* | 80.0*y* | 2392.07 | For electrical wire and cables, contact fingers, commutator segments, etc. |
| ...... | ...... | ... | ... | ... | ...... | ... | ... | ....... | ...... | ...... | .... | 12.95*y* | 80.0*y* | ....... | |
| ...... | ...... | ... | ... | ... | ...... | ... | 180 | 1958*b* | 8.89*b* | 0.3212 | .... | 18.85*y* | 55.0*y* | 1614.07 | For electrical wire and cables, etc. |
| ...... | ...... | ... | ... | ... | ...... | ... | ... | ....... | ...... | ...... | .... | 18.85*y* | 55.0*y* | ....... | |
| ...... | ...... | ... | ... | ... | ...... | ... | ... | 1929*b* | 7.569 | 0.273 | .... | ...... | ...... | ....... | For strength and resistance to corrosion. |
| ...... | ...... | ... | ... | ... | ...... | ... | ... | 1823*b* | 8.432*b* | 0.305 | .... | ...... | ...... | ....... | For architectural shapes. |
| ...... | ...... | ... | ... | 102 | 65–73 | ... | ... | 1751*b* | 8.23 ±.02 | 0.297 ±.01 | 94 | 58.48 | 20.8*p* | 725.7*p* | For springs, diaphragms, low duty bushings and bearings and Bourdon tubes. |
| ...... | ...... | ... | ... | 114 *m* | 112.5 *p* | ... | ... | ....... | 8.23 ±.02 | 0.297 ±.01 | .... | 59.8 | 17.3 *m* | 580.60 *m* | High resistance to fatigue. |

*k* At 18.1 deg. C.

*l* Yield point taken as the load producing an extension under stress of 0.50%. Yield point up to 1934 was taken as the load producing an extension under stress of 0.75%. At that time, however, the A.S.T.M. and Federal Specifications Board adopted the figure of 0.50%.

*m* Cold worked and heat-treated.

*n* Guertler-Tammann constitution diagram.

*p* Annealed, quenched and heat-treated.

*r* Smith constitution diagram.

*t* Stockdale constitution diagram.

*u* B.t.u. per sq. ft. per hr. per 1 in. thick per 1° F., at 68° F.

*v* Tafel constitution diagram.

*x* Bauer and Hansen constitution diagram.

*y* Hard at 77° F.

*z* Heycock-Neville constitution diagram.

beneficial effects, such as good casting characteristics. Therefore, it is either the major or one of the minor alloying constituents of many of the most corrosion-resistant alloys. Iron and copper lower the corrosion resistance, but properly heat-treated aluminum-copper alloys have sufficient corrosion resistance to be used for many applications without paint protection.

The thickness of an aluminum part has an important bearing on its ability to resist corrosion. Under corrosive conditions the rate of attack decreases with time, since the process of corrosion builds up a resistant film or coating which inhibits further attack. This action consumes some of the aluminum, and may result in complete perforation of thin sheet or foil, but in heavier sections the same amount of attack will have no appreciable effect on the mechanical integrity of the part.

In aircraft, the strong aluminum alloys are protected from corrosion by one of the several commercially available methods. Wrought Alclad products, consisting of a heat-treatable strong alloy core covered by a surface layer of high purity aluminum, or of corrosion-resistant alloy, combines mechanical strength and corrosion resistance. The surface layer covers the alloy core and so protects it from corrosion, and also, because of its higher solution potential, offers an electrolytic protection over cut edges and abraded areas, and prevents penetration into the core, even under severely corrosive conditions. Other non-heat-treatable Alclad products, employing the principle of a coating of higher solution potential than the core have high available resistance to destructive perforation.

Because of the high solution potential of aluminum and its alloys, contact with many of the commonly-used metals in the presence of a corrosive medium results in rapid deterioration of the aluminum, with a corresponding protection of the metal of lower solution potential. Hence, metallic contacts with other metals generally should be avoided. Electrical insulation between the two dissimilar metals is of value, although salts formed by the solution of the other metal may be plated out on the aluminum surface and set up electrolytic action. In spite of the rapid action in severely corrosive media, little corrosion occurs in ordinary atmosphere, unless a solution of relatively low electrical resistance accumulates. On the other hand, in some assemblies used in contact with strong electrolytes, it is even advisable to consider the relative solution potentials of various aluminum alloys which are in contact.

Exposure to the weather tends to darken and slightly roughen bright surfaces produced by rolling or extrusion, or highly polished surfaces. A bright, highly reflective surface may be maintained by periodic cleaning. Pitting of cooking utensils and similar articles, often is caused by plating out of particles of heavy metals which have been dissolved in the liquids being handled. This pitting can be avoided by regular cleaning.

**Table 1.—Properties of Aluminum**

| | |
|---|---|
| Atomic weight | 26.97 |
| Density at 68° F. | 168.56 lb. per cu. ft. |
| “ at melting point (solid) | 159.20 “ “ “ “ |
| “ “ “ “ (liquid) | 148.59 “ “ “ “ |
| Melting point | 1220° F. |
| Boiling point | 3272° F. |
| Coefficient of expansion per deg. F., 68–212° F. | 0.000 0132 |
| “ “ “ “ “ 68–572° F. | 0.000 0143 |
| “ “ “ “ “ 68–932° F. | 0.000 0154 |
| Heat of vaporization | 3510–3600 B.t.u. per lb. |
| Latent heat of fusion | 167.36 “ “ “ |
| Specific heat, 32–212° F. | 0.226 |
| Thermal conductivity, B.t.u. per hr. per sq. ft. per deg. F. per 1 in. thickness | 1509 |
| ELECTRICAL PROPERTIES | |
| Electrical resistivity at 20° C. | 2.688 microhms per cu. cm. |
| “ “ “ “ (commercial conductors) | 2.83 “ “ “ |
| Temperature coefficient of resistivity at 20° C. | 0.00403 |
| MECHANICAL PROPERTIES | |
| Tensile strength | 9000 lb. per sq. in. |
| Yield strength (set. = 0.2%) | 3000 “ “ “ “ |
| Elongation in 2 in., percent | 60 |
| Brinell hardness, 10-mm. ball, 500 kg. load | 15 |
| Modulus of elasticity | 10,000,000 |
| Modulus of rigidity (torsion) | 3,870,000 |
| Poisson's ratio | 0.33 |
| MISCELLANEOUS PROPERTIES | |
| Crystallization shrinkage | 6.6 percent |
| Reflectivity for white light | 87 percent |
| Magnetic susceptibility (c. g. s. units) | 0.000 000 58 |
| Compressibility (change in volume per megabar pressure) | 0.000 0014 |

**COMMERCIAL FORMS.**—Pure aluminum is soft and ductile but can be hardened by cold working and alloying; certain of the alloys also can be hardened by heat treatment. Both pure aluminum and its alloys are available in wrought form, as plate, sheet, rod, wire, tubing, and rolled and extruded sections. It also can be worked by forging, drawing, stamping, spinning, and other metal-working methods. Alloys are available in cast form, produced by sand, permanent mold, or die-casting processes.

## 2. CAST ALUMINUM ALLOYS

Pure aluminum does not cast easily, nor does such a casting have attractive commercial properties. The alloys of aluminum used for casting depend on the process employed, and are designated as follows: Sand cast alloys (Table 2); permanent mold casting alloys (Table 3); pressure die-casting alloys (Table 4).

**SAND CASTING.**—Large castings such as crank-cases for internal-combustion engines, oil pumps, manifolds, transmission housings, motor frames, architectural castings, etc., usually are made as sand castings. The ordinary methods of molding with green sand molds (see Foundry Practice, pp. 20–30 to 20–36) are employed, and good melting and patterns are necessary for the best results. Cores may be either green or baked sand, with binders of oil for hard cores, and pitch or resin for soft cores. Ample gates and risers are necessary. Cracks and shrinkage are avoided by the use of iron, aluminum or copper cores. Sand for molds must be of proper temper, and uniformly rammed. A hard-rammed mold is detrimental, as aluminum will not "lay" well against it.

**Melting.**—Iron pots, non-metallic crucibles or reverberatory furnaces generally are used for melting. The temperature preferably should be kept below 1500° F., and an oxidizing atmosphere is preferred to a reducing one. The metal must be stirred thoroughly to insure uniform composition. To obtain the strongest, most ductile and leak-proof castings, pouring temperatures should be the lowest at which the mold can be filled without misruns. Often the proper arrangement of gates and risers will facilitate the use of low pouring temperatures.

Hard spots which cause difficulty in machining, consist of foreign matter entrapped in the liquid alloys. They usually are the result of unclean melting practice.

**Maximum and Minimum Weights** range from 7000 lb. to as low as 1 oz. There is no inherent reason for weight limitations of sand castings.

**Section Thickness.**—Minimum section thickness of sand castings is about $^1/_8$ in. There is no maximum limit.

**Finish Allowance** on large sand castings ranges from $^3/_{16}$ to $^3/_4$ in., depending on design. For small bench-work castings, the finish allowance should be $^1/_8$ in. For disc grinding it should be $^1/_{32}$ in.

**Tolerances.**—Thickness tolerance on sand castings should be $^1/_{32}$ in. The weight tolerance with wood patterns should be $\pm 5\%$ and $\pm 3\%$ with metal patterns.

**PERMANENT MOLD CASTINGS** are poured in metal molds. Such castings are used for cooking utensils, vacuum sweeper and washing machine parts, cylinder heads, automobile pistons (made from the 122 or 132 alloy), etc. They may be made very close to size, and machine easily. As a result of the rapid chilling of the metal in the mold, the casting combines high hardness with toughness. Semi-permanent mold castings are made by using sand cores in conjunction with metal mold parts. This process gives great latitude in design, and at the same time gives the advantages of the permanent mold process.

**Maximum and Minimum Weights** of permanent mold castings range from 750 lb. to less than $^1/_2$ oz. The usual range, however, is from 20 lb. to $^1/_2$ lb.

**Section Thickness** should not be less than $^3/_{32}$ in., with no high limit.

**Finish Allowance** for permanent mold piston castings is as follows: Up to 3 in. diam., 0.075 in. on the diameter; up to 5 in. diam., 0.10 in.

**Tolerance.**—A dimensional tolerance of $\pm 0.010$ in. should be allowed. The weight tolerance with metal patterns should be $\pm 2\%$.

**PRESSURE DIE-CASTINGS** are made more nearly to size than either sand or permanent mold castings, with a consequent elimination of all or many machining operations. The die is filled quickly with metal under high pressure, and the casting solidifies so rapidly that crystallization shrinkage cannot be completely compensated, nor all of the entrapped air eliminated. The interior of the casting may be porous, but the outside shell usually is sound and strong. The average tensile strength of pressure die-castings is high, notwithstanding their porosity. Aluminum die castings may be made with thinner sections than castings made by other processes.

**Maximum and Minimum Weights.**—A pressure die-casting weighing 12 lb. is considered large. Castings as small as $^1/_2$ oz. are common.

### Table 2.—Composition and Properties of Sand-cast Aluminum Alloys *

| Alloy † | Nominal Chemical Composition | Tension,‡ Typical Values | | | Compression,¶ Typical Values | | Hardness, Typical Values | | Shear, Typical Values | Fatigue | Density |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Yield Strength,§ lb. per sq. in. | Ultimate Strength, lb. per sq. in. | Elongation, % in. 2 in. | Yield Strength,§ lb. per sq. in. | Ultimate Strength, lb. per sq. in. | Brinell 10-mm. ball 500-kg. load | Rockwell 1/8-in. ball 100-kg. load | Shearing Strength,‖ lb. per sq. in. | Endurance Limit,** lb. per sq. in. | Typical Values, lb. per cu. in. |
| 12 and 212, ASTM–B, SAE–30 | Cu 8% | 14,000 | 22,000 | 2.0 | 16,000 | 38,000 | 65 | E–75 | 20,000 | 7,500 | 0.102 |
| 43, ASTM–J, SAE–35 | Si 5% | 9,000 | 19,000 | 4.0 | 9,000 | 25,000 | 40 | E–41 | 15,000 | 6,500 | 0.096 |
| 47, ASTM–K, SAE–37 | Si 12.5% | 11,000 | 26,000 | 8.0 | 12,000 | 28,000 | 50 | E–59 | 18,000 | 6,000 | 0.095 |
| 108 | Cu 4%, Si 3% | 17,000 | 21,000 | 2.0 | 21,500 | 43,000 | 60 | E–70 | 20,000 | 8,500 | 0.099 |
| 109, ASTM–E, SAE–32 | Cu 12% | 18,000 | 24,000 | 1.5 | 23,500 | 45,000 | 75 | E–82 | 20,000 | 10,000 | 0.105 |
| 112, ASTM–C, SAE–33 | Cu 7%, Zn 2%, Fe 1.2% | 14,000 | 22,000 | 2.0 | 24,000 | 44,000 | 70 | E–79 | 20,000 | 8,500 | 0.103 |
| 122–T2 ASTM–F, SAE–34 | Cu 10%, Fe 1.2%, Mg 0.2% | 21,000 | 25,000 | 1.0 | 33,500 | 54,000 | 75 | E–82 | 25,500 | 9,500 | 0.106 |
| 122–T61 | | 30,000 | 36,000 | 1.0 | 40,500 | 80,000 | 100 | E–96 | 29,500 | ...... | 0.106 |
| 142 ASTM–H, SAE–39 | Cu 4%, Ni 2%, Mg 1.5% | 24,000 | 28,000 | 1.0 | 34,000 | 52,000 | 85 | E–88 | 24,000 | 8,000 | 0.101 |
| 142–T61 | | 30,000 | 37,000 | 0.5 | 58,000 | 70,000 | 100 | E–96 | 32,000 | 8,000 | 0.101 |
| 195–T4 ASTM–G, SAE–38 | Cu 4.5% | 16,000 | 31,000 | 8.0 | 27,000 | 43,000 | 65 | E–70 | 28,000 | 6,000 | 0.100 |
| 195–T6 | | 22,000 | 36,000 | 4.0 | 29,000 | 48,000 | 80 | E–82 | 30,000 | 6,500 | 0.100 |
| 195–T62 | | 27,000 | 40,000 | 2.0 | 45,500 | 56,000 | 95 | E–92 | 31,000 | 7,000 | 0.100 |
| 214 | Mg 3.7% | 12,000 | 25,000 | 9.0 | 12,000 | 50,000 | 50 | E–59 | 19,000 | 5,500 | 0.095 |
| 216 | Mg 6% | 16,000 | 27,000 | 6.0 | 15,000 | 57,000 | 60 | E–70 | 23,500 | ...... | 0.094 |
| 220–T4 | Mg 10% | 26,000 | 44,000 | 13.0 | 23,500 | 72,500 | 75 | E–82 | 33,500 | 7,500 | 0.092 |
| A–334 | Cu 3%, Si 4%, Mg 0.3% | 20,000 | 26,000 | 1.5 | 20,000 | 70,000 | 65 | E–79 | 24,000 | ...... | 0.099 |
| 355–T4 | Cu 1.2%, Si 5%, Mg 0.5% | 20,000 | 30,000 | 4.0 | 25,000 | 65,000 | 60 | E–70 | 30,000 | ...... | 0.097 |
| 355–T6 | | 27,000 | 35,000 | 2.0 | 29,000 | 68,000 | 80 | E–86 | 30,000 | ...... | 0.097 |
| A–355–T51 | Cu 1.4%, Si 5%, Mn 0.7% Ni 0.7%, Mg 0.5% | 24,000 | 28,000 | 1.5 | 24,000 | 54,000 | 65 | E–70 | 21,000 | 8,000 | 0.099 |
| A–355–T59 | | 21,000 | 25,000 | 2.0 | 21,000 | 52,000 | 60 | E–66 | 20,000 | 8,000 | 0.099 |
| 356–T4 | Si 7%, Mg 0.3% | 16,000 | 28,000 | 6.0 | 16,000 | 46,000 | 55 | E–70 | 22,000 | ...... | 0.095 |
| 356–T6 | | 22,000 | 32,000 | 4.0 | 21,000 | 48,000 | 70 | E–82 | 23,000 | 8,000 | 0.095 |
| 645 | Zn 11%, Cu 2.5%, Fe 1.5% | 22,000 | 29,000 | 4.0 | 34,000 | 50,000 | 70 | E–82 | 22,500 | 7,500 | 0.106 |

* Modulus of elasticity approximately 10,300,000 lb. per sq. in. for these alloys.

† Alloy numbers, except A.S.T.M. and S.A.E., refer to alloys of Aluminum Co. of America. "T" suffix indicates heat-treated alloys.

‡ Tension values determined from standard 1/2-in. diameter tensile test specimens cast in green sand molds and tested without machining off the surface.

§ Stress at which stress-deformation curve departs 0.2 percent from the initial modulus line produced.

¶ Results of tests on specimens have an $l/r$ ratio of 16–20. All specimens failed by lateral bending.

‖ Single-shear strength values obtained from double-shear tests.

** Endurance limit based on 500,000,000 cycles, using R. R. Moore type of rotating-beam machine.

**Section Thickness.**—The minimum section thickness for pressure die-castings is about 1/8 in., with no maximum limit.

**Finish Allowance** is not required if tolerances of ±0.0025 in. are permitted.

**Tolerances.**—A dimensional tolerance of ±0.0025 in. should be allowed. The weight tolerance should be ±2%.

**ALUMINUM ALLOY CASTINGS AT HIGH TEMPERATURES.**—Strength and hardness of commercial aluminum decreases progressively with rising temperature. Certain alloys, however, are sufficiently strong and hard to be used commercially at high temperatures. Examples are pistons and cylinder heads of internal-combustion engines. See Table 5 for properties of such alloys.

One alloy is stronger at 600° F. than commercial aluminum at ordinary temperature. The values shown, however, for 400, 500 and 600° F. are much lower than those usually

**Table 3.—Composition and Properties of Permanent Mold Aluminum Casting Alloys**

| Alloy* | Approximate Composition | Typical Values: Tensile Strength,† lb. per sq. in. | Typical Values: Elongation, % in 2 in. | Brinell Hardness 500-kg. 10-mm. ball ‡ | Approximate Density, lb. per cu. in. |
|---|---|---|---|---|---|
| 43 | Si 5% | 24,000 | 4.0 | 45– 55 | 0.097 |
| A–108 | Cu 4.5%, Si 5.5% | 26,000 | 1.5 | 65– 80 | 0.099 |
| 112 | Cu 7%, Zn 2%, Fe 1.2% | 27,000 | 2.0 | 70– 90 | 0.106 |
| B–113 | Cu 7%, Si 1.5%, Fe 1.2% | 29,000 | 1.0 | 70– 90 | 0.104 |
| 122 | Cu 10%, Fe 1.2%, Mg 0.25% | 32,000 | 0.4 | 85–110 | 0.106 |
| 122–T 52 | | 32,000 | 0.5 | 95–125 | 0.106 |
| 122–T551 | | 35,000 | .......... | 125–150 | 0.106 |
| 122–T552 | | 33,000 | .......... | 100–125 | 0.106 |
| 122–T 62 | | 42,000 | .......... | 125–150 | 0.106 |
| 132 | Si 14%, Cu 1%, Fe 1%, Mg 1%, Ni 2% | 32,000 | 0.5 | 85–110 | 0.098 |
| 132–T541 | | 34,000 | .......... | 90–115 | 0.098 |
| 132–T 7 | | 36,000 | 0.5 | 95–120 | 0.098 |
| 142 | Cu 4%, Ni 2%, Mg 1.5% | 35,000 | 0.5 | 90–115 | 0.100 |
| 142–T571 | | 38,000 | .......... | 90–120 | 0.100 |
| 142–T 61 | | 44,000 | .......... | 100–130 | 0.100 |
| 144 | Cu 10%, Si 4%, Mg 0.25% | 32,000 | .......... | 85–110 | 0.104 |
| 144–T4 | | 40,000 | 0.5 | 100–130 | 0.104 |
| B–195–T4 | Cu 4.5%, Si 3% | 38,000 | 6.0 | 70– 90 | 0.100 |
| D–195–T4 | Cu 5.5% | 40,000 | 8.0 | 60– 80 | 0.102 |
| A–214 | Mg 3.75%, Zn 2% | 24,000 | 4.5 | 50– 65 | 0.096 |
| 355–T4 | Si 5%, Cu 1.2%, Mg 0.5% | 35,000 | 5.0 | 70– 85 | 0.097 |
| 355–T6 | | 43,000 | 4.0 | 85–100 | 0.097 |

* Alloy numbers referred to are those of the Aluminum Co. of America. Prefixed letters denote changes in composition of standard alloy. Suffixed "T" and number designate heat-treated alloys.

† Properties obtained from standard 1/2-in. diameter test specimens, individually cast in permanent molds and tested without machining off surface.

‡ Hardness limits are based on casting Brinell tests. In the case of 122 and 142 alloys, the Brinell is that obtained on the heads of permanent mold pistons 1/2 in. or less in thickness, and at a point 1/2 in. from outer edge and 1/2 in. from gate. For heavier castings, semi-permanent and sand-cast pistons, the hardness decreases proportionately with increased section, the maximum decrease being approximately 20.

**Table 4.—Composition and Properties of Aluminum-base Die-casting Alloys**

| Alloy Numbers* | Desired Composition, percent: Copper | Silicon | Nickel | Aluminum | Typical Values: Tensile Strength, lb. per sq. in. | Elongation, % in 2 in. |
|---|---|---|---|---|---|---|
| IV | ........ | 5 | ........ | 95 | 29,000 | 3.5 |
| V | ........ | 12 | ........ | 88 | 33,000 | 1.5 |
| VI | 2 | 3 | ........ | 95 | 30,000 | 3.5 |
| VII | 4 | 5 | ........ | 91 | 32,000 | 2.0 |
| VIII | 1.5 | 1 | 2.25 | 95.25 | 29,000 | 4.0 |
| IX | 4 | 1.75 | 4 | 90.25 | 31,000 | 1.5 |
| XII | 8 | 1.5 | ........ | 90.50 | 33,000 | 1.0 |

* The alloy designations correspond to numbers used in the investigation on aluminum-base die-casting alloys carried out under the jurisdiction of Committee B-6 on Die Cast Metals and Alloys of the American Society for Testing Materials; the alloy numbers omitted correspond to alloy compositions which were investigated by the Committee, but which were not considered desirable for inclusion in these specifications. See *Proc.* A.S.T.M., vol. xxix, Part I, p. 192, 1929.

**Table 5.—Properties of Cast Aluminum Alloys at Elevated Temperatures after Stabilization at the Test Temperature**

| Alloy | | Property | Temperature, deg. F. | | | | |
|---|---|---|---|---|---|---|---|
| Designation from Tables 2 and 3 | Condition | | 75 | 300 | 400 | 500 | 600 |
| 112<br>ASTM–C<br>SAE–33 | Sand cast | T.S.* | 23,400 | 25,500 | 17,150 | 14,860 | 6,300 |
| | | Y.S.† | 14,800 | 22,750 | ......... | 9,700 | ......... |
| | | Elong.‡ | 1.8 | 1.5 | 1.5 | 3.5 | 19.0 |
| | | B.H.§ | 74 | 91 | 72 | 65 | 49 |
| 122–T551<br>SAE–34 | Permanent mold cast | T.S. | 35,000 | 34,910 | 23,300 | 18,770 | 9,070 |
| | | Y.S. | 34,000 | 33,500 | ......... | 10,750 | ......... |
| | | Elong. | 0.0 | 0.5 | 8.5 | 2.0 | 10.0 |
| | | B.H. | 130 | 112 | 87 | 74 | 54 |
| 132–T541 | Permanent mold cast | T.S. | 34,000 | 17,500 | 17,940 | 18,970‖ | 9,610 |
| | | Y.S. | 24,000 | 8,000 | ......... | 8,660‖ | ......... |
| | | Elong. | 0.5 | 2.0 | 1.0 | 2.0‖ | 3.0 |
| | | B.H. | 105 | ......... | 76 | 75‖ | 61 |
| 142–T571<br>ASTM–H<br>SAE–39<br>"Y" | Permanent mold cast | T.S. | 38,000 | 40,930 | 28,100 | 15,520 | 9,080 |
| | | Y.S. | 31,000 | 38,000 | ......... | 8,000 | ......... |
| | | Elong. | 0.5 | 0.5 | 2.5 | 8.0 | 41.0 |
| | | B.H. | 110 | 111 | 78 | 56 | 50 |
| A–355–T59 | Sand cast | T.S. | 25,000 | 21,560 | 14,940 | 11,050 | 8,720 |
| | | Y.S. | 21,000 | 14,750 | ......... | 6,700 | ......... |
| | | Elong. | 2.0 | 2.0 | 2.5 | 8.0 | 10.0 |
| | | B.H. | 55 | 65 | 52 | 51 | 48 |
| 195–T4<br>ASTM–G<br>SAE–38 | Sand cast | T.S. | 31,000 | 27,590 | 15,900 | 9,820 | 4,050 |
| | | Y.S. | 16,100 | 19,750 | 9,750 | 6,100 | ......... |
| | | Elong. | 8.0 | 4.0 | 16.5 | 24.0 | 61.0 |
| | | B.H. | 60 | 78 | 55 | 49 | 39 |
| 254 | Sand cast | T.S. | 27,000 | 27,000 | 20,970 | 17,700 | 14,700 |
| | | Y.S. | 22,000 | 18,000 | 12,750 | 10,000 | ......... |
| | | Elong. | 2.0 | 2.0 | 3.0 | 3.5 | 11.5 |
| | | B.H. | 75 | 78 | 71 | 74 | 76 |

* Tensile strength, lb. per sq. in.
† Yield strength, lb. per sq. in.
‡ Elongation, percent in 2 in.
§ Brinell hardness at room temperature after stabilization and testing at elevated temperatures.
‖ Properties after 18 days at temperature; not completely stabilized.

associated with such temperatures, because of the stabilization of the alloys at the test temperature prior to the test itself.

Except for the values given at 500° F. for 132 alloy, the minimum time of exposure at test temperature prior to testing was 75 days. The effect on the material is indicated by the hardness values, which were determined at room temperature after the test piece had been stabilized at the test temperature. For example, the room temperature hardness of the 132 alloy changes from 130 to 54 by holding at 600° F. for 75 days.

For the properties of aluminum alloys at elevated temperatures, see R. L. Templin and D. A. Paul, The Mechanical Properties of Aluminum and Magnesium Alloys at Elevated Temperatures, in Symposium on Effect of Temperature on the Properties of Metals, issued jointly by A.S.T.M. and A.S.M.E., 1931.

## 3. WROUGHT ALUMINUM ALLOYS

**CLASSIFICATION.**—Aluminum wrought alloys may be classified as: 1. Alloys in which various tempers are produced by cold working; 2. "Strong alloys" or those in which the desired properties are obtained by heat treatment, and which in general have higher mechanical properties than those of (1).

Table 6 gives the composition and Table 7 the specific gravity and weight of commercial wrought alloys. The designating number indicates the composition. The suffix S indicates that the alloy is wrought or capable of being wrought.

**Temper** is designated by a suffix following the alloy designation, as follows: Soft or annealed alloys, suffix –O; hard temper, suffix –H; for intermediate cold-worked temper, a fraction precedes the suffix H. For example, 3S – $^{1}/_{2}$H indicates alloy 3S, half-hard temper. Tempers produced by cold working are defined by the tensile strengths of the products, rather than by the amount of reduction during fabrication.

**Table 6.—Nominal Composition of Wrought Aluminum Alloys**

| Alloy | 2S | 3S | 4S | 11S | 14S | 17S* | A17S | 18S | 24S | 25S | 32S | 51S | 52S | 53S |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Aluminum, % | 100 | 98.75 | 97.75 | 94 | 93.7 | 95 | 97.2 | 94 | 93.7 | 93.9 | 86.2 | 98.4 | 97.25 | 97.8 |
| Copper, % | .... | .... | .... | 5.0 | 4.4 | 4.0 | 2.5 | 4.0 | 4.2 | 4.5 | .... | .... | .... | .... |
| Silicon, % | .... | .... | .... | .... | 0.8 | .... | .... | .... | .... | 0.8 | 12.0 | 1.0 | .... | 0.7 |
| Manganese, % | .... | 1.25 | 1.25 | .... | 0.75 | 0.5 | .... | .... | 0.6 | 0.8 | .... | .... | .... | .... |
| Magnesium, % | .... | .... | 1.0 | .... | 0.35 | 0.5 | 0.3 | .... | 1.5 | .... | 1.0 | 0.6 | 2.5 | 1.25 |
| Lead, % | .... | .... | .... | 0.5 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| Nickel, % | .... | .... | .... | .... | .... | .... | .... | 2.0 | .... | .... | 0.8 | .... | .... | .... |
| Bismuth, % | .... | .... | .... | 0.5 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| Chromium, % | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 0.25 | 0.25 |

**Table 7.—Specific Gravity and Weight of Alloys**

| Alloy | Specific Gravity | Weight, lb. per cu. in. | Alloy | Specific Gravity | Weight, lb. per cu. in. |
|---|---|---|---|---|---|
| Pure Al (99.97%) | 2.70 | 0.0976 | 24S | 2.76 | 0.100 |
| 2S | 2.71 | 0.098 | 25S | 2.79 | 0.101 |
| 3S | 2.73 | 0.099 | 51S | 2.69 | 0.097 |
| 4S | 2.72 | 0.098 | 52S | 2.67 | 0.096 |
| 17S | 2.79 | 0.101 | 53S | 2.69 | 0.097 |

* Duralumin.

**Heat Treatment Designation.**—Strong alloys in which the maximum strength has been developed by a solution heat treatment, *i.e.*, heating to the proper temperature, quenching in cold water, and aging, are designated by the suffix —T.

Some strong alloys age at room temperature. Others must be subjected to higher temperatures to precipitate certain constituents. These latter alloys are designated 25S–W, 51S–W, 53S–W, and C17S–W when they have been quenched and not subjected to precipitation treatment, although 51S–W and C17S–W will age somewhat spontaneously at room temperature.

The strong alloys have lower mechanical properties and may be subjected to more difficult forming operations before aging, than when they are in the fully heat-treated "T" condition. Since the alloys which age at room temperature do not acquire the full properties of the "T" condition until about four days have elapsed after quenching, the practice of forming, particularly driving 17S rivets, immediately after quenching frequently is followed. Room temperature aging also can be effectively retarded by holding the material at a low temperature after quenching. This can be accomplished by packing in ice or in solid carbon dioxide (dry ice).

The suffix –RT indicates alloys that have been cold worked after heat treatment, by an amount carefully controlled to retain suitable ductility in the product. Such treatment produces in some alloys higher yield strengths than are obtained by heat treatment alone.

**MECHANICAL PROPERTIES.**—Table 8 gives the mechanical properties of wrought aluminum alloy sheet. These properties are generally applicable to all products, except that the elongation depends somewhat on the size and shape of test specimen. In certain products, as heavy bar and plate, the reduction may not be sufficient to develop the maximum properties of the alloy on subsequent heat treatment.

**EFFECTS OF COLD WORKING.**—Many commercial wrought products of aluminum and its alloys are supplied in various tempers determined by the amounts of cold working given the products after final annealing. See R. L. Templin in The Aluminum Industry, vol. II, chapter xix, p. 396, and the A.I.M. & M.E., Institute of Metals Division, p. 466, 1930, for a discussion of the effect of cold working on the physical properties of aluminum and its alloys. The formulas and diagrams below are taken from these references.

Cold working is defined as the change in cross-sectional area of the metal resulting from rolling, forging, drawing, or extrusion at about room temperature, *i.e.*, 60 to 80° F. Generally, change in cross-sectional area of the metal amounts to a reduction in area. In rolling wide sheet, a reduction in cross-sectional area of the product may be considered as resulting solely from the reduction in thickness of the product.

If aluminum and its alloys be worked at temperatures above room temperature, the effects of cold working are less in proportion to the amount of work done than when they are worked at room temperature. Conversely, the work done at temperatures below room temperature is more effective than the same amount of work at room temperature.

Cold working is expressed as the percentage change in the original cross-sectional area (commonly called *reduction*) as follows:

In general, $C = 100\ (A - a)/A$, where $C$ = amount of cold working, percent; $A$ = initial area of material in properly annealed condition; $a$ = area of the cold worked material.

For sheet, $C = 100\ (t_o - t_f)/t_o$, where $t_o$ = initial thickness of the material and $t_f$ = final thickness.

The effect of cold working on the tensile strength of wrought aluminum and its alloys is shown by $\text{T.S.} = \text{T S.}_o(1 + 0.9R)$, where T.S. = tensile strength in lb. per sq. in.; $\text{T.S.}_o$ = tensile strength of the properly-annealed material; $R$ = reduction in area expressed as a decimal.

The average tensile strength of the annealed commercial alloys which are fabricated into products of various intermediate tempers by cold working are as follows:

| Metals or Alloys | 2S–O (Pure Al) | 3S–O | 4S–O |
|---|---|---|---|
| Tensile strength, lb. per sq. in. | 14,000 | 15,500 | 26,000 |

### Table 8.—Typical Mechanical Properties of Wrought Aluminum Alloys *

Determined in accordance with A.S.T.M. Designation E8–33

| Alloy and Temper | Tension | | | Hardness | | Shear | Fatigue |
|---|---|---|---|---|---|---|---|
| | Yield Strength,† lb. per sq. in. | Ultimate Strength, lb. per sq. in. | Elongation,‡ percent in 2 in. | Brinell (500-kg. 10-mm. ball) | Shore (Magnifier Hammer) | Shearing Strength,§ lb. per sq. in. | Endurance Limit, ‖ lb. per sq. in. |
| 2S–O | 4,000 | 13,000 | 35 | 23 | 7 | 9,600 | 5,000 |
| 2S–1/4 H | 13,000 | 15,000 | 12 | 28 | 11 | 10,000 | 6,000 |
| 2S–1/2 H | 14,000 | 17,000 | 9 | 32 | 13 | 11,000 | 7,000 |
| 2S–3/4 H | 17,000 | 20,000 | 6 | 38 | 15 | 12,000 | 8,000 |
| 2S–H | 21,000 | 24,000 | 5 | 44 | 18 | 13,000 | 8,500 |
| 3S–O | 5,000 | 16,000 | 30 | 28 | 10 | 11,000 | 7,000 |
| 3S–1/4 H | 15,000 | 18,000 | 10 | 35 | 14 | 12,000 | 8,000 |
| 3S–1/2 H | 18,000 | 21,000 | 8 | 40 | 16 | 14,000 | 9,000 |
| 3S–3/4 H | 21,000 | 25,000 | 5 | 47 | 19 | 15,000 | 9,500 |
| 3S–H | 25,000 | 29,000 | 4 | 55 | 23 | 16,000 | 10,000 |
| 4S–O | 10,000 | 26,000 | 20 | 45 | 15 | 16,000 | 14,000 |
| 4S–1/4 H | 25,000 | 31,000 | 6 | 55 | 22 | 17,000 | 14,500 |
| 4S–1/2 H | 31,000 | 35,000 | 5 | 65 | 23 | 19,500 | 15,000 |
| 4S–3/4 H | 35,000 | 39,000 | 3 | 73 | 26 | 21,000 | 15,500 |
| 4S–H | 38,000 | 42,000 | 3 | 80 | 30 | 23,000 | 16,000 |
| 52S–O | 14,000 | 29,000 | 25 | 45 | .......... | 18,000 | 17,000 |
| 52S–1/4 H | 26,000 | 34,000 | 12 | 62 | .......... | 20,000 | 18,000 |
| 52S–1/2 H | 29,000 | 37,000 | 10 | 67 | .......... | 21,000 | 19,000 |
| 52S–3/4 H | 34,000 | 39,000 | 8 | 74 | .......... | 23,000 | 20,000 |
| 52S–H | 36,000 | 41,000 | 7 | 85 | .......... | 24,000 | 20,500 |
| 11S–T | 35,000 | 55,000 | 23 | 100 | 30 | 39,000 | .......... |
| 17S–O | 10,000 | 26,000 | 20 | 45 | 15 | 18,000 | 11,000 |
| 17S–T | 35,000 | 58,000 | 20 | 100 | 30 | 35,000 | 15,000 |
| 17S–RT | 46,000 | 61,000 | 13 | 110 | 35 | 36,000 | .......... |
| Alclad 17S–T | 32,000 | 55,000 | 18 | .......... | .......... | 32,000 | .......... |
| Alclad 17S–RT | 40,000 | 57,000 | 11 | .......... | .......... | 32,000 | .......... |
| A17S–O | 8,000 | 22,000 | 24 | 38 | 11 | 15,000 | .......... |
| A17S–T | 24,000 | 43,000 | 24 | 70 | 22 | 25,000 | 13,500 |
| 24S–O | 10,000 | 26,000 | 20 | 42 | .......... | 18,000 | .......... |
| 24S–T | 43,000 | 65,000 | 20 | 105 | .......... | 40,000 | 14,000 |
| 24S–RT | 53,000 | 68,000 | 13 | 116 | .......... | 41,000 | 14,500 |
| Alclad 24S–T | 40,000 | 60,000 | 18 | .......... | .......... | 39,000 | .......... |
| Alclad 24S–RT | 49,000 | 62,000 | 11 | .......... | .......... | 39,000 | .......... |
| 25S–O | 10,000 | 26,000 | 20 | 45 | 15 | 18,000 | 9,000 |
| 25S–W | 25,000 | 48,000 | 18 | 80 | 23 | 30,000 | 14,500 |
| 25S–T | 35,000 | 58,000 | 20 | 100 | 30 | 35,000 | 15,000 |
| 27S–T | 50,000 | 60,000 | 9 | 115 | .......... | 37,000 | 13,000 |
| 51S–O | 6,000 | 16,000 | 30 | 28 | 10 | 11,000 | 6,500 |
| 51S–W | 20,000 | 35,000 | 24 | 64 | 20 | 24,000 | 10,500 |
| 51S–T | 38,000 | 48,000 | 14 | 95 | 28 | 30,000 | 10,500 |
| 53S–O | 7,000 | 16,000 | 25 | 26 | 9 | 11,000 | 7,500 |
| 53S–W | 20,000 | 33,000 | 22 | 65 | 21 | 22,000 | 10,000 |
| 53S–T | 32,000 | 38,000 | 14 | 80 | 26 | 26,000 | 11,000 |

[*] Modulus of elasticity is approximately 10,300,000 lb. per sq. in.

[†] Stress at which stress-deformation curve departs 0.2 percent from the initial modulus line produced. The yield strength in compression is substantially equal to the yield strength in tension for wrought alloys.

[‡] Values for 14-gage sheet; for heavier sections the elongations are higher.

[§] Single-shear strength values obtained from double-shear tests.

[‖] R. R. Moore type of rotating-beam machine; endurance limit based on 500,000,000 cycles.

**Table 9.—Relation of Cold Working to Temper**

| Temper of metal | Soft | 1/4 Hard | 1/2 Hard | 3/4 Hard | Hard |
|---|---|---|---|---|---|
| Approximate amount of cold working * | 0 | 21 | 37 | 60 | 80 |
| Approximate B. & S. gage No. reduction | | | | | |
| Sheet | 0 | 2 | 4 | 8 | 14 |
| Wire | 0 | 1 | 2 | 4 | 7 |

* Percent reduction in cross-sectional area.

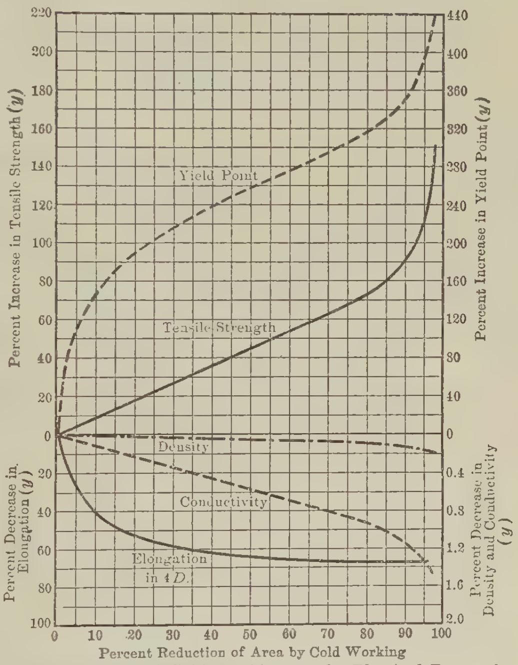

FIG. 1. Effects of Cold Working on the Physical Properties of Aluminum

Table 9 gives the nominal amounts of cold working together with the approximate Brown & Sharpe gage numbers reduction for the various tempers.

The yield strength of aluminum and its alloys is the stress corresponding to a limiting permanent set of 0.2%. In properly-annealed wrought metal this value will be approximately one-third of the ultimate tensile strength. For all of the other tempers of a cold-worked product, the yield strength will vary from 80 to 95% of the ultimate tensile strength.

Elongation is affected considerably both by the size and shape of the test specimen, as well as by the ductility and elongation.

The relation between tensile strength and Brinell hardness of wrought aluminum and its alloys is approximately T.S. = 755 × Brinell hardness number.

Fig. 1 shows graphically the effects of cold working on the physical properties of aluminum and its alloys.

**COLUMN FORMULAS.**—The following formulas may be used in the design of columns of aluminum subjected to axial loads:

$$\text{Round Ends}\begin{cases}\text{For } L/r \text{ values of 0 to } C, & f = B - K(L/r)\\ \text{For } L/r \text{ values of } C \text{ and up} & f = 33{,}000{,}000/(L/r)^2\end{cases}$$

$$\text{Fixed Ends}\begin{cases}\text{For } L/r \text{ values of 0 to } C & f = B - K(L/r)\\ \text{For } L/r \text{ values } C \text{ and up} & f = 132{,}000{,}000/(L/r)^2\end{cases}$$

where $f$ = safe design stress in axially-loaded column, lb. per sq. in.; $L$ = unsupported length of column, in.; $r$ = least radius of gyration of column, in.; $L/r$ = slenderness ratio of column. Values of $C$, $B$ and $K$ are given in Table 10.

**Table 10.—Values of Constants for Column Formulas**

| Alloy and Temper | Round Ends | | | Fixed Ends | | |
|---|---|---|---|---|---|---|
| | *C* | *B* | *K* | *C* | *B* | *K* |
| 3S–1/2 H | 128 | 6,000 | 31 | 256 | 6,000 | 15.5 |
| 3S–H | 111 | 8,000 | 48 | 222 | 8,000 | 24 |
| 4S–O | 128 | 6,000 | 31 | 256 | 6,000 | 15.5 |
| 4S–1/2 H | 100 | 10,000 | 67 | 200 | 10,000 | 33.5 |
| 4S–H | 87 | 13,000 | 100 | 174 | 13,000 | 50.0 |
| 17S–T | 81 | 15,000 | 123 | 162 | 15,000 | 61.5 |
| 51S–W | 105 | 9,000 | 57 | 210 | 9,000 | 28.5 |
| 51S–T | 84 | 14,000 | 111 | 168 | 14,000 | 55.5 |

**DESIGN STRESSES FOR ALUMINUM ALLOYS.**—In Table 11 are given the stresses considered safe for ordinary structural design at temperatures below 100° F. Where temperatures above that point are to be considered, the values in Table 11 should be multiplied by the factors in Table 12.

**Table 11.—Design Stresses for Aluminum Alloys**

Applicable to temperatures below 100° F.

| | Alloy and Temper | Tension, lb. per sq. in. | Compression, lb. per sq. in. | Bearing, lb. per sq. in. | Shear, lb. per sq. in. |
|---|---|---|---|---|---|
| Wrought | 3S–O | 4,000 | 4,000 | 7,000 | 3000 |
| | 3S–1/2 H | 6,000 | 6,000 | 9,000 | 3500 |
| | 3S–H | 8,000 | 8,000 | 13,000 | 4000 |
| | 4S–O | 6,000 | 6,000 | 11,000 | 4000 |
| | 4S–1/4 H | 9,000 | 9,000 | 14,000 | 4500 |
| | 4S–1/2 H | 10,000 | 10,000 | 15,000 | 5000 |
| | 4S–H | 13,000 | 13,000 | 19,000 | 6000 |
| | 17S–T | 15,000 | 15,000 | 26,000 | 9000 |
| | 51S–W | 9,000 | 9,000 | 15,000 | 6000 |
| | 51S–T | 14,000 | 14,000 | 21,000 | 7500 |
| Sand Cast | 12 | 5,000 | 9,000 | 9,000 | 5000 |
| | 43 | 5,000 | 6,000 | 6,000 | 4000 |
| | 195–T62 | 11,000 | 15,000 | 18,000 | 7000 |
| | 220–T4 | 11,000 | 15,000 | 18,000 | 8000 |
| | 356–T6 | 10,000 | 12,000 | 15,000 | 6000 |

**Table 12.—Factors for Design Stresses at Elevated Temperatures ***

| | Alloy and Temper | 100° F. | 200° F. | 300° F. | 400° F. | 500° F. |
|---|---|---|---|---|---|---|
| Wrought | 3S–O | 1.0 | 0.85 | 0.70 | 0.60 | 0.40 |
| | 3S–1/2 H | 1.0 | 0.75 | 0.60 | 0.45 | 0.30 |
| | 3S–H | 1.0 | 0.75 | 0.60 | 0.35 | 0.25 |
| | 4S–O | 1.0 | 0.90 | 0.80 | 0.50 | 0.40 |
| | 4S–1/2 H | 1.0 | 0.90 | 0.65 | 0.30 | 0.25 |
| | 4S–H | 1.0 | 0.80 | 0.55 | 0.25 | 0.20 |
| | 17S–T | 1.0 | 0.90 | 0.60 | 0.30 | 0.15 |
| | 51S–W | 1.0 | 1.00 | 0.80 | 0.30 | 0.15 |
| | 51S–T | 1.0 | 0.75 | 0.55 | 0.20 | 0.10 |
| Sand Cast | 220–T4 | 1.0 | 0.85 | 0.70 | 0.50 | 0.30 |

* To be applied to values of Table 11.

## 4. HEAT TREATMENT OF ALUMINUM ALLOYS

**SOLUTION HEAT TREATMENT.**—A change in properties as a result of heat treatment of aluminum alloys usually is associated with a change in solubility of one or more of the alloying ingredients with change in temperature. Thus, over 5% copper is soluble in solid aluminum just below the melting point of the eutectic; at room temperature the solubility is well below 1%. If aluminum alloys containing 4% or more of copper be heated to a temperature slightly below the melting point of the eutectic, the copper will dissolve in the aluminum at that temperature. If the alloy then be cooled quickly to a low temperature, it becomes supersaturated in copper at the lower temperature. This treatment is known as a solution heat treatment. In certain alloys, like 17S or duralumin, the supersaturated solid breaks down at ordinary temperatures. This change, known as aging, changes the physical properties.

In carrying out the solution heat treatment, temperature must be closely controlled. Thin pieces of wrought metal may be heated to the proper temperature and held there only a short time before cooling, with good results. The operation may be carried on in a hearth or pit furnace or in a bath of fused niter. Castings and large wrought pieces, however, must be held at the solution heat treating temperature for considerable periods of time, and automatic temperature control practically is necessary. Cooling, after the proper heating cycle, may be done in cold or hot water, oil, or an air blast.

**PRECIPITATION HEAT TREATMENT.**—In certain alloys, the changes produced at ordinary temperatures are insufficient for the requirements, and the desired results are effected by elevated temperatures. Such a treatment is known as the precipitation heat treatment, or artificial aging. In general, aging will increase hardness and strength of aluminum alloys. The same is true of the precipitation heat treatment, if it is not carried too far. The latter, assuming prior solution heat treatment, is capable of producing a wide range of properties without change in the alloy composition.

The precipitation heat treatment may be effected by heating in pressure vessels with steam, or by heating in baths such as oil, or in hearth or pit furnaces. The temperature should be well controlled for best results, and if an ordinary furnace is used, air circulation is desirable. The rate of cooling after the precipitation heating usually is not an important factor.

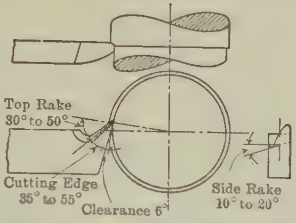

Fig. 2. Lathe Tool That May Be Used with Aluminum

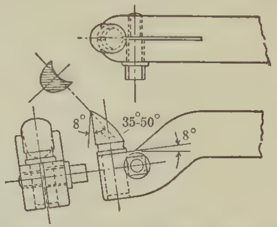

Fig. 3. Lathe Tool Recommended for Use with Aluminum

**OTHER EFFECTS OF HEAT TREATMENT** of aluminum alloys are the removal of internal strains, stabilization of dimensions of parts used or processed at elevated temperatures, and change in particle size of constituents such as silicon, which may be present in amounts greater than can be dissolved in solid aluminum even at the eutectic temperature.

## 5. FABRICATION OF ALUMINUM

**FORGING.**—Several of the strong aluminum alloys may be forged hot either under a hammer or in a forging press. Alloy 25S T is used for general forgings including connecting-rods, aircraft propellers and crank-cases, because of its superior hot working characteristics. Alloy 51S–T is used for crank-cases and nose pieces of more intricate design for radial aircraft engines, since this alloy is even more easily forged into the large thin sections. For a more limited class of forgings 17S also is used. Forged pistons are commonly manufactured from 32S which has a low coefficient of expansion, or from 18S.

**MACHINING.**—Aluminum and its alloys may be machined readily. In many instances, satisfactory results may be obtained by using tools ordinarily used for steel, provided they are sharp and in good condition. Owing to its inherent machining characteristics, better results may be obtained, however, when certain changes in tool shape and machining practice are observed. Generally speaking, aluminum and most of its alloys have familiar machining characteristics.

Cutting tools for aluminum resemble those for cutting hard wood in many respects. It should be emphasized that the cutting edges must be ground to a sharp included angle, must be keen, and should be provided with smoothly finished surfaces. The tools generally are ground with fine abrasive wheels, supplemented by hand stoning with an oilstone to give the desired finish of the surfaces and edges.

**Lathe, Shaper and Planer Tools.** Figs. 2–5 illustrate typical lathe and planer tools for machining aluminum. Attention is called to the unusually large amount of top and side rake.

**Milling Cutters** for aluminum should be of the coarse-tooth spiral type, with a considerable amount of top rake on the cutting edges. The more recent helical milling cutters, designed primarily for machining steel, work especially well with aluminum and its alloys, if the cutting edges are provided with suitable top rake.

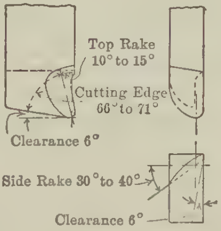

Fig. 4. Planer Tool for Roughing Aluminum

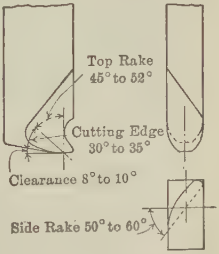

Fig. 5. Planer Tool—Finishing

**Drills.**—Ordinary twist drills work fairly satisfactorily in aluminum if properly ground with keen edges, but a better type of drill is one with more twist to the flutes.

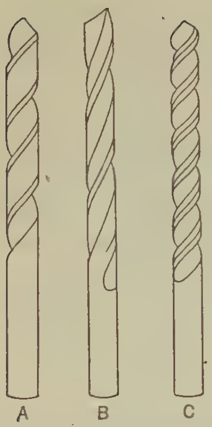

FIG. 6. Three Types of Twist Drills, $^3/_8$ in. diam.

*A*, Single flute twist-drill, 40° spiral angle
*B*, Double flute or standard twist-drill, 24° spiral angle
*C*, Special double fluted twist drill, 47° spiral angle

Both types are illustrated in Fig. 6. Fairly high speeds may be employed if ample cutting compound is used.

**Reamers** of the spiral fluted type produce the best results for reaming holes in aluminum.

**Taps** generally designated "gun" taps or spiral pointed taps, Fig. 7, which have a short spiral ground at the nose, work very satisfactorily in aluminum, especially for through holes where the cuttings can be pushed ahead. Better results may be obtained by the use of spiral fluted taps, now offered by a number of the manufacturers of small tools.

**Sawing.**—The shapes of saw teeth should be patterned after those of the cutting tools described above, so far as practicable. Teeth usually should have some top rake or hook, the amount depending on the type of saw, although very little can be used with band saws or circular saws where the work is fed by hand. The teeth should be comparatively coarse, with curved gullets, free from sharp corners and burrs or scratches which are apt to pick up cuttings.

Circular saws for aluminum may be operated at peripheral speeds of about 10,000 ft. per min. Speeds of about 5,000 ft. per min. are recommended for band saws. It is important that the machine be free from vibration, otherwise fatigue failure of saw blades or saw teeth will ensue.

A cutting lubricant or coolant is desirable, and for heavy work is necessary. Mineral-base lubricating oils thinned with kerosene and, in some instances, soluble oil cutting compounds have been found satisfactory.

Hand hack-saw blades of the "wavy-set" type work especially well with aluminum.

**Files.**—Some of the coarse-tooth files with top and side rake to the cutting teeth work well on aluminum.

**Cutting Speeds and Feeds.**—A wide range of cutting speeds and feeds may be used in machining aluminum and its alloys. Roughing cuts may be quite heavy. For highly finished surfaces, cuts only a few thousandths of an inch deep may be used, provided the cutting tools are properly shaped and finished. Speeds necessarily depend on such factors as the kind of machining operation, type and condition of machine tool and kind of cutting material. Cutting speeds faster than those ordinarily used for free-cutting brass have been used with success in some instances, while in others better results have been obtained by using slower speeds.

**Cutting Lubricants.**—Some of the alloys of aluminum have been machined successfully without any lubricant or coolant, using tools of the form described, but the use of some form of compound is desirable for best results. The soluble cutting oils usually are satisfactory for milling, sawing, and drilling. For such operations as turning, shaping, and tapping, a mixture of equal parts of lard oil and kerosene generally is recommended. Cutting compounds of paraffin oil, such as are used in machining brass, are not satisfactory for aluminum.

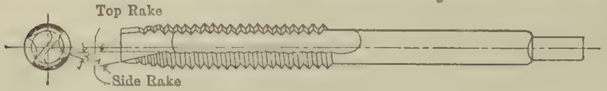

FIG. 7. "Gun" Tap

**Tool Materials.**—Tools for machining aluminum may be either of high-carbon or high-speed steel. Cemented tungsten-carbide tools have been found indispensable, especially for machining aluminum alloys with high silicon content; they also are excellent for high production.

## 6. JOINING OF ALUMINUM

**WELDING.**—Aluminum and its alloys may be welded by the torch, electric arc or resistance processes. In fusion welding, the equipment is the same as that used for welding steel, but in electrical resistance welding, machines of slightly different design and greater electrical capacity are required. The technique varies slightly from that required for steel. The metal arc generally will make stronger welds than can be made with the torch, but $^1/_{16}$ in. is the minimum thickness of metal that can be so welded.

For metal arc electrodes, and for filler rod for torch welds of strong alloys, an alloy of 5% Si, 95% Al is used. For torch welds of common alloys, as 2S and 3S, especially if a polished surface is desired, pure aluminum filler rod is used. A flux is used in fusion welding. In the metal arc process, the flux is best applied as a heavy coating on the electrode.

A welded joint, except in fully-annealed material, will have lower mechanical properties than those of the unwelded metal. The heat of welding has an annealing effect on the common alloys, and a somewhat deleterious heat treating effect on the strong alloys. In some cases, it also affects adversely the corrosion resistance. Heat treatment of strong alloy welds will improve the properties of the joint. If maximum efficiency of the joint is to be developed, a well-designed mechanical joint is superior to torch welds, and often superior to arc welds.

**SOLDERING** of aluminum is more difficult than the soldering of other non-ferrous metals. With ordinary soldering technique, oxide film on aluminum prevents wetting the metal. Many solders fail to make corrosion-resistant joints when applied to aluminum. However, there are available a variety of solders and fluxes adapted to nearly all soldering requirements. The use of the proper solder and flux enables aluminum parts to be joined in much the same way and for about the same type of service as soldered joints of other metals. Suitable directions can be obtained from the supplier of solder and flux.

**RIVETING.**—In general, the alloy used for rivets for aluminum alloy structures is the same as that of the structure, if sufficient strength may be obtained in this way. When greater strength is needed, rivets are made from stronger alloys than the rest of the structure; sometimes steel rivets are used.

Aluminum alloy rivets should be driven cold, where this is possible. Rivets of non-heat-treated alloys, as 3S and 4S, and some of heat-treated alloys, as 51S–W, are ready for driving when received from the maker. Heat-treated alloy rivets, as 17S, which are too hard to drive satisfactorily when fully heat-treated (T temper) and which do not have any stable intermediate temper (W temper), must be heat-treated immediately before driving. The treatment consists of heating to a temperature of 940 to 960° F. and quenching in cold water. For about 2 to 4 hours after quenching, the rivets are soft enough to be driven with comparative ease. Rivets so driven become fully aged in about 4 days, after which they exhibit the properties of 17S–T alloy. Most of this aging takes place in the first day, which accounts for the difficulty of driving after the lapse of 2 to 4 hours after quenching. Rivets that become too hard to drive may be re-heat-treated. The rate of hardening of 17S alloy rivets may be retarded by storing the quenched rivets at low temperatures. Temperatures of 32° F., for example, will keep the rivets soft enough for driving for at least 24 hours. Electric refrigeration or dry ice may be used to maintain even lower temperatures, with a resulting substantial decrease in the rate of hardening.

Strong aluminum alloy rivets up to $^3/_8$ in. may be driven cold, but cold driving of larger sizes is almost impossible, unless large squeeze riveters are available. Hot-driven rivets should be heated to heat-treating temperature (940 to 960° F. for 17S) and driven with a minimum of delay. The contact of the rivet with the relatively cold material around the hole provides a quench. Rivets as large as 1 in. have been driven successfully in this manner, and the properties of the material are practically as good as water-quenched material. In hot driving the heating equipment must have exact temperature control and be close to the work. In general, the squeeze method is best for driving aluminum alloy rivets. If pneumatic hammers are used, they should be of large size, as a few heavy blows are superior to a number of light, rapid blows.

Steel rivets are used in aluminum alloy structures only when the heating effect is of no consequence, and rusting may be prevented. Then the relatively large shear strength of steel rivets, coupled with the ease of driving, makes them very desirable. The procedure for driving hot steel rivets in aluminum alloy structures is the same as for steel structures, except that it is advisable to drive the rivets at random in order to avoid localized heating.

**Rivet Holes** in aluminum may be drilled, punched, or sub-punched and reamed as desired. When holes are drilled or sub-punched and reamed, the diameter of the hole generally is made $^1/_{32}$ in. greater than the diameter of the rivet. For heavier work, where holes are punched, the hole generally is made $^1/_{16}$ in. greater in diameter than the rivet. Taper bridge reamers with spiral flutes are recommended for reaming sub-punched holes to size.

## 7. FINISHING OF ALUMINUM

**FROSTED FINISH** usually is applied to aluminum by etching in a hot solution of sodium hydroxide (caustic soda) followed by treatment with strong nitric acid. For ordinary operations, the solution contains from 2 to 5% of caustic soda, although solutions containing as much as 25% are used. The time of immersion is determined by the temperature of the solution, its concentration, and the depth of etch required. Usually treatments of about one minute are sufficient.

After removal from the caustic solution, the object is rinsed in clear water and immersed in strong nitric acid. A satisfactory solution is two parts concentrated nitric acid and one part water. Occasionally, a weaker solution, say 10% to 15% nitric acid, may be used. The nitric acid has little solvent action on the aluminum, but removes the colored film left on the surface by the caustic dip and leaves the metal with a clean "frosted" finish.

**ELECTROPLATING.**—Aluminum may be electroplated with other metals. Such plating is used primarily for ornamental reasons rather than as a protection against corrosion. The aluminum surface first is roughened by special etching procedures, which depend on the alloy being coated, to provide a firm anchorage for subsequently-applied nickel plating. Once the nickel has been properly applied, any desired metal may be plated over it. Zinc applied directly to the aluminum surface will electrochemically protect the aluminum. Chromium directly applied will resist abrasion or chemical attack by mild alkalis.

**OXIDE COATINGS.**—The electrolytic or anodic oxide coating process, sometimes known as deplating, is carried out in a special electrolyte in which the piece to be finished is immersed as the anode (positive electrode). Applied to aluminum and many of its alloys, it forms a hard, abrasion-resistant coating of superior protective value. It usually is white to light gray in appearance, but on aluminum-silicon alloys it forms a dark gray coating, which is of artistic value, particularly on sand blasted castings. The electrolytic oxide coating also may be given marked absorptive properties, which permit its being colored by dyes and pigments. They combine the luster and brilliance of the metal with the color of the dye, which is highly attractive.

**PAINTING AND LACQUERING.**—Aluminum may be coated with paint or lacquer by much the same technique as followed for other metals. Surface preparation is important. Normally, all that is required is a cleansing from grease and dirt by wiping the surface with mineral spirits or other solvent. Certain chemical cleansing agents, as a phosphoric acid-alcohol mixture, sold under various trade formulas, not only clean the surface but give greatly improved paint adhesion where service conditions are severe. With castings, a light sand blast improves adhesion of paint. Sheet or extruded shapes, with either an electrolytic or chemical oxide coating, provide exceptionally good adhesion, especially in painting for protection against salt water corrosion.

Paint in the form of aluminum paint, oil-base enamels or lacquers, may be applied by spraying, brushing, or dipping. Synthetic resin varnishes and enamels materially increase the durability of such finishes. Except under severe exposure conditions, such as on aircraft, a special priming coat on aluminum surfaces is unnecessary. Where extra protection is desired, a priming coat containing zinc chromate pigment is effective.

Transparent varnishes or lacquers may be used to preserve polished surfaces. Many of these are sufficiently durable to be applied to exterior ornamental work. A varnish of light color and one which is relatively insensitive to light exposure, should be selected. Synthetic resins have increased durability; no clear finish, however, will last as long as pigmented finishes.

**Aluminum Paint** is made by mixing aluminum-bronze powder with a suitable vehicle. Aluminum-bronze powder is supplied either as a dry powder or as a paste mixed with mineral spirits. Aluminum paint, in the proportion of two pounds of powder to a gallon of vehicle, forms a coating of high moisture resistance and superior protective qualities. By proper choice of vehicle it can be used on metal, wood, plaster, or concrete, and is particularly recommended for the painting of aluminum and its alloys.

**POLISHING.**—Aluminum in any form can be polished or buffed. The degree of polish depends on the hardness and freedom from porosity of the metal. Thus, a piece of hard-rolled sheet will take a more brilliant polish than a soft casting. Polishing may consist of four steps, although if the surface is already quite smooth, as in a cooking utensil made from sheet, only the last two operations may be necessary. See also pp. 21–88 to 21–96.

**1. Roughing (Canvas or Rag Wheel).**—The wheel is made of sewed buffs, glued together. The polishing head, of No. 60 to No. 100 emery or silicon-carbide abrasive, is glued to the face of the wheel, which should have a peripheral speed of about 6,000 ft. per min. A lubricant consisting of polishing oil and tallow generally is used, the surface of the article being smeared with lubricant during the operation.

**2. Oiling.**—A solid felt or compress canvas wheel, with a polishing head of No. 100 to No. 160 emery or silicon-carbide abrasive glued to the face, is used. The same lubricant as for roughing is used, applied to the wheel.

**3. Buffing.**—The buffing wheel is made up of sewed-piece or full disc packed buffs. The full disc buffs should be a close weave, say 84 × 92. The shape of the wheel is determined by the shape of the work. Tripoli composition is satisfactory for polishing aluminum. The peripheral speed of buffing wheels generally is 7000 to 7500 ft. per min. After buffing, the work is cleaned in benzine and dried in sawdust.

**4. Coloring** is best done with a full disc buff or one of the varieties of patented folded buff. Tripoli, rouge, and soft silica compositions are used to bring out the silvery white color of aluminum.

**SCRATCH BRUSH AND SATIN FINISH.**—Aluminum can be so finished as to diffuse the rays of light, producing a surface similar in appearance to frosted or etched glass. These finishes, produced by the use of a rotating wire brush, are called "Scratch Brush" or "Satin" finish. A scratch brush finish may be done in a large lathe, using a soft steel-wire brush, about 10 in. outside diameter, revolving about 2000 r.p.m. The wire size is about 0.015 in. diameter. A satin finish, of finer texture, is produced, using wire brush wheels with wires about 0.005 or 0.003 in. diameter. A satin finish also may be obtained by means of a buffing wheel, using Lea greaseless compound and Learock on the wheel. See booklet, Buffing and Polishing Methods, issued by the Lea Mfg. Co., Waterbury, Conn.

**SAND BLAST FINISH** frequently is applied to castings. It is of varying degrees of fineness, depending on the nature of the work and the ultimate finish desired. The use of coarse sand is satisfactory for surfaces which are to be oxide coated, but is somewhat limited in its use because of its heavy abrasive action, resulting in excessive warpage of large thin sections. The finer sizes of sand are used for high-lighted and satin-finished work, since their milder abrasive action reduces the subsequent cost of grinding and polishing.

## 8. FABRICATED ALUMINUM PRODUCTS

**SHEET AND PLATE.**—Plate can be supplied up to 2 in. thick, with a maximum weight limitation of 2000 lb. per individual plate. Sheet from No. 10 to 36 gage also can be supplied in coils. Tables of commercial sizes of sheet and plate are given on pp. 6–08 to 6–12.

**TUBING.**—The operations which are necessary to straighten 17S–T tubing, after it has been quenched, increase the yield strength by about 25% as compared with the value normally obtained in commercial sheet. Tubing in round, square, and streamline cross-section also is produced in 17S–RT, having a yield strength approximately 30% higher than that of 17S–T tubing, and about 60% higher than that of 17S–T sheet. If these materials are heat-treated by the user, yield strengths comparable to those of 17S–T sheet will be obtained. Tubing also is available in other shapes, such as rectangular, hexagonal, octagonal, streamline, and odd shapes.

**STRUCTURAL SECTIONS.**—Aluminum is rolled into various structural shapes, as beams, channels, angles, tees, zees, and streamlined shapes. For dimensions and properties, see pp. 6–12 to 6–16.

**WIRE, ROD, AND BAR** are available in a wide variety of sizes and shapes. Wire is produced by drawing from large rod sizes to the desired dimensions; flattened wire by rolling round wire to produce a strip with round edges; flattened and slit wire and strip are produced by rolling a round wire or a rectangle to the desired width to produce a strip with square edges. Round wire is made in sizes from No. 00 B. & S. gage to 38 gage, and round rod up to 8 in. diameter.

**ALCLAD PRODUCTS,** in sheet or wire, consist of surface material of exceedingly high corrosion resistance, alloyed and integral with a strong alloy core. The thickness of the surface metal is such as to retain maximum physical properties consistent with adequate protection of the core. The coating protects the alloy which it covers, and also electrolytically protects exposed edges, or base alloy which may be exposed by scratches or abrasions. It also protects 17S rivets, which are used to join Alclad sheet.

**ALUMINUM FOIL.**—Aluminum sheet 0.005 in. and less in thickness is termed foil. Foil may be rolled commercially to gages as thin as 0.00025 in., but thicknesses of 0.0003 to 0.0007 in. are commonly used. For these extremely thin gages a special grade of pure metal is used. Where additional strength is required, as for diaphragms and shims, and other purposes, alloys of aluminum are used. Alloy foils, however, being less ductile, usually are 0.001 in. thick and heavier.

Aluminum foil is used widely in electrolytic condensers for radio equipment, power factor condensers, etc., because of its film forming characteristic. It also is used in foil and paper type condensers, in contacts for burglar alarm systems, protecting tape for underground cables, etc. Aluminum foil, either plain or backed with waxed paper and lacquered or printed in colors, is used for articles which must be protected from the atmosphere or from handling, including photographic films, friction tape, typewriter ribbons, etc. Its non-toxic quality has resulted in its being used extensively for wrapping food

**Table 13.—Aluminum Foil**

| Thickness | | Area, sq. in. per lb. | Thickness | | Area, sq. in. per lb. | Thickness | | Area, sq. in. per lb. |
|---|---|---|---|---|---|---|---|---|
| in. | mm. | | in. | mm. | | in. | mm. | |
| 0.00025 | 0.00635 | 41,000 | 0.0006 | 0.01524 | 17,000 | 0.0015 | 0.03810 | 6800 |
| .0003 | .00762 | 34,100 | .00065 | .01651 | 15,700 | .0020 | .05080 | 5100 |
| .00035 | .00889 | 29,200 | .0007 | .01778 | 14,600 | .0025 | .06350 | 4100 |
| .0004 | .01016 | 25,600 | .0008 | .02032 | 12,800 | .0030 | .07620 | 3400 |
| .00045 | .01143 | 22,700 | .0009 | .02286 | 11,400 | .0040 | .10160 | 2550 |
| .0005 | .01270 | 20,500 | .0010 | .02540 | 10,250 | .0050 | .12700 | 2050 |
| .00055 | .01397 | 18,600 | ...... | ...... | ...... | ...... | ...... | .... |

products. It also is used in the manufacture of artificial flowers, light-bulb reflectors, box wrapper and liners, labels, can liners, bottle caps and cap liners, magazine and book covers. Aluminum foil is regularly supplied in coils or in rectangular form. It may be paper interleaved, wax paper backed, or cement or asphalt paper mounted, embossed, printed, or lacquered in standard or special colors or designs.

Foil thinner than 0.001 in. does not permit accurate measurements of thickness by a micrometer. Gages thinner than 0.001 in. are determined by weighing a sample of foil of a known area and calculating the thickness, using the value of 2.70 as the density of aluminum. Covering area in square inches per pound of foil for foil gages 0.005 in. and lighter are given in Table 13.

## 9. ELECTRICAL USES OF ALUMINUM

Aluminum was first used as a conductor in high tension wires in the U. S. in 1898. The first conductors were made entirely of aluminum, but later a cable with an extra high strength steel reinforcing core for the aluminum strands was developed and has since been used with marked success.

**CONDUCTIVITY.**—The A.I.E.E. standard of conductivity for hard drawn aluminum wire is 60.97% of the International Annealed Copper Standard. On this scale the conductivity of commercial hard-drawn copper usually is taken at 97%.

The electrical conductivity of annealed high-purity (99.97%) aluminum is 64.6%, based on the International Annealed Copper Standard. The electrical conductivity is lowered by the addition of alloying elements, the reduction varying with the nature of the element and the amount added. Heat treatment and mechanical working also influence this property to a marked degree. The following are the approximate conductivities of various grades of aluminum and aluminum alloys:

| Metal............ | 99.97% Al | Hard-drawn Conductors | 2S–O | 3S–O | 3S–H | 17S–T | 25S–T | 51S–T |
|---|---|---|---|---|---|---|---|---|
| Conductivity, percent of Int. Annealed Copper Std...... | 64.6 | 61 | 57 | 50 | 40 | 30 | 35 | 45 |

**CABLES AND CONDUCTORS.**—Aluminum cable, steel-reinforced, is used for messenger cables of overhead catenary railway lines. In the smaller sizes, it also is used for signal circuits, and for telephone lines supported on the same structure with power lines, where its high strength permits the use of as long spans as the transmission conductors. Large all-aluminum cables have been used as railway feeders, where they carry heavy currents at relatively low voltages. Particularly in large conductors, the use of aluminum effects a great saving in weight. Conductors for high-voltage installations such as outdoor sub-stations may advantageously be made of aluminum tubing. These are rigid and light, and permit the use of long spans which often are necessary to obtain electrical clearances, and show economy in supporting structures.

**SWITCHBOARD CONSTRUCTION.**—Aluminum may be used to advantage in bus bar construction, both on switchboards and for general transmission of power. Aluminum bus bars are rolled and are available in the usual rectangular cross-sections, as well as in channels and certain other structural shapes, which offer definite advantages in strength and current-carrying capacity. The current rating of aluminum bus bars may be conveniently taken as 80% of the accepted values for copper of equal cross-section and similar shape.

**ELECTRICAL MACHINERY.**—Aluminum is used for the rotor windings of squirrel-cage induction motors and also for field windings in certain types of motors and high-speed turbo-generators. The remarkable property of insulating aluminum wire by means of an electrolytically-applied oxide film is of service in constructing windings for electrical machinery where space is limited, or where a heat-resistant insulation is necessary.

## 10. STRUCTURAL USES OF ALUMINUM

Aluminum alloys are used in many structural fields where savings in dead weight, or freedom from rust stains are particularly desirable. They are widely used in aircraft, and for the main structural parts of truck bodies, railroad cars, storage tanks, dragline crane booms, traveling shop cranes, bridges, and special types of buildings. They may be sheared, sawed, punched, and otherwise handled in the routine fabricating operations without difficulty, and with the same tools that ordinarily are used in structural steel shops. Certain alloys are more important than others. These are as follows:

**3S Alloy** is used where welding is necessary, or in structural work where strength is not of primary importance. The hardest temper suitable to the necessary shop work generally is used. Typical applications are tank cars, storage containers, car and bus body plates, corrugated roofing, etc.

**4S Alloy** is used for applications similar to those of 3S alloy. For a given temper, however, 4S is considerably stronger than 3S. In the harder tempers, yield strength is comparable to that of the heat-treated alloys, but the ultimate strength is lower and the amount of forming which can be done is limited.

**17S Alloy** ages naturally at room temperatures following heat treatment, and hence does not have a stable as-quenched condition. Its tensile and yield strengths compare with those of ordinary structural steel, and the alloy is used for important strength members. When cold worked after heat treatment, the alloy, known as 17S–RT, has higher properties, as indicated in Table 8. 17S–RT tubing and sheet are widely used in aircraft construction.

**51S Alloy** in the as-quenched, or W, condition is adapted to severe forming operations. It can be artificially aged to 51S–T after forming, thereby gaining additional strength. 51S–T has the same yield strength as 17S–T, but its other mechanical properties are somewhat lower. See Table 8. 51S–W is used in body framing of railway cars and buses, and in various welded constructions where the strength of 3S is insufficient.

**52S Alloy** has mechanical properties comparable with those of 4S, and forming characteristics which compare with those of 3S. Rolled or extruded shapes, however, cannot be cold worked sufficiently to obtain the properties of the harder tempers. The properties of shapes usually are intermediate between those of the annealed and $^1/_4$ H tempers. Largely because of this, the structural uses of 52S generally are confined to the forms of sheet and plate.

**53S Alloy** is quite similar to 51S. Its mechanical properties are slightly lower and its ductility somewhat greater. It has better corrosion resistance, especially when compared with 51S–T. Its uses are similar to those of 51S alloy. Additional uses are window sash, railings, store-fronts, and various architectural trim.

**Casting Alloys.**—No. 12 alloy can be used structurally to about the same extent as cast iron. When a tight, leak-proof casting is needed, No. 43 alloy is recommended.

Alloys Nos. 195–T6, 220–T4, and 356–T6 are heat treated to obtain the maximum mechanical properties. They are used to replace malleable iron castings and, in some cases, steel castings. The foundry should be consulted in the selection of the alloy best suited to the conditions under consideration.

# NICKEL AND ITS NON-FERROUS ALLOYS

By Robert Worthington

## 1. PROPERTIES OF NICKEL

**GENERAL PROPERTIES.**—Nickel is a white, malleable, ductile, non-corrodible metal, capable of being rolled into thin sheet or drawn into fine wire. To its alloys it imparts strength, combined with ductility, a white color and resistance to oxidation and corrosion. The commercial grades are shown in Table 1. General properties are given in Table 2, and mechanical properties in Table 3.

The principal non-ferrous alloys of nickel are as follows:

**Nickel-copper.**—Nickel content up to 70%; used for corrosion resistance, electrical resistance and where strength, workability and appearance are desired.

**Nickel-copper-zinc.**—Nickel content, 5 to 30%; used for corrosion resistance and where appearance and strength are desired.

### Table 1.—Commercial Grades of Nickel Produced in the United States and Canada

| Grades | Typical Composition | | | | | | | Uses |
|---|---|---|---|---|---|---|---|---|
| | Ni + Co | Fe | Cu | C | Si | Mn | S | |
| Malleable | 99.00+ | 0.30 | 0.25 | 0.10 | 0.10 | 0.25 | 0.010 | Rolling, forging, drawing. |
| Malleable, 5% manganese | 94.00–95.50 | 0.40 | 0.25 | 0.15 | 0.10 | 4.25–5.25 | 0.010 | Spark-plug wire, high temperature applications, etc. |
| Malleable, 2% manganese | 97.00–98.00 | 0.40 | 0.25 | 0.15 | 0.10 | 1.75–2.25 | 0.010 | Electrical resistance wire; spark-plug wire. |
| Anodes; rolled and cast | 99.00+ | ...... | ...... | ...... | ...... | ...... | ...... | Electroplating. |
| Electrolytic, cathodes 27 by 36. in. and smaller cuttings therefrom | 99.95 | 0.01–0.04 | 0.01–0.03 | Trace | Trace | ...... | Trace | Highest-grade material for alloying in steel and non-ferrous metals in electric, open-hearth, or crucible furnaces. |
| Ingot (10, 25, 50 lb.) made by remelting electrolytic cathodes | 99.55 | 0.15 | 0.15 | 0.06 | 0.08 | ...... | 0.010 | For alloying of ferrous metals, particularly where this form is more adapted to the process involved. |
| Shot, made by remelting electrolytic cathodes | 99.55 | 0.15 | 0.15 | 0.06 | 0.08 | ...... | 0.010 | Principally used for crucible charges in ferrous and non-ferrous alloys involving small amounts of nickel. |
| "F" shot and ingot (5 lb.) low melting point | 92.00 | 2.00 | 0.25 | 0.25 | 5.00–6.00 | ...... | 0.05 | For cupola and ladle additions of nickel to gray iron. |

### Table 2.—Properties of Nickel

| | | | |
|---|---|---|---|
| Specific gravity | 8.85 | Pouring temperature, deg. F. | 2895–3000 |
| Density, lb. per cu. in. | 0.319 | Forging temperature, deg. F. | 1600–2300 |
| Melting point, deg. F. | 2624 | Annealing temp., open anneal, deg. F. | 1650–1740 |
| Specific heat, mean, 68°–2624° F. | 0.134 | " " box " deg. F. | 1400 |
| Coefficient of expansion per deg. F. | | Pattern shrinkage, in. per ft. | 1/4 |
| 77°–212° F. | 0.000 00733 | Electrical resistivity at 20° C., ohms per mil.-ft. | 64 |
| 77°–572° F. | 0.000 00800 | | |
| 77°–1112° F. | 0.000 00861 | Temperature coefficient of resistivity at 20–100° C. | 0.00537 |
| Thermal conductivity, mean, 32–212° F., B.t.u. per sq. ft. per hr. per deg. F. per in. thickness | 406.4 | Young's modulus | 30,000,000 |
| | | Modulus in torsion | 10,000,000 |

### Table 3.—Mechanical Properties of Nickel

| | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | Elastic Limit, 1000 lb. per sq. in. | Elongation, per cent in 2 in. | Reduction in area, per cent | Brinell Hardness | | Shore Scleroscope Hardness | Rockwell B Hardness | Izod Impact, ft.-lb. |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | 500 kg. | 3000 kg. | | | |
| Hot-rolled; forged | 70– 85 | 30– 40 | 20–30 | 40–50 | 55–65 | 105–125 | 125–140 | 15–19 | 60– 70 | 95–100 |
| Cold-rolled & cold-drawn | | | | | | | | | | |
| Annealed* | 70– 85 | 20– 30 | 17–23 | 43–53 | 65–75 | 85–105 | 115–130 | 11–14 | 55– 65 | 80– 85 |
| Sheet, full hard | 95–110 | 90–100 | ...... | 2–8 | ...... | ...... | ...... | 35–40 | 95–100 | ...... |
| Strip, " " | 95–115 | 90–105 | ...... | 2–5 | ...... | ...... | ...... | 35–40 | 95–100 | ...... |
| Rod, normalized | 80– 95 | 70– 80 | 40–50 | 25–35 | 40–60 | 215–240 | 200–240 | 35–45 | 90–100 | ...... |
| Tubing, as drawn | 100–105 | 85– 95 | ...... | 15–20 | ...... | ...... | ...... | 30–35 | 85– 90 | ...... |
| Cast | 60– 70 | 20– 30 | ...... | 15–35 | 30–50 | 75– 95 | 80–100 | ...... | 51– 59 | ...... |

* Endurance limit 26,000–30,000 lb. per sq. in.

**Nickel-chromium.**—Nickel content, over 80%; used for electrical and oxidation resistance.

**MALLEABLE NICKEL** is ductile and can be fabricated by drawing, stamping, spinning and machining. Cold working improves its strength, as does forging in the lower temperature ranges where the stresses of working are not relieved, *i.e.*, in the temperature range of 1000–1200° F. The strength of malleable nickel, as regularly supplied, ranges from 60,000–115,000 lb. per sq. in., although the strength of spring wire will be as high as 150,000 lb. per sq. in. with correspondingly low ductility. Malleable nickel is made into plate, sheet, strip, bar, rod, wire, tubing, forgings and castings.

**Uses.**—Malleable nickel is used in the construction of caustic evaporators and food processing and chemical equipment, because of its high corrosion resistance. It is used in automotive spark-plug wire and furnace equipment because of its resistance to high-

Table 4.—Physical Properties of Nickel-copper Alloys

| Property | Percent Nickel | | | |
|---|---|---|---|---|
| | 20 | 30 | 45 | 70 |
| Specific gravity | 8.96 | 8.93 | 8.90 | 8.80 |
| Density, lb. per cu. in. | 0.324 | 0.323 | 0.322 | 0.318 |
| Melting point, deg. F. | 2120 | 2165 | 2210 | 2460 |
| Thermal expansion per deg. F., at 95° F. | 0.000 008 67 | 0.000 008 44 | 0.000 008 28 | 0.000 007 78 |
| Thermal conductivity, B.t.u. per sq. ft. per hr. per deg. F. per 1 in. thickness | 238 | 206 | 160 | 116 |
| Electrical resistivity, ohms per mil. ft. | 164 | 242 | 292 | 268 |
| Coefficient of resistivity per deg. C., at 20–100° C. | ............ | 0.0001 | 0.0000 | 0.0019 |
| Young's modulus | 20,000,000 | ............ | ............ | 26,000,000 |

Table 5.—Mechanical Properties of Annealed Copper-nickel Alloys
(Typical values)

| Nickel Content, percent | Tensile Strength, lb. per sq. in. | Proportional Limit, lb. per sq. in. | Elongation in 2 in., percent | Reduction in Area, percent | Brinell Hardness, No. |
|---|---|---|---|---|---|
| 2 | 34,000 | ....... | 50 | .. | .. |
| 15 | 46,000 | 11,900 | 47 | 83 | 65 |
| 20 | 48,000 | 13,800 | 46 | 79 | 72 |
| 40 | 59,000 | 11,900 | 44 | 71 | 88 |
| 70 | 75,000 | 25,000 | 42 | 70 | 120 |

temperature oxidation. It is used in coinage because of its toughness and appearance. It also is used in vacuum tubes, because of its high melting point, purity and ease of fabrication.

**Nickel-clad Steel** is used for tanks, tank cars, and heavy plate construction. It consists of mild steel with 10 to 20% nickel bonded to one side of it. The bond is permanent under pressure, vacuum, temperature change and deformation in forming.

## 2. ALLOYS OF NICKEL AND COPPER

Copper alloys with nickel in all proportions to form solid solutions. Strength and toughness increase with increase of nickel content up to 70–80% Ni. The alloys of high copper content are relatively soft, highly ductile and easily fabricated. Alloys are pink in color up to about 25% Ni. Above 25% Ni, the alloys are white. Nickel-copper alloys are malleable both hot and cold. Table 4 gives typical properties of nickel-copper alloys.

Cold working increases the strength and hardness of nickel-copper alloys, the effect being greater on alloys of high Ni content than on low Ni alloys. Thus, the tensile strength of spring temper Monel metal is about 150,000 lb. per sq. in., and that of 80–20 cupro-nickel is about 85,000 lb. per sq. in. Typical values of the tensile properties of nickel-copper alloys are given in Table 5.

**USES.**—Nickel-copper alloys are made for a wide variety of uses under different trade names. The following are the more important of these alloys, together with their trade names: 65–75% Ni, Monel metal, Luceto; general purpose alloys. 40–45% Ni, Constantan, Advance, Copel, Excelsior, Cupron, Ferry; resistance wire, pyrometer wire. 30% Ni, Everbrite, Davis Metal; valve castings, etc. 20–30% Ni, Cupro-nickel, Ambrac (5% Zn), Adnic; condenser tubes. 25% Ni; coinage. 20% Ni, Cupro-nickel; turbine blading. 15–20% Ni, Cupro-nickel; bullet jackets. 2–4% Ni; heat exchanger tubes.

### Monel Metal*

The composition of Monel metal is as follows: Ni, 65–70%; Cu, 26–30%; Fe, up to 3%; Mn, up to 1.5%; Si, up to 0.25%; C, up to 0.25%. In spring wire, the Si content may range up to 1.25%, or Mn may be as high as 2.50%. High-strength castings may have a Cu content up to 34% and Si up to 4%. Monel metal is white, malleable and ductile, both hot and cold, and corrosion-resistant. Its appearance, physical and mechanical properties resemble closely those of malleable nickel. It is fabricated into cold-drawn, hot-rolled and forged rod, bars and shapes, cold-drawn wire, hot-rolled plate, cold-rolled sheet and strip, seamless tubing, and castings. Table 6 gives typical properties of Monel metal. Table 7 and 8 show the tensile properties at normal and elevated temperatures.

**"K" MONEL METAL** is a special grade of Monel metal, containing about 3.75% aluminum, which is hardened and made stronger both by heat treatment and cold working. Quenching in water from 1400–1500° F. softens it. Slow furnace cooling from 1000–1100° F., with or without previous cold working, hardens it. See Table 9 for properties.

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* Trademark of International Nickel Co.

USES.—Monel metal is used for interior architectural decoration, trim on household appliances and sheet metal work in kitchens, restaurants, etc., and other places where appearance is of importance. Its high corrosion resistance makes it useful in the handling of acids, alkalis, salt solutions, waters, food products, organic substances and atmospheres at normal and elevated temperatures. It also is used in the construction of pickling equipment, tanks, pumps, valves and agitators used in connection with acids, alkalis, brines and food products. Its strength and toughness causes it to find application to

### Table 6.—Physical Properties of Monel Metal

| | | | |
|---|---|---|---|
| Specific gravity | 8.80 | Forging temperature, deg. F. | 1830–2152 |
| Density, lb. per cu. in. | 0.318 | Annealing temperature, deg. F., open anneal | 1700–1795 |
| Melting point, deg. F. | 2460 | Annealing temp., deg. F., box anneal | 1380 |
| Specific heat | 0.128 | Electrical resistivity at 20° C., ohms per mil-ft. | 268 |
| Coefficient of expansion, per deg. F. | | Temperature coefficient of resistivity per deg. C., 20–100° C. | 0.00196 |
| 77–212° F. | 0.000 007 78 | Magnetic transformation, deg. C. | 95 |
| 77–572° F. | 0.000 008 33 | Young's modulus | 26,000,000 |
| 77–1112° F. | 0.000 009 06 | Modulus in torsion | 9,000,000 |
| Thermal conductivity, B.t.u. per sq. ft. per hr. per deg. F. per 1 in. thickness | 116 | | |
| Pouring temperature, deg. F. | 2800–2850 | | |

### Table 7.—Mechanical Properties of Monel Metal

| Product | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Proportional Limit, lb. per sq. in. | Elongation in 2 in., percent | Reduction in Area, percent |
|---|---|---|---|---|---|
| Rod and Bar: | | | | | |
| Cold-drawn, annealed | 70,000– 85,000 | 25,000– 35,000 | 20,000–30,000 | 35–50 | 65–75 |
| " as drawn | 85,000–125,000 | 60,000– 95,000 | ............ | 15–35 | 50–65 |
| Hot-rolled | 80,000– 95,000 | 40,000– 65,000 | 25,000–40,000 | 30–45 | 50–65 |
| Forged | 80,000–105,000 | 60,000– 85,000 | 45,000–65,000 | 20–40 | ..... |
| Wire: | | | | | |
| Cold-drawn, annealed | 70,000– 85,000 | ............ | ............ | ..... | ..... |
| " No. 1 temper | 95,000–110,000 | ............ | ............ | ..... | ..... |
| " regular | 110,000–140,000 | ............ | ............ | ..... | ..... |
| " spring | 140,000–175,000 | ............ | ............ | ..... | ..... |
| Plate, hot-rolled | 60,000– 75,000 | 25,000– 30,000 | ............ | 25–35 | ..... |
| Sheet and Strip: | | | | | |
| Full-finished sheet | 65,000– 80,000 | 25,000– 35,000 | 20,000–30,000 | ..... | ..... |
| Cold-rolled, annealed | 65,000– 80,000 | 25,000– 35,000 | 20,000–30,000 | ..... | ..... |
| " full-hard sheet | 100,000–120,000 | 90,000–110,000 | ............ | ..... | ..... |
| " full-hard strip | 100,000–125,000 | 90,000–115,000 | ............ | ..... | ..... |
| Tubing, cold-drawn | | | | | |
| Annealed | 65,000– 80,000 | 25,000– 35,000 | 20,000–30,000 | ..... | ..... |
| As drawn | 90,000–105,000 | 60,000– 75,000 | ............ | 15–25 | ..... |
| Castings | 65,000–100,000 | 30,000– 60,000 | ............ | 5–35 | 5–35 |

### Table 8.—Effect of Temperature on Tensile Properties of Hot-rolled Monel Metal

| Temperature, deg. F. | Room | 600 | 800 | 1000 | 1200 | 1400 | 1600 | 1800 | 2000 |
|---|---|---|---|---|---|---|---|---|---|
| Tensile strength, lb. per sq. in. | 85,000 | 80,000 | 70,000 | 55,000 | 36,000 | 22,000 | 12,000 | 7500 | 5000 |
| Limiting creep strength, lb. per sq. in. | ...... | ...... | 45,000 | 15,000 | 2,500 | ...... | ...... | .... | .... |
| Yield point, lb. per sq. in. | 45,000 | 34,000 | 30,000 | 27,000 | 23,000 | ...... | ...... | .... | .... |
| Elongation in 2 in., percent | 45 | 44 | 40 | 30 | 18 | 22 | 31 | 42 | 55 |

### Table 9.—Mechanical Properties of "K" Monel Metal

| Grade | Tensile Strength, 1000 lb. per sq. in. | Yield Point, 1000 lb. per sq. in. | Proportional Limit, 1000 lb. per sq. in. | Elongation in 2 in. percent | Reduction of Area, percent | Brinell Hardness (3000 kg. load) |
|---|---|---|---|---|---|---|
| Soft | to 120 | to 80 | to 60 | 40+ | 50+ | to 215 |
| Intermediate hardness | 120–160 | 80–120 | 60–100 | 20+ | 25+ | 215–315 |
| Hard | 160+ | 120+ | 100+ | 15+ | 20+ | over 315 |

turbine blading, valves, and other steam handling equipment, marine propeller shafting, springs, bolts on railway signal and electrical systems.

**WORKING QUALITIES OF MONEL METAL.—Hot-working.**—Monel metal may be hot worked in the temperature range 1900–2100° F. With care, the temperature can be extended to 2150° F., but Monel metal loses all malleability at about 2200° F. Hot forging should not be attempted between 1200–1600° F., and only under special conditions between 1600–1900° F. Sulphur-free fuel should be used in the heating furnaces, and a definitely reducing atmosphere is desirable.

**Cold-working** operations, as cupping, drawing, swaging, die-forging, power-hammering, bending and forming, are similar to those performed on mild steel, except that more power is required. Annealing may be either by box or open annealing methods. Sulphur-bearing atmospheres must be avoided, and a reducing atmosphere is desirable.

**Machining.**—Cutting tools for Monel metal should be ground to a 15-deg. to 20-deg. back rake. They should be of high-speed steel. For thread chasing, self-opening die-heads are necessary. For machining operations, sulphur base oil, cut with a paraffin base oil, is better than straight paraffin base oil.

**Welding** may be performed by the various electric and gas methods. A special deoxidizing flux gives the best results in arc welding. In gas welding, the flame must be close to neutral, and on the reducing side. The work must be performed rapidly without rewelding, using fused boric acid as a flux.

**Soldering** may be done with soft solders, either high or low in tin. Lead fillets are used for special purposes. For strong joints, subject to corrosive substances, brazing and silver solders are used.

**Pickling.**—In commercial pickling, the oxide first is reduced by heating and cooling in a reducing atmosphere. The metal then is submerged in a solution of 6 lb. sodium nitrate, 3 lb. sodium chloride, 1 gal. concentrated sulphuric acid per 40 gal. of water, held at 180–200° F. If the reducing treatment cannot be used, the metal is dipped in a hot, moderately strong solution of hydrochloric acid and scrubbed with abrasive.

### Other Nickel-Copper Alloys

**Nickel 40–45 Percent.**—These alloys have a low temperature coefficient of electrical resistance combined with relatively high resistivity (see Table 14). They are used for rheostats and other resistances operating at black heat, and also for low temperature heaters. A high thermoelectric force is developed between these alloys and either iron, copper or nickel-chromium, which permits the use of these combinations in thermocouples for pyrometric work.

**Nickel 20–30 Percent.**—Cupro-nickel tubing made of copper-nickel alloys of 20–30% Ni, either with or without zinc, is used for the more severe air impingement conditions of marine and stationary condensers handling sea water, or corrosive inland waters, at high velocity.

**Nickel 15–20 Percent,** known as high ductility cupro-nickel, may be subjected to excessive cold work in drawing and forming, without intermediate anneal. These alloys are used as bullet envelopes and for deep drawing.

## 3. ALLOYS OF COPPER, NICKEL, AND ZINC

Alloys of copper, nickel and zinc, known commercially as Nickel Silver or German Silver usually are designated by the Nickel content, *e.g.*, 18% nickel-silver. Nickel-silver ranges from about 10–25% Ni and 3–40% Zn. These ranges cover the common casting and cold-working alloys. Alloys containing up to 10% nickel, with high zinc, regularly are extruded. Some alloys of 10–15% Ni are extruded in the simpler shapes. Alloys containing up to 10% Zn can be hot worked, but not hot extruded. Table 11 shows the composition of wrought and cast nickel-silver compositions in commercial use. Table 10 gives the physical properties and Tables 12 and 13 the mechanical properties of nickel-silver.

**Table 10.—Physical Properties of Nickel-silver Alloys**

| Composition | | Specific Gravity | Lb. per cu. in. | Melting Point, deg. F. | Thermal Expansion per deg. F. | Composition | | Electrical Resistivity Ohms per mil.-ft. |
|---|---|---|---|---|---|---|---|---|
| Ni | Zn | | | | | Ni | Zn | |
| 30 | 23 | 8.74 | 0.316 | ........ | .......... | 28.6 | 9.8 | 250 |
| 20 | 5 | 8.84 | .319 | 2100 | 0.000 009 06 | 17.4 | 26.7 | 166 |
| 18 | 27 | 8.68 | .314 | 1920 | .......... | 19.8 | 18.5 | 178 |
| 18 | 17 | 8.75 | .316 | ........ | .......... | 15.75 | 29.7 | 167 |
| ...... | ...... | ........ | ...... | ........ | .......... | 7.80 | 31.8 | 111 |

Cold working increases the strength of nickel-silver alloys. For example, spring temper sheet and strip, of 18% Ni, 27% Zn, has a tensile strength of about 120,000 lb. per sq. in. Spring temper wire of the same composition has a tensile strength of about 140,000 lb. per sq. in.

## 4. ALLOYS OF NICKEL AND CHROMIUM. RESISTANCE ALLOYS

Alloys of nickel with 20% or less of chromium have both high electrical resistivity and good resistance to temperatures up to 2150° F. This makes them highly suitable for use as elements in electrical resistance heating devices. Data on these alloys are given in Table 14. Nickel-chromium alloys, particularly those containing iron, are extensively used in high temperature service other than electrical resistance service, and in handling corrosive substances. For data concerning the Ni-Cr-Fe heat-resisting alloys containing upwards of 25% Fe, and the Ni-Cr-Fe austenitic stainless steels containing 70–74% Fe, see pp. 3–19 to 3–24.

**Table 11.—Composition of Wrought and Cast Nickel-silver Alloys**

| | Composition, percent | | | | | | |
|---|---|---|---|---|---|---|---|
| | Ni | Cu | Zn | Pb | Fe | Sn | Al |
| | Wrought Alloys | | | | | | |
| Cutlery and knife stock.. | 15–25 | 55–65 | 14–20 | ......... | 0.5–1.5 | ......... | ......... |
| Key stock............. | 8–18 | 55–65 | 15–35 | 1–2 | ......... | ......... | ......... |
| Jeweler's wire........... | 5–25 | 53–63 | 25–32 | ......... | ......... | ......... | ......... |
| Brazing solder.......... | 8–20 | 35–40 | 40–55 | ......... | ......... | ......... | ......... |
| Watchcase metal........ | 10–28 | 55–65 | 16–30 | 0–1 | ......... | ......... | ......... |
| Spoon and fork stock.... | 10–20 | 57–66 | 20–30 | ......... | ......... | ......... | ......... |
| Platers' bars and cores... | 5–25 | 56–70 | 18–24 | ......... | ......... | ......... | ......... |
| | Cast Alloys | | | | | | |
| No. 1 *............. | 20 | 62 | 18 | ......... | ......... | ......... | ......... |
| No. 2 †............. | 20 | 65 | 6 | 5 | ......... | 4 | ......... |
| No. 3 †............. | 20 | 52 | 15 | 10 | ......... | 3 | ......... |
| No. 4 ‡............. | 20 | 65 | 10 | ......... | ......... | 0.5 | 2.5 |

* A strong, tough, ductile alloy, commonly used for marine hardware.

† Less ductile but more easily machinable than No. 1. Makes watertight castings with greater facility. No. 2 is used considerably for marine and builders' hardware and is specified by the U. S. Government for certain uses. No. 3 is softer and used for plumbing goods.

‡ Used for light castings of uniform section where perfection of surface is desired; not suitable for pressure work.

**Table 12.—Mechanical Properties of Annealed Wrought Nickel-silvers**

| Composition, percent | | | Tensile Strength, lb. per sq. in. | Elongation in 2 in., percent | Composition, percent | | | Tensile Strength, lb. per sq. in. | Elongation in 2 in., percent |
|---|---|---|---|---|---|---|---|---|---|
| Ni | Zn | Cu | | | Ni | Zn | Cu | | |
| 10.6 | 30.1 | 59.3 | 50,600 | 67 | 18.4 | 26.3 | 55.2 | 66,100 | 45 |
| 15.9 | 24.4 | 59.7 | 57,200 | 47 | 17.6 | 10.7 | 71.3 | 54,500 | 38 |
| 19.3 | 20.5 | 60.2 | 61,000 | 41 | 6.7 | 28.6 | 64.7 | 49,400 | 63 |
| 17.9 | 17.8 | 64.0 | 59,000 | 36 | 17.8 | 16.8 | 65.4 | 57,700 | 37 |
| 18.0 | 26.2 | 55.6 | 66,100 | 46 | 20 | 5 | 75 | 50,000 | 30 |

**Table 13.—Mechanical Properties of Nickel-silver Castings**

| Type | Composition, percent | | | | | | Tensile Strength, lb. per sq. in. | Elongation in 2 in., percent |
|---|---|---|---|---|---|---|---|---|
| | Ni | Zn | Cu | Pb | Sn | Al | | |
| Green sand | 19.9 | 5.3 | 64.2 | 5.9 | 4.3 | ......... | 41,200 | 8.5 |
| | 20.0 | 13.7 | 64.7 | ......... | ......... | ......... | 41,600 | 43 |
| | 26.6 | 15.4 | 56.9 | 0.2 | ......... | ......... | 40,900 | 34 |
| Dry sand | 7.8 | 31.8 | 60.2 | ......... | ......... | ......... | 33,800 | 47 |
| | 15.8 | 29.7 | 54.3 | ......... | ......... | ......... | 37,300 | 39 |
| | 19.8 | 18.5 | 61.6 | ......... | ......... | ......... | 42,800 | 33 |
| | 26.7 | 13.2 | 60.1 | ......... | ......... | ......... | 53,200 | 35 |
| Chill cast | 19.5 | 31.1 | 49.4 | ......... | ......... | 0 | 50,100 | 31 |
| | 18.0 | 32.7 | 48.2 | ......... | ......... | 1.1 | 75,300 | 4 |
| Die-cast | 18.5 | 32.5 | 48.0 | ......... | ......... | 1.0 | 75,000 | 5 |

**Table 14.—Properties of Resistance Alloys**

| Composition, percent | Specific Gravity | Melting Point, deg. F. | Max. Working Temp., deg. F. | Thermal Expansion, per deg. F.* | Resistivity at 20° C. (68° F.) Ohms per mil.-ft. | Relative Resistivity at Elevated Temperatures Resistivity at 20° C. = 1.000 | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Temperature, deg. C. | | | | | | | | | |
| | | | | | | 100 | 200 | 300 | 400 | 500 | 600 | 700 | 800 | 900 | 1000 |
| Ni 80<br>Cr 20 | 8.4 | 2550 | 2010 | $0.0_5899$† | 650 | 1.014 | 1.028 | 1.044 | 1.057 | 1.066 | 1.067 | 1.063 | 1.060 | 1.060 | 1.064 |
| Ni 61<br>Cr 16<br>Fe 23 | 8.2 | 2460 | 1920 | $.0_5899$† | 675 | 1.013 | 1.029 | 1.045 | 1.059 | 1.070 | 1.078 | 1.082 | 1.088 | 1.127 | 1.142 |
| Ni 100 | 8.85 | 2640 | 1470 | $.0_5855$‡ | 64 | 1.43 | 2.06 | 2.83 | 3.52 | 3.87 | 4.15 | 4.44 | 4.73 | 5.04 | 5.38 |
| Ni+<br>Mn 2 | 8.8–8.9 | 2615 | 1470 | $.0_5750$§ | 84 | 1.36 | 1.90 | 2.58 | 3.02 | 3.26 | 3.49 | 3.73 | 3.96 | 4.22 | 4.50 |
| Ni+<br>Mn 5 | 8.8–8.9 | 2615 | 1470 | $.0_5750$§ | 117 | 1.29 | 1.68 | 2.10 | 2.31 | 2.46 | 2.62 | 2.78 | 2.94 | 3.13 | 3.32 |
| Ni 68<br>Cu 29 | 8.8 | 2460 | 1200 | $.0_5894$‡ | 268 | 1.157 | 1.217 | 1.254 | 1.294 | 1.339 | 1.384 | 1.434 | 1.488 | 1.550 | 1.626 |
| Ni 44<br>Cu 56 | 8.9 | 2210 | 930 | $.0_5799$† | 290 | 0.9992 | 0.9995 | 1.002 | 1.006 | 1.020 | 1.039 | ..... | ..... | ..... | ..... |
| Ni 30<br>Cr 2<br>Fe 68 | 8.25 | 2695 | 1110 | $.0_5950$§ | 530 | 1.093 | 1.198 | 1.292 | 1.365 | 1.425 | 1.472 | 1.504 | 1.533 | 1.585 | ..... |

* The subscript indicates the number of zeros after the decimal point; thus $0.0_5899 = 0.00000899$.
† Mean from 70°–1598° F. ‡Mean from 77°–1112° F. §Mean from 32°–212° F.

# MAGNESIUM AND ITS ALLOYS

By John A. Gann

## 1. PROPERTIES

**PROPERTIES.**—Magnesium is a silver-white metal, produced commercially by the electrolysis of molten anhydrous magnesium chloride, with an average magnesium content of 99.94%. A special grade with a magnesium content of 99.99% may be made by sublimation of commercial magnesium. The properties of magnesium are as follows: Atomic weight, 24.32; density, lb. per cu. in., at 68° F., 0.0629, at melting point (solid), 0.0592, at melting point (liquid), 0.0567; boiling point, 2006.6° F.; mean coefficient of thermal expansion, 68° to 572° F., 0.0000157; 68° to 932° F., 0.0000166 per deg. F.; freezing point, 1204° F.; thermal conductivity, 1102 B.t.u. per hr. per sq. ft. per deg. F. per 1 in. thickness; heat of vaporization, 2340–2700 B.t.u. per lb.; latent heat of fusion, 126 B.t.u. per lb.; mean specific heat, 32 to 212° F., 0.249. The mechanical properties are: tensile strength, 27,000 lb. per sq. in.; yield strength (set = 0.2%), 10,000 lb. per sq. in.; elongation in 2 in., 15%; Brinell hardness, 10-mm. ball, 500-kg. load, 37; modulus of elasticity, 6,250,000. The electrical resistivity at 20° C. is 4.4611 microhms per cu. cm., or 26.83 ohms per mil-ft. The electrical conductivity at 20° C. in percent of International Annealed Copper Standard is, mass, 197.7, volume, 38.6; the temperature coefficient of resistivity at 20° C. is 0.0040. The crystal form of magnesium is close packed hexagonal. Its reflectivity for white light is 73%.

**USES.**—The chemical uses of magnesium are based on its chemical reactivity in a finely divided form or at high temperature, and include pyrotechnics, flares, desulphurization of nickel and nickel alloys. Used as an alloying metal, it improves the physical properties of aluminum and zinc die-casting alloys. Magnesium is the basis of the lightest structural alloys, both in the form of castings and shapes. See Magnesium Alloys below.

## 2. MAGNESIUM ALLOYS

**PROPERTIES.**—Commercial alloys have a magnesium content of from 88–99%. The composition, trade names and uses of such alloys are given in Table 1. Lightness is the principal characteristic of these alloys. Taking the weight of magnesium alloys as 1 the relative weights of other alloys are: aluminum, 1.5; cast iron and steel, 4; brass and bronze, 5. On an equal weight basis, the strength of magnesium alloys is equal to or superior to that of most metals. Strength decreases with increase in temperature, and in general, magnesium alloys are not suited for use at temperatures over 400° F. The characteristics of magnesium alloys are high tensile strength, toughness, and good fatigue endurance. Table 2 shows typical physical and mechanical properties of magnesium alloys and Table 3 compares mechanical properties of various structural alloys.

**CORROSION.**—Magnesium alloys are stable under ordinary exposure conditions, the surface gradually darkening with no decrease in strength. In heavy industrial areas, or in humid atmospheres along the seacoast, surface roughening may occur, and some protection may be desirable. Aqueous salt and acid solutions corrode magnesium alloys, with the exception of pure chromic acid, pure concentrated hydrofluoric acid, and alkali metal fluorides, chromates, and bichromates. The alloys are resistant to attack by most alkalies and many organic chemicals, including hydrocarbons, phenols, and oils.

**Table 1.—Trade Designations, Analyses, and Uses of Magnesium Alloys**

| Designation and Trade Names | | | | | | Composition, percent | | | | | Uses |
|---|---|---|---|---|---|---|---|---|---|---|---|
| A.S.T.M. Alloy No. | U. S Army Designation | U S. Navy Designation | Dow Chem. Co., Dowmetal | Amer. Magnesium Corp. | Bohn Aluminum and Brass Corp., Bohnlite | Al | Mn | Zn | Sn | Mg | |
| 1 * | 57–741b Grade 1 | M-112c Alloy 1 | A | AM-241 | X-5 | 8.0 | 0.2 | ..... | ..... | 91.8 | Sand castings. Heat treatment not required for general use. Heat-treated for highly stressed parts requiring good impact toughness. |
| 2 * | 57–741b Grade 2 | M-112c Alloy 2 | G | AM-240 | X-2 | 10.0 | 0.1 | ..... | ..... | 89.9 | Die castings. Heat-treated sand castings requiring high yield strength with moderate toughness. |
| 3 * | ........ | ........ | B | AM-246 | X-7 | 12.0 | 0.1 | ..... | ..... | 87.9 | Heat-treated sand castings requiring maximum yield strength and hardness. Impact toughness relatively low. |
| 4 * | ........ | M-112c Alloy 5 | H | AM-265 | X-8 | 6.0 | 0.2 | 3.0 | ..... | 90.8 | Sand castings with improved salt-water corrosion resistance. Heat-treated castings combine high yield and ultimate strength with good impact toughness. |
| 6†, ‡, § | ........ | M-111b Alloy 1 | F | AM-53S | X-1s | 4.0 | 0.3 | ..... | ..... | 95.7 | Forgings, sheet and plate, extruded shapes, bars, rods, and tubes. |
| 8 ‡, § | ........ | M-126a Grade 2 | J | AM-57S | ........ | 6.5 | 0.3 | 0.75 | ..... | 92.45 | Forgings and extruded bars, rods and shapes. |
| 9 ‡, § | ........ | M-126a Grade 3 | ...... | AM-58S | ........ | 8.5 | 0.2 | 0.5 | ..... | 90.8 | Hot-pressed forgings for highly stressed parts. More difficult to forge than other alloys. |
| 10 ‡ | ....... | M-126a Grade 1 | ...... | AM-61S | X-11s | ..... | 1.0 | ..... | 6.0 | 93.0 | Extruded rods and shapes. Hammer forgings for moderately stressed parts. |
| 11 †, § | ........ | M-111b Alloy 2 | M | AM-3S | X-6 | ..... | 1.5 | ..... | ..... | 98.5 | Sheet, plate, and rod for moderately stressed parts requiring resistance to salt water. |

* Tentative specifications for magnesium-base alloy sand castings (B80–36T).

† Tentative specifications for magnesium-base alloy sheet (B90–36T).

‡ Tentative specifications for magnesium-base alloy forgings (B91–36T).

§ Tentative specifications for magnesium-base alloy bars, rods, and shapes.

**Protection.**—Paints, varnishes, lacquers and enamels can be applied to magnesium alloys. Synthetic resin varnishes and enamels are the best coatings for protection and high-grade finish. Baked finishes have excellent adhesion, hardness and toughness of film and can be applied rapidly.

Before painting, the metal should be cleaned and then chrome-pickled by a short-time dip in a bath of 1.5 lb. $Na_2Cr_2O_7$ and 1.5 pt. of concentrated nitric acid per gallon of solution. A thorough wash and rapid drying should follow the dip. This treatment produces a dull, iridescent bronze finish, combining a mechanical "tooth" and a chemically inhibitive film which improves adhesion and the protective value of the paint coating.

**Table 2.—Typical Physical and Mechanical Properties of Magnesium Alloys**

Modulus of elasticity = 6,500,000

| A.S.T.M. Designation | Condition * | Treatment † | Strength, 1000 lb. per sq. in. | | | | Elongation, percent in 2 in. | Hardness | | Charpy Impact, ft.-lb. | Endurance Limit, 1000 lb. per sq. in. ‖ | Thermal Conductivity, c.g.s. units, 100 to 300° C. | Electrical Resistivity, microhm-cm. at 20° C. | Specific Gravity | Weight, lb. per cu. in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Tensile ‡ | Yield (set = 0.2 percent) § | Compressive | Shearing | | Brinell ¶ | Rockwell E Scale | | | | | | |
| 1 | SC | ........ | 25 | 11 | 46 | 17 | 4 | 48 | 55 | 0.9 | 7 | 0.18 | 13.0 | 1.80 | 0.065 |
| | SC | H.T. 1 | 33 | 11 | 46 | 17 | 9 | 48 | 55 | 2.2 | 7.5 | ...... | 15.0 | 1.80 | 0.065 |
| 2 | DC | ........ | 29 | 20 | .... | .... | 1 | 63 | 75 | ...... | ...... | 0.17 | 15.0 | 1.81 | 0.066 |
| | SC | H.T. 1 | 33 | 12 | 51 | 20 | 7 | 51 | 61 | 2.0 | 11.5 | ...... | 17.5 | 1.81 | 0.066 |
| | SC | H.T. 2 | 33 | 16 | 51 | 21 | 3 | 60 | 72 | 1.0 | 10. | ...... | 16.0 | 1.81 | 0.066 |
| | SC | H.T. 3 | 33 | 19 | 55 | 22 | 1 | 69 | 80 | 0.7 | 10.5 | ...... | 14.0 | 1.81 | 0.066 |
| 3 | SC | H.T. 3 | 29 | 21 | .... | .... | 0.5 | 75 | 84 | ...... | 9. | 0.16 | 14.0 | 1.82 | 0.066 |
| 4 | SC | ........ | 28 | 11 | 46 | 17 | 5 | 48 | 55 | 1.0 | 10. | ...... | ...... | 1.84 | 0.066 |
| | SC | H.T. 1 | 35 | 12 | 46 | 17 | 10 | 51 | 61 | 2.0 | 10. | ...... | ...... | 1.84 | 0.066 |
| | SC | H.T. 3 | 40 | 20 | 50 | 18 | 5 | 68 | 79 | 1.0 | 10. | ...... | ...... | 1.84 | 0.066 |
| 6 | Ex** | ........ | 40 | 26 | 58 | 20 | 15 | 48 | 55 | 2.2 | 13 | 0.23 | 10.0 | 1.77 | 0.064 |
| | F | ........ | 38 | 23 | .... | .... | 12 | 50 | 60 | ...... | 12 | ...... | ...... | 1.77 | 0.064 |
| | Sh | ........ | 43 | 30 | .... | .... | 8 | 65 | 76 | ...... | ...... | ...... | ...... | 1.77 | 0.064 |
| | Sh | annealed | 35 | 19 | .... | .... | 14 | 55 | 66 | ...... | ...... | ...... | ...... | 1.77 | 0.064 |
| 8 | Ex** | ........ | 43 | 28 | 54 | 22 | 15 | 55 | 66 | 2.0 | 16 | ...... | ...... | 1.81 | 0.065 |
| | F | ........ | 42 | 28 | .... | .... | 9 | 59 | 71 | 2.2 | ...... | ...... | ...... | 1.81 | 0.065 |
| 9 | F | ........ | 45 | 32 | 40 | 23 | 6 | 72 | 82 | 1.4 | 16 | ...... | ...... | 1.83 | 0.066 |
| 10 | F | ........ | 35 | 19 | .... | .... | 5 | 45 | 49 | 2.0 | 9 | ...... | ...... | 1.85 | 0.067 |
| 11 | Ex** | ........ | 42 | 26 | 57 | 19 | 7 | 42 | 40 | ...... | 7 | 0.30 | 5.0 | 1.76 | 0.064 |
| | Sh | ........ | 33 | 21 | .... | .... | 6 | 50 | 60 | ...... | ...... | ...... | ...... | 1.76 | 0.064 |

* SC = sand cast. DC = die cast. Ex = extruded. F = forged. Sh = sheet.

† H.T. 1 = solution heat treatment. H.T. 2 = solution heat treatment + partial aging. H.T. 3 = solution heat treatment + complete aging.

‡ Standard A.S.T.M. tension specimens.

§ Yield strength is defined as the stress at which the stress-deformation curve deviates 0.2 percent from the modulus line.

¶ Obtained with 500-kg. load, 10-mm. ball.

‖ R. R. Moore type rotating beam machine, 500 million cycles.

** Properties of extruded material for round and square up to 1 1/2 in.

**Table 3.—Comparative Mechanical Properties of Structural Material**

| | Specific Gravity | Tensile Strength, lb. per sq. in. | Yield Strength, lb. per sq. in. | Endurance Limit, lb. per sq. in. | Specific Tensile Strength§ | Specific Yield Strength§ | Specific Endurance Limit § |
|---|---|---|---|---|---|---|---|
| Cast: | | | | | | | |
| Dowmetal A, H.T. 1 *.. | 1.80 | 34,000 | 12,000 | 7,000 | 18,900 | 6,700 | 3,900 |
| Dowmetal H, H.T.3 †.. | 1.83 | 40,000 | 20,000 | 10,000 | 21,800 | 10,900 | 5,500 |
| Aluminum alloy S.A.E. 30........... | 2.83 | 22,000 | 14,000 | 7,500 | 7,800 | 5,000 | 2,700 |
| Aluminum alloy S.A.E. 38, H.T. ‡.... | 2.77 | 36,000 | 22,000 | 6,000 | 13,000 | 7,900 | 2,200 |
| Cast iron (Gray)....... | 7.2 | 40,000 | ......... | 20,000 | 5,600 | ......... | 2,800 |
| Wrought: | | | | | | | |
| Dowmetal F........... | 1.77 | 40,000 | 29,000 | 14,000 | 22,600 | 16,400 | 7,900 |
| Aluminum sheet 1/2 hard | 2.71 | 17,000 | 14,000 | ......... | 6,300 | 5,200 | ......... |
| Duralumin............ | 2.79 | 58,000 | 35,000 | 15,000 | 20,800 | 12,500 | 5,400 |
| Mild steel............. | 7.85 | 60,000 | 36,000 | 35,000 | 7,600 | 4,600 | 4,500 |
| Cr-Mo steel, H.T. ‡.... | 7.85 | 122,000 | 86,000 | 68,000 | 15,600 | 11,000 | 8,700 |

* H.T. 1 = solution heat treatment. † H.T. 3 = solution heat treatment + complete aging.

‡ H.T. = heat treated.

§ Specific strength values are the normal strength values divided by the specific gravity.

## 3. FABRICATION AND USE OF MAGNESIUM ALLOYS

Magnesium alloys are available as castings and wrought forms. Sheet is rolled in sizes up to 20 ft. long and 48 in. wide. Structural shapes comprise round, square and hexagonal bars, angles, I-beams, channels and special sections. They are produced by extrusion.

**CASTINGS** range in weight from a few ounces up to 700 lb. Die-castings can be made up to 3 lb., with a projected area up to 100 sq. in. Castings normally are made in green sand. Small amounts of fluoride, sulphur, boric acid or other materials should be added to the sand to inhibit the action of water vapor on the hot metal. Castings also may be made in dry sand. Pattern equipment used for aluminum will be satisfactory for magnesium alloys.

**Shrinkage Allowance** for moderate size castings, or with unrestrained shrinkage is $^3/_{16}$ in. per ft. For large castings it is $^5/_{32}$ in. per ft.

The properties of as-cast magnesium alloys are satisfactory for many industrial applications. Heat treatment, however, will greatly improve the properties of most of the alloys, and is recommended for severe service conditions.

**HOT AND COLD WORKING.**—Magnesium alloys can be extruded, rolled, forged and formed at temperatures of 400–800° F. Working speeds should be moderate and bend radii generous. The alloys work-harden at room temperatures more rapidly than other metals, and cold working operations should be limited to simple steps with small changes in shape or cross-section.

**JOINING.**—**Riveting.**—Magnesium alloy parts should be riveted with aluminum alloy rivets containing no nickel or copper. Rivet holes should be $^1/_{64}$ to $^1/_{32}$ in. larger than rivet diameter. Bearing strength of magnesium alloy sheet is approximately 50% greater than its tensile strength. Dissimilar metals in contact with magnesium alloys should be separated by bitumastic paint or reinforced red lead to inhibit galvanic corrosion.

**Welding** may be performed by the oxyacetylene, electric spot, and seam welding processes. The technique is similar to that used with aluminum. Oxyacetylene welds should be made with a filler rod, of the same approximate composition as the base metal, and a flux. The welds will be as strong as as-cast metal. Complete removal of flux after welding by washing in hot water, followed by a chrome pickle, is necessary to avoid corrosion.

Electric spot and seam welds are made on machines with water-cooled electrodes. Welds on $^1/_8$ in.-sheet have a strength of approximately 1800 lb. per spot, or 20,000 lb. per lineal foot of seam.

**MACHINING.**—Magnesium alloys may be machined at higher speeds and with greater depth of cut than any other common metal. Sharpness and proper clearance of cutting tools are essential. Tools for turning, shaping, planing and parting should have a minimum side clearance of 6 deg., with little or no top rake, and an included tool angle of 75 deg. Spiral-type milling and reaming tools are recommended. They should be ground without land and have about 6 deg. relief behind the cutting edge. Ordinary twist drills ground to standard shapes are satisfactory. Threads may be cut in the lathe or by standard four-fluted taps. The alloys finish smoothly and with no tendency to tear or drag. Cutting compounds are recommended for screw machine work and other high-speed operations.

**USES.**—Magnesium alloys are used for airplane parts, including crank-cases, starting equipment, instrument housings, and landing and tail wheels. Sheet, plate and structural shapes are used to decrease the dead-load weight of trucks and trailers. The alloys also are used in portable tools and high-speed automotive equipment to decrease weight and increase operating efficiency.

# TUNGSTEN AND MOLYBDENUM

By Zay Jeffries

## 1. TUNGSTEN

Tungsten is a metallic element with peculiar properties that render it highly important in industry. Tungsten ingots for working are pressed from dry powder and heated to a temperature near the melting point to consolidate the metal. The resulting ingots are brittle when cold, but somewhat malleable when hot.

In swaging an ingot from about $^1/_4$ in. square, the work all being done at high temperature, the rod is brittle cold and malleable hot until the swaged rod is about 50 mils diameter. At a size near 50 mils diameter, above or below as the working temperature is low or high respectively, the swaged rod becomes ductile when cold.

**Table 1.—Tensile Strength of Tungsten**

| Kind of Material | Diameter, mils | Tensile Strength, lb. per sq. in. | Kind of Material | Diameter, mils | Tensile Strength, lb. per sq. in. |
|---|---|---|---|---|---|
| Sintered tungsten ingot | 200 × 250 | 18,000 | Drawn wire........ | 5.78 | 366,000 |
| Swaged rod.......... | 216 | 50,000 | " " ........ | 5.50 | 378,000 |
| " " .......... | 125 | 107,000 | " " ........ | 3.96 | 483,000 |
| " " .......... | 80 | 176,000 | " " ........ | 1.14 | 590,000 |
| " " .......... | 26 | 215,000 | " " ........ | 1.00 | 650,000 |
| Drawn wire.......... | 18 | 264,000 | " " ........ | 0.50 | 700,000 |
| " " .......... | 7.23 | 340,000 | | | |

**PHYSICAL AND MECHANICAL PROPERTIES.**—The properties of tungsten are: atomic weight, 184.0; density, from 0.672 to 0.690 lb. per cu. in.; melting point, 6120° F. (highest of all metals, and, among the elements, exceeded only by carbon); coefficient of linear expansion, 0.000 002 467 per deg. F. at 70° F. (Worthing); electrical resistivity at 0 deg. C., 0.000 004 91 ohm-cm. for pure single crystal wire, 0.000 005 05 for ordinary polycrystalline wire; at 20 deg. C., 0.000 005 50. Very fine wire as drawn may have a value as high as 0.000 008, which is reduced to about 0.000 006, or somewhat less, by heating to a high temperature. The average temperature coefficient of resistivity between 0 and 100 deg. C. is about 0.0047 per deg. C. Resistance at 2100 deg. C. is about 12 times, and at 2400 deg. C. about 15 times the resistance at room temperature. Modulus of elasticity is about 57,000,000. Poisson's ratio is 0.17, independent of temperature. The thermal conductivity is 812 B.t.u. per sq. ft. per hr. per deg. F. per 1 in. thickness at 2632° F., 1152 B.t.u. at 32° F., 1610 B.t.u. at −310° F., and 2380 B.t.u. at −422° F. It is weakly paramagnetic. Tungsten crystallizes with a body-centered cubic space lattice, the length of the side of the unit cube being 3.155 Angstrom units,* and the nearest approach of atom centers, 2.732 Angstrom units. Table 1 shows the improvement in tensile strength of tungsten by working.

Drawn tungsten can be treated to a higher temperature than any other metal without removing the effects of working (strain hardening). A 25-mil drawn wire had a tensile strength of 101,000 lb. per sq. in. at 1635° F., a value not equaled by any other metal.

Drawn tungsten wire has some peculiar properties. When ductilized by the swaging and drawing process, it is typically ductile. That is, it can be drawn further cold, but it is not typically malleable; it splits into many fibers when hammered on an anvil. When ordinary drawn metals are heated so as to cause the strain-hardened grains to recrystallize, the ductility, largely lost because of the drawing, is restored. Ductile tungsten so treated becomes brittle at ordinary temperature, and to make it again ductile it must be further worked hot, but below the recrystallization temperature.

**CORROSION.**—Powdered tungsten is black when finely divided and gray when coarse. In air, at ordinary temperatures, the coherent and dense metal slowly oxidizes on the surface until inhibited by a thin film of oxide. Tungsten is dissolved by fused alkali nitrates and slowly attacked by hot *aqua regia* or strong hydrogen peroxide. It is scarcely attacked by hydrochloric, nitric or sulphuric acids, because of the formation of a protective film, or tungstic acid. It is rapidly dissolved by hydrofluoric acid containing nitric acid, giving a clear solution. When heated above a red heat, it oxidizes fairly rapidly. When exposed at high temperatures to carbon monoxide or hydrocarbon gases, two tungsten carbides are formed (reported to be WC and $W_2C$).

**USES.**—Drawn tungsten wire is used for the pressed filaments of incandescent lamps. Tungsten also is used for talking machine needles, filaments for radio and rectifier tubes, electric furnace resistors, and lamp filament supports. Wrought tungsten, made in rods up to about $^{3}/_{4}$ in. diameter, and in sheet, is used principally for x-ray targets and contacts for internal-combustion ignition appliances. Most of the tungsten produced is used in alloy-steel, principally high-speed steel, which contains about 18% tungsten, and in magnet steel. Tungsten is added to steel either as tungsten powder, or preferably as ferro-tungsten containing at least 75% tungsten.

**CEMENTED CARBIDES.**—Tungsten is the most important element in the so-called cemented carbides used extensively for cutting tools, wire drawing and other dies, and wearing parts of machinery. The cemented tungsten carbide is made by mixing very fine tungsten powder approximating the composition of WC, with about 3 to 20% by weight of metallic cobalt, according to the hardness and wearing properties desired, and sintering at a temperature far below the melting point of the tungsten carbide. This material is relatively expensive as compared with the common cutting tools, and tools usually are made by copper-brazing a small tip of cemented carbide to a steel shank. Drawing dies usually are made by mounting a small piece of cemented carbide in a steel casing. Other carbides, as tantalum carbide and titanium carbide, have been added to tungsten carbide, principally for enhancing the steel-cutting qualities.

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* 1 Angstrom unit = $10^{-8}$ centimeter.

## 2. MOLYBDENUM

Molybdenum is a hard, silvery-white metallic element. Wire is made by hot drawing, at least in the early stages of drawing. The ingots are prepared by pressing powdered metal and consolidating below the melting point. These are brittle when cold and must be worked hot. After the hot working has progressed to a certain point, the rod becomes ductile when cold.

**PHYSICAL AND MECHANICAL PROPERTIES.**—Some of the properties of molybdenum are: atomic weight, 96.0; density, 0.3688 lb. per cu. in.; coefficient of linear expansion, 0.000 002 77 per deg. F. at 70° F. (Worthing); electrical resistivity at 0 deg. C., 0.000 005 08 ohm-cm.; at 2000 deg. C., 0.000 006 15; average temperature coefficient of resistivity between 0° and 100° C., 0.0047; thermal conductivity, 992.4 B.t.u. per sq. ft. per hr. per deg. F. per 1 in. thickness. It is weakly paramagnetic. Molybdenum crystallizes with a body-centered cubic space lattice, the length of a side of the unit cube being 3.142 Angstrom units and the nearest approach of atom centers 2.720 Angstrom units. Table 2 gives typical mechanical properties of molybdenum wire, 25 mils diameter, treated in various ways.

Brittle molybdenum can be ductilized like tungsten by working below the recrystallization temperature. Unlike tungsten, drawn wire can be recrystallized with an increase in ductility if the annealing temperature is low, say 1920° F., so as to produce a fine-grained structure. Higher annealing temperatures produce larger grains, causing a diminution in cold ductility.

**CORROSION.**—Dense and coherent molybdenum slowly oxidizes superficially, until action is stopped by air at ordinary temperatures with the formation of a protective film of oxide. At red heat, the oxidation proceeds rapidly with the formation and volatilization of molybdenum trioxide. Molybdenum is attacked or dissolved by nearly all reagents which attack tungsten. It is soluble in nitric acid, sulphuric acid and hydrogen peroxide.

**USES.**—Molybdenum is used mostly as an alloying element in steel. Although it imparts characteristics to steel somewhat similar to those imparted by tungsten, the two metals are not interchangeable, volume for volume. Molybdenum never has been as successful in high-speed steel as tungsten. It is used chiefly in structural steels for automobiles in amounts less than 1%. The wrought metal is used for x-ray targets, the wave length being quite suitable for the study of crystal structure.

**Table 2.—Mechanical Properties of Molybdenum**

| Description | Tensile Strength, lb. per sq. in. | Elongation in 2 in., percent | Reduction of Area, percent |
|---|---|---|---|
| Swaged and drawn hot (1000–1300° C.) from 25 to 125 mils | 148,000 | 4.7 | 68 |
| Swaged and drawn hot (black heat–800°) from 25 to 125 mils | 154,000 | 5.7 | 65 |
| Annealed to produce large grains........................ | 42,500 | 3.15 | None |
| Annealed to produce very small grain size............... | 125,000 | 20.0 | 66 |

# DIE-CASTING ALLOYS

**By J. C. Fox**

Die-casting alloys may be classified as: 1. Tin-base alloys; tin, alloyed with copper, antimony and tin. 2. Lead-base alloys; lead, alloyed with antimony or antimony and tin. 3. Zinc-base alloys; zinc, alloyed with aluminum, magnesium and copper. 4. Aluminum-base alloys. 5. The brasses and bronzes; copper alloyed with zinc, tin, lead, aluminum, nickel, manganese, etc. 6. Magnesium-base alloys; magnesium, alloyed with aluminum, manganese and nickel.

General properties and application of typical alloys of each group are described below:

**TIN-BASE ALLOYS.**—The compositions of typical tin-base alloys, used in die-casting, are as follows:

| Alloy No.............. | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| Copper, percent........ | 4.50 | 6.00 | 7.75 | 3.00 |
| Antimony, “ ........ | 4.50 | 8.00 | 7.75 | 10.50 |
| Lead, “ ........ | ..... | ..... | 1.75 | 25.00 |
| Tin, “ ........ | 90.00 | 86.00 | 82.75 | 61.50 |

Alloy No. 1 is the highest quality "babbitt" mixture corresponding to S.A.E. specifications for No. 10 Babbitt. It is used for main-shaft and connecting-rod bearings in the automotive and aircraft industries. No. 2 fulfils S.A.E. specification No. 11, and is used for bearings and other applications requiring a high-class tin-base alloy. No. 3 is a special automotive bearing composition. No. 4 (S.A.E. Specification No. 12), because of the high lead content, costs less than the others.

**LEAD-BASE ALLOYS.**—The composition of typical lead-base die-casting alloys is as follows:

| Alloy No. | 5 | 6 | 7 | 8 |
|---|---|---|---|---|
| Antimony, percent | 5 | 10 | 17 | 15 |
| Tin, " | .. | .. | .. | 5 |
| Lead, " | 95 | 90 | 83 | 80 |

Lead-base alloys are used where strength, hardness and other mechanical properties are not important, and a cheap non-corrosive metal is required. In chemical industries, lead-base alloys are used to resist the action of strong mineral acids. Because of their opacity to x-rays, these alloys are used in x-ray apparatus.

**ZINC-BASE ALLOYS** in common use conform to A.S.T.M. Designation B86–34T and S.A.E. Standards 921 and 903.* Composition of the alloys is as follows:

| | Cu | Al | Mg | Fe * | Pb * | Cd * | Sn | Zn |
|---|---|---|---|---|---|---|---|---|
| A.S.T.M. alloy XXI<br>S.A.E. alloy 921 | 2.50–3.50 | 3.50–4.30 | 0.02–0.10 | 0.100 | 0.007 | 0.005 | 0.005 | Remainder |
| A.S.T.M. alloy XXIII<br>S.A.E. alloy 903 | 0.10† | 3.50–4.30 | 0.03–0.008 | 0.100 | 0.007 | 0.005 | 0.005 | |

Stability of properties, dimensions and corrosion resistance require that the percentages of lead, tin and cadmium, as given above, be not exceeded. Highest purity metals must be used, *i.e.*, zinc, 99.99%+; electrolytic copper; aluminum, 99%+; magnesium, 99.9%+. Contamination by injurious metals during melting and casting must be avoided. Minimum physical properties, as called for by the A.S.T.M. and S.A.E. standards are as follows:

| | Tensile Strength | Charpy Impact | Elongation in 2 in. |
|---|---|---|---|
| Alloy XXI | 44,000 lb. per sq. in. | 6.0 ft.-lb. | 2% |
| Alloy XXIII | 35,000 " " " | 12.0 " | 3% |

**Table 1.—Physical and Mechanical Properties of Zinc-base Die-casting Alloys**

| Property | Alloy XXI | Alloy XXIII |
|---|---|---|
| Specific gravity | 6.7 | 6.64 |
| Density, lb. per cu. in. | 0.2421 | 0.2399 |
| Melting point, deg. F. | 734 | 728 |
| Specific heat | 0.10 | 0.10 |
| Thermal conductivity, B.t.u. per sq. ft. per hr. per deg. F. per 1 in. thickness | 725.7 | 783.8 |
| Thermal expansion, per deg. F. | 0.000 001 52 | 0.000 001 54 |
| Electrical conductivity at 25° C., microhms-cm. | 144,000 | 155,000 |
| Tensile strength, lb. per sq. in. | 48,000 | 40,000 |
| Elongation in 2 in., percent | 5 | 5 |
| Compressive strength, lb. per sq. in. | 93.000 | 60,000 |
| Shearing strength, lb. per sq. in. | 45,800 | 31,000 |
| Transverse deflection, in. | 0.22 | 0.27 |
| Brinell hardness No. | 83 | 74 |
| Charpy impact strength, ft.-lb. (1/4 × 1/4 in. bar) | 19 | 20 |

Table 1 gives average physical and mechanical properties of alloys XXI and XXIII, as determined from die-cast test bars, made in accordance with the A.S.T.M. specifications. Although alloy XXI has greater tensile strength and hardness than alloy XXIII, the latter, because of its greater dimensional stability and higher impact strength, which remains unchanged even under the severest conditions, usually is favored for applications requiring strict maintenance of properties and dimensions. Pressure die-castings made from these alloys have high strength, high ductility, resistance to impact, good corrosion resistance, and are easily machined. Practically every type of commercial finish may be satisfactorily applied. Zinc-base die-castings are used for automotive parts, household and office appliances, and miscellaneous articles.

**ALUMINUM-BASE ALLOYS.**—See p. 4–37.

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* These alloys also conform to the "Zamak" series of the New Jersey Zinc Co. (U. S. Patents 1596761, 1779525, 1852441).

† Maximum.

**COPPER-BASE ALLOYS.**—The composition of typical copper-base alloys, cast by a process perfected by Doehler Die Casting Co., in 1931, are as follows:

| | Copper | Zinc | Tin | Silicon |
|---|---|---|---|---|
| Alloy No. 1.... | 57.0–59.0 | 40.0–42.0 | 0.5–1.50 | ......... |
| " No. 2.... | 63.0–66.0 | 32.0–34.0 | ........ | 0.75–1.25 |
| " No. 3.... | 80.0–81.5 | 13.5–15.0 | ........ | 4.0–5.0 |

The mechanical properties are:

| | Tensile Strength | Yield Strength | Elongation in 2 in. | Brinell Hardness |
|---|---|---|---|---|
| Alloy No. 1.. | 55,000–65,000 lb. per sq. in. | 35,000–40,000 lb. per sq. in. | 15–20% | 120–130 |
| " No. 2.. | 65,000–75,000 " " " | 35,000–40,000 " " " | 15–20% | 110–120 |
| " No. 3.. | 85,000–95,000 " " " | 65,000–70,000 " " " | 10–15% | 160–180 |

**MAGNESIUM-BASE ALLOYS** are described on p. 4–55. The A.S.T.M. has issued a specification, Designation B94–34T, for an alloy with a specific gravity of 1.80, as follows: Magnesium, 88.0%; aluminum, 9.0—11.0%; manganese, min., 0.10%; nickel, max., 0.03%; copper, max., 0.05%.

## BEARING METALS

By Christopher H. Bierbaum

**BEARING METALS** are alloys composed of two or more metals in proportions in which they do not enter into a solid solution. They consist of relatively hard and soft microscopic particles or crystals intimately mixed. The function of the hard particles or bearing crystals is to support the load and resist the wear at times when actual metallic contact exists between the bearing surfaces. The softer particles, in a measure, owing to their plasticity, allow the harder particles to adjust themselves to the surface requirements of the journal. The softer particles also wear down lower than the harder particles and thus form slight depressions on the apparently smooth bearing surface. These depressions retain some of the lubricant and thereby effect a residuary lubrication, sufficient to prevent scoring or cutting during metallic contact. The property of a metal to retain lubrication upon a "run-in" bearing surface characterizes it as a bearing metal, and its capacity for doing so largely determines its value as such. A bearing metal, therefore, may be defined as an alloy that is capable of retaining a lubricant upon a bearing surface. Heretofore a run-in bearing surface could be produced only under service conditions, wherein it was necessary that all the disturbed metal on the surface, incident to tooling, should first be worn off; this was tedious and often caused bearing surface inaccuracies before a run-in condition was obtained. Since 1935, however, a high state of perfection of diamond tooling has enabled the production of bearing surfaces that approach insensibly close to those run-in in service.

This effects a twofold economy: it reduces the time for fitting and running-in; and it enormously prolongs the life of bearings. Nevertheless, journal and bearing should mutually polish each other. Bearing and journal therefore should bear some relation to each other, *i.e.*, their hardest particles, for best results, should be of the same order of hardness. These requirements would not be imperative, were the journal to possess infinite hardness and the cylindrical surface be ideally true. A highly accurate, ground and polished journal surface of chromium plate, nitrided nitralloy steel, or hardened tooled steel closely approaches this ideal state. The mating member, nevertheless, should have the microstructure of a bearing metal.

It is essential that, with two surfaces bearing together, one, say the journal, should have the highest possible microscopic homogeneity, and the other a heterogeneity, corresponding to that of a bearing metal.

**BABBITT METALS.**—The alloys in Table 1 are from A.S.T.M., 1933 Standards, Part 1, p. 845. These alloys cover the field very completely and can be recommended for the uses outlined below:

**No. 1** is suited for crank-pin service in steam and internal-combustion engines. It should be well confined in its backing or be used as a thin layer sweated on bronze or steel. It is the least likely to crack and the most plastic of the tin-base alloys.

**No. 2** is suited for substantially the same service as No. 1. It has, however, a higher bearing value, and is less likely to pound out. Like No. 1, it is especially suited for sweating on, owing to its high tin content.

**No. 3** is the hardest babbitt and has the highest bearing value. When allowed to cool and solidify very slowly, it develops the $SnCu_3$ crystal, which is the hardest copper-tin

Table 1.—Composition and Physical Properties * of White Metal Bearing Alloys

| Alloy Number | Specified Composition of Alloys, percent † | | | | Specific Gravity | Composition of Alloys Tested, percent | | | | Yield Point, lb. per sq. in. ‡ | | Johnson's Apparent Elastic Limit, lb. per sq. in. § | | Ultimate Strength, lb. per sq. in. ‖ | | Brinell Hardness ¶ | | Melting Point | Temperature of Complete Liquefaction | Proper Pouring Temperature |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Copper | Tin | Antimony | Lead | | Copper | Tin | Antimony | Lead | At 68 °F. | At 212 °F. | At 68°F. | At 212°F. | At 68°F. | At 212°F. | At 68°F. | At 212°F. | Deg.F. | Deg.F. | Deg.F. |
| 1 | 4.5 | 91.0 | 4.5 | ...... | 7.34 | 4.56 | 90.9 | 4.52 | None | 4400 | 2650 | 2450 | 1050 | 12,850 | 6950 | 17.0 | 8.0 | 433 | ........ | 825 |
| 2 | 3.5 | 89.0 | 7.5 | ...... | 7.39 | 3.1 | 89.2 | 7.4 | 0.03 | 6100 | 3000 | 3350 | 1100 | 14,900 | 8700 | 24.5 | 12.0 | 466 | 669 | 795 |
| 3 | 8 1/3 | 83 1/3 | 8 1/3 | ...... | 7.46 | 8.3 | 83.4 | 8.2 | 0.03 | 6600 | 3150 | 5350 | 1300 | 17,600 | 9900 | 27.0 | 14.5 | 464 | 792 | 915 |
| 4 | 3.0 | 75.0 | 12.0 | 10.0 | 7.52 | 3.0 | 75.0 | 11.6 | 10.2 | 5550 | 2150 | 3200 | 1550 | 16,150 | 6900 | 24.5 | 12.0 | 363 | 583 | 710 |
| 5 | 2.0 | 65.0 | 15.0 | 18.0 | 7.75 | 2.0 | 65.5 | 14.1 | 18.2 | 5050 | 2150 | 3750 | 1500 | 15,050 | 6750 | 22.5 | 10.0 | 358 | 565 | 690 |
| 6 | 1.5 | 20.0 | 15.0 | 63.5 | 9.33 | 1.5 | 19.8 | 14.6 | 63.7 | 3800 | 2050 | 3550 | 1800 | 14,550 | 8050 | 21.0 | 10.5 | 358 | 531 | 655 |
| 7 | ...... | 10.0 | 15.0 | 75.0 | 9.73 | 0.11 | 10.0 | 14.5 | 75.0 | 3550 | 1600 | 2500 | 1350 | 15,650 | 6150 | 22.5 | 10.5 | 464 | 514 | 640 |
| 8 | ...... | 5.0 | 15.0 | 80.0 | 10.04 | 0.14 | 5.2 | 14.9 | 79.4 | 3400 | 1750 | 2650 | 1200 | 15,600 | 6150 | 20.0 | 9.5 | 459 | 522 | 645 |
| 9 | ...... | 5.0 | 10.0 | 85.0 | 10.24 | 0.06 | 5.0 | 9.9 | 84.6 | 3400 | 1550 | 2400 | 950 | 14,700 | 5850 | 19.0 | 8.5 | 459 | 493 | 620 |
| 10 | ...... | 2.0 | 15.0 | 83.0 | 10.07 | 0.12 | 2.05 | 15.7 | 82.0 | 3350 | 1850 | 2250 | 1200 | 15,450 | 5750 | 17.5 | 9.0 | 468 | 507 | 630 |
| 11 | ...... | ...... | 15.0 | 85.0 | 10.28 | 0.19 | 0.09 | 14.8 | 84.7 | 3050 | 1400 | 2750 | 1100 | 12,800 | 5100 | 15.0 | 7.0 | 471 | 504 | 630 |
| 12 | ...... | ...... | 10.0 | 90.0 | 10.67 | 0.12 | 0.11 | 9.9 | 89.4 | 2800 | 1250 | 2250 | 950 | 12,900 | 5100 | 14.5 | 6.5 | 473 | 498 | 625 |

* The compression test specimens were cylinders $1\frac{1}{2}$ in. long and $\frac{1}{2}$ in. diameter, machined from chill castings 2 in. long and $\frac{3}{4}$ in. diameter. The Brinell tests were made on the bottom face of parallel machined specimens cast in a 2-in. diameter by $\frac{5}{8}$-in. deep steel mold at room temperature.

† Permissible variations in percentages of the desired elements are as follows:

| | | | | |
|---|---|---|---|---|
| Percentage of element specified..... | not over 2 | 2–5 incl. | 5–10 incl. | over 10 |
| Permissible variation, units of percent | 0.25 | 0.50 | 0.75 | 1.0 |

‡ The values for yield point were taken from stress-deformation curves at a deformation of 0.125 percent reduction of gage length.

§ Johnson's apparent elastic limit is taken as the unit stress at the point where the slope of the tangent to the curve is $1\frac{1}{2}$ times its slope at the origin.

‖ The ultimate strength values were taken as the unit load necessary to produce a deformation of 25 percent of the length of the specimen.

¶ These values are the average Brinell number of three impressions on each alloy using a 10-mm. ball and a 500-kg. load applied for 30 sec.

constituent; it, therefore, never should be mated with the softest steels. It is ideal for the highest service requirements when well seated and not subjected to loose pounding.

**No. 4** alloy has excellent bearing value for engine main bearings and general machinery, machine tools, and the like. It will withstand severe punishment and give general satisfaction. It is the most economical tin-base babbitt.

**Nos. 5 and 6** are intermediate alloys, neither tin nor lead base. They are alloys in which the soft matrix contains the tin-lead eutectic, for which reason they become mushy at a relatively low temperature. They are not to be recommended for either service or economy, as compared with Nos. 4 and 7.

**No. 7** is the most economical of the lead-base alloys. Its entire tin content is combined with antimony, giving a high percentage of bearing crystal; for that reason it is very serviceable. It is even suitable for use in many places where the high-tin babbitts are specified.

**Nos. 8, 9, and 10** are all lead-base babbitts belonging to the same general class. Their bearing value decreases with a decrease of tin content. They are serviceable with relatively light loads; with good workmanship they stand high speeds and are suitable for mating with the softer steels.

**Nos. 11 and 12** are antimonial lead alloys and constitute the cheapest grade of babbitt. They should be used for the lightest service only, such as line-shafting and less important bearings; they are suited to mate with the softest steels.

**BRONZE ALLOYS.**—Up to 1936 a vast number of bronze bearing formulas have been evolved, many of them of even doubtful value. The whole range of engineering work, however, can be covered satisfactorily by a comparatively small number of alloys. Those given in Table 2 are an approved list, from which the engineer can select a bearing for any and all requirements encountered in practice. This table is the result of selections and rejections of bearing formulas, during a period of more than 39 years, in an effort to reduce this entire field of alloys to an efficient minimum number.

The phosphorus content of a bronze, be it a trace or an amount not in excess of 0.25%, has no appreciable effect, either on the physical properties or bearing value of the resulting bronze; it is important, however, that deoxidation with phosphorus be specified and a trace of it required in the resulting alloy.

Considerable work and research have been done on casting bearing bronzes centrifugally; so far, however (1937), there is no proved advantage either as to physical properties or bearing value of any centrifugally-cast material. The advantages claimed are entirely due to the chill effect incident to the process, and the question is one of economical production. Chilling, however, can best be controlled in stationary molds.

The importance of high-class workmanship cannot be over-estimated, especially with the harder bearing metals and their harder mating journals. Grinding or lapping is ideal for the production of accurate surfaces, but, owing to the microconstituents of bearing metals, great care should be taken not to leave grinding particles embedded in the softer constituents of the bearing surface.

Two distinct tendencies, during the 16 years preceding 1937, for the improvement of bearing service apply to both bearing alloys and their steel journals: a large number of the more suitable alloys are now being produced, not only as sand cast, but also in a preferred chill cast condition; there is a very general tendency to increase the hardness of journals and shafts. Both conditions have aided materially in improving serviceability of bearings. It therefore has become a more exacting engineering problem to specify the most efficient combinations of bearings and journals. For this reason rather full comments are made on the following list of bronzes and their application. Furthermore, the mere selection of the proper alloy for any particular service does not insure that the bearing will be satisfactory. Many factors, including design, quality of workmanship, accuracy of fitting, clearance, oil grooving, etc., still enter the problem. See pp. 13–03 to 13–58.

**Alloy T–1** is a high-quality, worm gear bronze. As such it may be designated an international standard. It should not be used with very soft or low-carbon steel. In the sand-cast condition it is specified as: S.A.E. 65; A.S.T.M. B22–21 — grade D; A.R.E.A. grade D. Chill casting increases its physical properties. In this condition it has the widest application for motor car, truck, and bus worm gear drives. It also is widely applied in worm gear speed reducers. For best service it should mate with hardened steel, accurately finished and well polished.

**Alloy T–2** is a high-grade gear bronze. Its lead content does not detract from its physical properties, yet is sufficient to permit tooling to be done with greater accuracy than on T–1. It is suitable for elevator worm gears, worm gear speed reducers, spur gears meshing with small steel pinions, where quiet running and the wearing quality of a bronze is essential, feed nuts for working tools, and like service. When sand cast it is specified as S.A.E. 63. Chill casting increases its physical properties, making it an exceptionally high-

### Table 2.—Composition and Properties of Bronze Bearing Alloys

| Properties | T–1 * | T–1 † | T–2 * | T–2 † | T–3 * | T–3 † | T–4 * | T–4 † | T–5 * | T–6 * | T–7 * | T–7 † | T–8 * | T–8 † | T–9 * | T–10* | T–11* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Composition: | | | | | | | | | | | | | | | | | |
| Percentage of copper | 88.75 | 88.75 | 87.50 | 87.50 | 85.0 | 85.0 | 84.0 | 84.0 | 84.00 | 80.0 | 80.0 | 80.0 | 78.0 | 78.0 | 70.0 | 70.0 | 10.0 |
| " aluminum | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 4.0 |
| " tin | 11.00 | 11.00 | 11.00 | 11.00 | 10.00 | 10.00 | 10.00 | 10.00 | 16.00 | 20.00 | 10.0 | 10.0 | 8.0 | 8.0 | 9.0 | 4.0 | ...... |
| " lead | 0.25 | 0.25 | 1.50 | 1.50 | 5.0 | 5.0 | 2.5 | 2.5 | ...... | ...... | 10.0 | 10.0 | 14.0 | 14.0 | 21.0 | 26.0 | ...... |
| " nickel | ...... | ...... | ...... | ...... | ...... | ...... | 3.5 | 3.5 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| " phosphorus | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | ‡ | Zn 86.0 |
| Tensile properties: | | | | | | | | | | | | | | | | | |
| Ultimate strength, 1000 lb. per sq. in. | 30–54 | 40–55 | 45–57 | 45–52 | 40–50 | 40–57 | 45–60 | 45–60 | 30–40 | 26–33 | 35–45 | 35–45 | 30–40 | 36–42 | 27–29 | 18–23 | 32–36 |
| Yield point, " " " " " | 21–24 | 22–25 | 20–25 | 25–27 | 20–23 | 21–23 | 27–32 | 28–33 | 25–28 | 18–21 | 19–22 | 20–22 | 17–19 | 18–22 | 17–19 | 12–16 | 32–36 |
| Elastic limit, " " " " " | 13–16 | ...... | 11–16 | ...... | 15–16 | ...... | 19–22 | ...... | 13–19 | 17–18 | 10–13 | ...... | 6–13 | ...... | ...... | ...... | 32–36 |
| Elongation in 2 in., percent | 20–40 | 10–20 | 15–40 | 10–20 | 20–40 | 10–20 | 7–20 | 8–20 | 1–2 | 0.5 | 20–40 | 6–15 | 15–30 | 10–15 | 14–17 | 12–20 | 0 |
| Reduction of area, percent | 18–35 | ...... | 14–37 | ...... | 18–38 | ...... | 6–18 | 7–18 | 0.8–1.5 | 0.4 | 18–27 | 5–13 | 12–24 | 8–13 | 12–14 | 10–15 | 0 |
| Compression load, 1000 lb. to produce compression of 0.001 in. | 15–17 | 22 | 15–17 | 17–19 | 13–14 | 14–16 | 20–24 | 24 | 18–25 | 24–28 | 13–15 | 18–20 | 13–15 | 17–18 | 12 | 9–10 | 25 |
| 0.1 in. | 53–54 | 72 | 44–51 | 60–67 | 40–48 | 50 | 58–64 | 66 | ...... | 100 | 45–47 | 50–54 | 38–48 | 46–55 | ...... | ...... | 75 |
| 0.15–0.17 in. | ...... | 100 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 50 | ...... | ...... |
| 0.1– 0.2 in. | ...... | ...... | ...... | 100 | ...... | ...... | 100 | 100 | 100 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 0.22–0.25 in. | 100 | ...... | ...... | ...... | 100 | 100 | ...... | ...... | ...... | ...... | ...... | 100 | ...... | 100 | ...... | ...... | ...... |
| 0.23–0.28 in. | ...... | ...... | 100 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 100 | ...... | ...... | ...... | ...... | ...... | ...... |
| 0.27–0.30 in. | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 50 | ...... |
| 0.3 –0.33 in. | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 100 | ...... | ...... | ...... | ...... |
| Brinell hardness No. | 67–77 | 74–93 | 60–75 | 70–80 | 57–72 | 65–74 | 80–93 | 83–93 | 83–93 | 130–143 | 55–70 | 65–73 | 48–54 | 61–65 | 59–63 | 41–45 | 114–119 |
| Specific gravity | 8.5 | ...... | 8.8 | ...... | 8.7 | ...... | 8.8 | ...... | 8.6 | 8.7 | 8.9 | ...... | 9.4 | ...... | 9.1 | 9.3 | 6.9 |
| Weight, lb. per cu in. | 0.307 | ...... | 0.316 | ...... | 0.314 | ...... | 0.318 | ...... | 0.310 | 0.314 | 0.322 | ...... | 0.339 | ...... | 0.346 | 0.376 | 0.250 |
| Patternmaker's shrinkage, in. per ft. | 3/16 | 1/8 | 3/16 | 1/8 | 3/16 | 1/8 | 3/16 | 1/8 | 1/8 | 1/8 | 3/16 | 1/8 | 3/16 | 1/8 | 3/16 | 3/16 | 1/8 |
| Electrical conductivity | 10.1 | ...... | 10.2 | ...... | 11.3 | ...... | 9.6 | ...... | 7.3 | 6.7 | 9.2 | ...... | 11.2 | ...... | 7.7 | 13 | 25.7 |
| Impact test (Izod) | 3–7 | ...... | 3–6 | ...... | 4–7 | ...... | 2–4 | ...... | 1–1.3 | ...... | 2–4 | ...... | 4–5 | ...... | 3.7 | 4.3 | ...... |
| Endurance (Landgraf-Turner) | 200–600 | ...... | 50–250 | ...... | 40–690 | ...... | 14–42 | ...... | 0 | ...... | 150–500 | ...... | 150–350 | ...... | 36–116 | 32–106 | ...... |
| Modulus of elasticity × $10^5$ | 124 | ...... | 124 | ...... | 108 | ...... | 125 | ...... | 97 | ...... | 85 | ...... | 88 | ...... | ...... | ...... | ...... |

* Sand cast. † Chill Cast. ‡ Deoxidized with phosphorus.

grade bearing bronze for very high speeds, as in steam turbine worm wheels. With fair workmanship, at lower speeds it will support relatively heavy loads.

**Alloy T-3**, an intermediate bronze between T–1 and T–7, is a much-used composition for machine tool bearings. It is well adapted for worm gears where the softer steels are used, and also is suitable for steel mill bronze inserts and segmental bearing sections, feed nuts, bushings, and general machinery bearings. Its lead content permits easy and accurate tooling. Chill casting increases the physical properties, especially compressive strength, making it more serviceable for use with heavier loads.

**Alloy T-4** is a nickel-phosphor-bronze. It has the lowest coefficient of thermal expansion of all the copper-tin bearing and phosphor bronzes. It is suitable for heavy punishment when the bearing surface of its mating member is either hardened steel, nitrided nitralloy, or chromium plated, accurately finished and polished, and is also suitable for valve stem guides for internal-combustion engines. The lead content gives it an accurate tooling quality. Chill casting increases its density and refines the grain. It then has wide application for the highest requirements of worm wheel service and speed reducers, both in the largest and smallest sizes. It is favored for small worm wheels in portable tools. For hard bronzes in Diesel engines and like service, if combined with corresponding workmanship and mating material, it is unsurpassed.

**Alloy T–5** is especially adapted to service requiring high compressive strength. It is suitable for low-speed and heavy-pressure journals not exceeding 1500 lb. per sq. in. It is well adapted for thrust bearings, thrust washers, and step bearings, pivot bearings, dies, and trunnions, and is suitable for bascule and lift bridges. With light loads and accurate workmanship it will support an exceedingly high speed. It is specified as: A.S.T.M. B22–21, grade B; and A.R.E.A. grade B.

**Alloy T–6** is adapted to very heavy duty on thrust and pivot bearings for lift and swing bridges and the like, with very high pressures and comparatively low speeds. For this purpose it is second only to Alloy T–11. It should be used only against hardened steel surfaces. Accurately fitted, it will withstand bearing pressures in excess of 1500 lb. per sq. in. It is specified as: Federal Specification Board Q–Q–B–691, composition 10; U. S. Navy 46–B22, grade 4; U. S. Army Ordnance Dept. 57–702, grade 10; A.S.T.M. B22–21, grade A; A.R.E.A. grade A.

**Alloy T–7** is a leaded phosphor-bronze, an old standard for general machine construction, suitable for crosshead and crank-pin bearings and for general machinery, lathes, grinders, etc., for relatively high speeds and journals, not heat-treated. When sand cast, it is specified as: U. S. Navy 46–B22, grade 2; U. S. Army Ordnance Dept. 57–70B, grade 8; S.A.E. 64. Chill casting increases its physical properties materially, causing a finer and more perfect distribution of the lead with a general refinement of grain, making a superior bearing metal for severe punishment.

**Alloy T-8** is a relatively high-lead bronze suitable for brass and copper rolling mill neck bearings and bushings, where the spindles are of soft steel and the speed relatively high. In general, it is adapted for the intermediate or softer grades of steel journals, with intermediate or light pressures and relatively high speeds. It is substantially the famous Penn. R. R. alloy known as Ex. B metal. When sand cast it is specified as: U. S. Navy 46-B22, grade 1; U. S. Army Ordnance Dept. 57-70B, grade 7; S.A.E. 67. Chill casting increases its physical properties materially, owing to the finer distribution of the lead and a general grain refinement, with the result that in the chilled condition this alloy is capable of resisting greater punishment with higher bearing and wearing qualities.

**Alloy T 9** is a relatively high-lead bronze. It is serviceable for bearings where the danger of melting babbitts exists. It fills a field where the harder babbitts are otherwise especially serviceable. On account of its lead content and resulting plasticity, it is serviceable with poor workmanship and a soft steel mating member, in that the copper-tin crystal is hard enough to produce a mutual polishing effect between the bearing and its journal.

**Alloy T 10** is a very high-lead alloy, representing the probable limit of usefulness of high lead content. Its composition and serviceability are substantially those of the old plastic bronzes. It is suitable for poor workmanship, and competes with the intermediate and cheaper grades of babbitt, having the advantage of not melting out when heating.

**Alloy T 11**, Lumen bronze, is a unique hard zinc-base alloy. Its capacity for holding an oil film after it has once been established is unequaled by any other alloy. It is hard and unyielding, and therefore requires care in fitting. It is especially serviceable for use where high compressive strength is required, and has an enviable record for all kinds of bridge bearings, for pivots, thrust collars, bascule trunnion bushings, and bearings in draw, swing, and lift bridges. It is suitable for turntable pivots, cold neck rolling mill table bearings, crane bearings, etc. In electrical and in general machine tool service it has phenomenally long life, and is well suited for fractional-horsepower motors. It must not be subjected to excessive pounding and impact.

**THERMAL EXPANSION OF BEARING BRONZES.**—Table 3 gives the linear coefficients of thermal expansion for the bearing bronzes. It supplies exact data for computing expansions and clearance of bearings at different temperatures. The expansion of these alloys is rather irregular, and therefore, it is necessary that these data be given in each case for several temperature intervals.

**Table 3.—Linear Coefficient of Thermal Expansion of Bearing Bronzes**

(In. per in. per deg. F.)

Made on Sand-cast Specimens

| Alloy No. | Between 70° F. and | | | | |
|---|---|---|---|---|---|
| | 200° F. | 300° F. | 400° F. | 500° F. | 600° F. |
| T- 1 | 0.000 009 63 | 0.000 009 74 | 0.000 010 02 | 0.000 010 31 | 0.000 010 58 |
| T- 2 | .000 009 67 | .000 009 89 | .000 010 13 | .000 010 40 | .000 010 61 |
| T- 3 | .000 009 83 | .000 009 98 | .000 010 25 | .000 010 48 | .000 010 67 |
| T- 4 | .000 009 48 | .000 009 69 | .000 009 87 | .000 010 15 | .000 010 44 |
| T- 5 | .000 009 79 | .000 009 98 | .000 010 18 | .000 010 46 | .000 010 72 |
| T- 6 | .000 010 08 | .000 010 14 | .000 010 37 | .000 010 61 | .000 011 02 |
| T- 7 | .000 009 92 | .000 010 04 | .000 010 23 | .000 010 40 | .000 010 69 |
| T- 8 | .000 010 13 | .000 010 21 | .000 010 33 | .000 010 54 | .000 010 67 |
| T- 9 | .000 010 11 | .000 010 23 | .000 010 44 | .000 010 63 | .000 010 84 |
| T-10 | .000 010 06 | .000 010 28 | .000 010 43 | .000 010 59 | .000 010 62 |
| T-11 | .000 014 17 | .000 015 30 | .......... | .......... | .......... |

# SOLDERS

The two types of solders most generally used are lead-and-tin solders and silver solders. The latter are used in connection with high operating temperatures and for high-strength joints. They also can be used on other metals than copper, including stainless steel. Brazing alloys containing copper and phosphorus cost less than silver solders and are used on copper and copper alloys.

**LEAD-TIN SOLDERS.**—The composition and melting point of lead-tin alloys used as solders, and commercially known as soft solders, are given in A.S.T.M. standard specifications, Designation B32–21. Class A solders are required to be made of new or virgin metal. Class B solders are made from at least one-half virgin metal, the balance being recovered or secondary metals. Only Class A solders should be used on zinc and galvanized iron.

Table 1 shows the composition and melting point of these solders. The solders are completely solid below the temperature given for melting, and completely liquid at temperatures above the liquefaction temperature. Grade 4 is particularly suitable for making wiped joints.

**SILVER SOLDERS** contain a greater or less amount of silver together with varying quantities of copper and zinc, and are used for brazing. Table 2 gives the composition and characteristics of silver solders covered by A.S.T.M. standard specifications, Designa-

**Table 1.—Composition and Melting Points of Lead-Tin Solders**

| Grade | Tin | 0A | 1A | 1B | 2A | 2B | 3A | 3B | 4A | 4B | 5A | 5B | Lead |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Sn, % | 100.0 | 63.00 | 50.00 | 49.25 | 45.00 | 43.50 | 40.00 | 38.00 | 37.50 | 35.50 | 33.00 | 31.00 | ..... |
| Pb, % | ..... | 37.00 | 50.00 | 50.00 | 55.00 | 55.00 | 60.00 | 60.00 | 62.50 | 62.50 | 67.00 | 67.00 | 100 |
| Sb, % | ..... | 0.12 | 0.12 | 0.75 | 0.12 | 1.50 | 0.12 | 2.00 | 0.12 | 2.00 | 0.12 | 2.00 | ..... |
| Melts at, °F. | 449.6 | 357.8 | 357.8 | 365.0 | 357.8 | 370.4 | 357.8 | 370.4 | 357.8 | 370.4 | 357.8 | 370.4 | 620.6 |
| Liquefies at, °F. | 449.6 | 357.8 | 415.4 | 397.4 | 437.0 | 428.0 | 458.6 | 442.4 | 467.6 | 411.8 | 485.6 | 455.0 | 620.6 |

**Table 2.—Composition and Properties of Silver Solders**

| Grade | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| Ag, % | 10 | 20 | 20 | 45 | 50 | 65 | 70 | 80 |
| Cu, % | 52 | 45 | 45 | 30 | 34 | 20 | 20 | 16 |
| Zn, % | 38 | 35 | 30 | 25 | 16 | 15 | 10 | 4 |
| Cd, % | † | † | 5 | Nil | Nil | Nil | Nil | Nil |
| Impurities, % * | 0.15 | 0.15 | 0.15 | 0.15 | 0.15 | 0.15 | 0.15 | 0.15 |
| Melts at, °F. | 1510 | 1430 | 1430 | 1250 | 1280 | 1280 | 1335 | 1360 |
| Flows at, °F. | 1600 | 1500 | 1500 | 1370 | 1425 | 1325 | 1390 | 1460 |
| Color | Yellow | Yellow | Yellow | Nearly white | Nearly white | White | White | White |

* Maximum. † The addition of not over 0.50% cadmium to assist fabrication shall not be considered a harmful impurity.

tion B73–29. Grades 1–3 are suitable for brazing where a freer flowing solder than the ordinary copper-zinc brazing solders is required. Grades 4 and 5 will flow more freely at lower temperatures than grades 1–3. They are used where strong joints are required, and where the temperature necessary to flow grades 1–3 would injure the metal to which the solder is applied. Grades 6–8 are used where a high degree of malleability or ductility is required.

**SOLDERING FLUXES.**—Neutral fluxes, as resin, or resin in alcohol, are non-corrosive. They are used where corrosive action must be avoided, and where washing is impracticable. Their cleaning action is slight and they are not suitable for oxidized surfaces. More active fluxes comprise zinc chloride or ammonium chloride, or both, dissolved in water or mixed with petrolatum paste in various proportions. They are corrosive, and the joints, therefore, should be washed in hot water after soldering. For silver solder and brazing alloys, powdered borax is used as a flux.

# Section 5

# NON-METALLIC MATERIALS

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## REFRACTORIES

Revised by J. Spotts McDowell

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## WOOD

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## CORK

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## RUBBER

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## CARBON

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## MISCELLANEOUS MATERIALS

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# NON-METALLIC MATERIALS

## REFRACTORIES

**Revised by J. Spotts McDowell**

### 1. FIREBRICK

Refractories, generically classed as firebrick, are structural materials used at high temperatures in industrial furnaces. The life of refractories is affected by the high temperatures to which they are subjected, and usually by two or more of the following influences: abrasion; corrosion through chemical action by slags and fluxes; erosion by mechanical action of molten metal or slag, or by dust-laden gases moving at high velocity; spalling of various types; the effect of various gases. The principal types of refractories are fireclay, high-alumina, silica, magnesite and chrome brick. Other types, in the special refractories class, are less extensively used. "Firebrick" when used alone is generally understood to mean fireclay brick. A glossary of the terms relating to refractories is given in Modern Refractory Practice, published by Harbison-Walker Refractories Co., Pittsburgh, Pa.

**REFRACTORINESS OF FIREBRICK (ALL TYPES).**—The terms fusion point, softening point and melting point are indefinite when applied to ceramic materials, which soften over a wide temperature range. No definite temperature marks the transition from the distinctly solid to the distinctly liquid state. Instead, the Pyrometric Cone Equivalent (P.C.E.) is generally used. It is an index of the heat-resisting qualities, or refractoriness of firebrick and other ceramic bodies.

**PYROMETRIC CONE EQUIVALENT** (P.C.E.) is defined, in the case of refractories, as the number of that standard cone whose tip would touch the supporting plaque simultaneously with a cone of the material being investigated, when tested in accordance with the standard method of test for softening point of fireclay brick (A.S.T.M. Designation C24–35). The determination of the P.C.E. (A.S.T.M. designation C24–35) consists in testing cones made of the ceramic material by comparing them with standard pyrometric cones, all heated together in the same furnace. It is made as follows:

1. A sample of the material to be tested is pulverized. The standard grind adopted by the A.S.T.M. is 60 mesh. (U. S. Std. Sieve Series.) 2. From this are molded test cones of similar size and shape as the standard pyrometric cones. 3. A test cone is placed in a suitable furnace with a series of standard cones, and heated at a definite rate until it softens and bends sufficiently for its tip to touch the base. At this temperature, one standard cone usually will have softened to approximately the same extent as the test cone. The number of this standard cone is the Pyrometric Cone Equivalent of the material being tested. Fusion points of standard pyrometric cones are given in Table 1.

**Table 1.—Softening Point of Standard Pyrometric Cones Used in Connection with Refractories***

*Proc.*, A.S.T.M., Vol. xxxiv, Part I, 1934

| Cone No. | Softening Point | | Cone No. | Softening Point | | Cone No. | Softening Point | | Cone No. | Softening Point | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Deg. C. | Deg. F. | | Deg. C. | Deg. F. | | Deg. C. | Deg. F. | | Deg. C. | Deg. F. |
| 15 | 1435 | 2615 | 20 | 1530 | 2786 | 29 | 1640 | 2984 | 34 | 1760 | 3200 |
| 16 | 1465 | 2669 | 23 | 1580 | 2876 | 30 | 1650 | 3002 | 35 | 1785 | 3245 |
| 17 | 1475 | 2687 | 26 | 1595 | 2903 | 31 | 1680 | 3056 | 36 | 1810 | 3290 |
| 18 | 1490 | 2714 | 27 | 1605 | 2921 | 32 | 1700 | 3092 | 37 | 1820 | 3308 |
| 19 | 1520 | 2768 | 28 | 1615 | 2939 | 33 | 1745 | 3173 | 38 | 1835 | 3335 |

* Fairchild and Peters, Characteristics of Pyrometric Cones, *Journal* Am. Ceramic Soc., vol. ix, No. 11, p. 701, Nov., 1926. These temperatures, which were determined for a heating rate of 150 deg. C. per hr. for cones 15 to 20, and of 100 deg. C. per hr. for cones 23 to 38, other conditions being the same as specified, apply satisfactorily for all the conditions of this test method, but do not apply to conditions of the commercial firing and use of refractory materials.

**FIRECLAY BRICK.**—The most widely used refractory brick are those made from flint fireclays and plastic fireclays, the essential ingredients of which are hydrous alumina silicates of the general type formula $Al_2O_3 \cdot 2SiO_2 \cdot 2H_2O$. A member of this group is kaolinite, a crystalline mineral of comparatively rare occurrence. Formerly, all clays were thought to contain it as their clay base, but the term is now restricted to the native mineral. It occurs in Colorado, Wales, and a few other localities. See The Kaolin Minerals, C. S. Ross and P. F. Kerr, Professional Paper 165–E, U. S. Geological Survey, 1931.

**Table 2.—Typical Fireclay Analyses**

| Brand of Clay | $SiO_2$ | $Al_2O_3 + TiO_2$ | $Fe_2O_3$ | CaO | MgO | Alkalies | Loss on Ignition |
|---|---|---|---|---|---|---|---|
| Penna. Hard | 45.3 | 38.2 | 2.1 | 0.22 | 0.30 | 1.17 | 13.0 |
| Penna. Soft | 49.6 | 35.0 | 2.4 | .49 | .35 | 1.85 | 10.5 |
| Penna. Plastic | 56.6 | 28.8 | 2.7 | .36 | .55 | 2.65 | 8.7 |
| Kentucky Hard | 42.9 | 40.2 | 1.9 | .30 | .20 | 0.60 | 13.2 |
| Kentucky Soft | 51.0 | 34.7 | 2.1 | .30 | .40 | 0.80 | 10.9 |
| Missouri Hard | 43.3 | 39.8 | 1.8 | .20 | .60 | 0.60 | 13.8 |
| Missouri Plastic | 55.9 | 29.5 | 2.6 | .30 | .80 | 1.10 | 10.0 |
| Ohio Hard | 43.8 | 38.5 | 2.0 | .40 | .60 | 0.80 | 13.6 |
| Ohio Plastic | 53.2 | 31.2 | 2.1 | .30 | .80 | 1.40 | 11.5 |
| Alabama Hard | 54.6 | 31.8 | 1.6 | .20 | .30 | 0.30 | 11.2 |
| Alabama Plastic | 56.3 | 29.8 | 2.6 | .50 | .40 | 1.80 | 8.7 |
| Georgia Kaolin | 45.5 | 38.0 | 1.4 | .30 | .20 | 0.30 | 13.7 |
| Maryland Hard | 52.0 | 34.2 | 1.3 | .40 | .20 | 0.50 | 11.6 |
| Maryland Plastic | 61.5 | 25.7 | 1.9 | .60 | .40 | 1.20 | 8.4 |
| New Jersey Soft | 67.8 | 21.2 | 1.4 | .30 | .30 | 2.10 | 6.7 |
| Colorado | 51.3 | 34.5 | 1.5 | .40 | .20 | 0.70 | 11.9 |

**Table 3.—Typical Analyses of High Heat Duty Fireclay Brick**

(P.C.E. = Cone 31 or over)

| Raw Materials | Silica $SiO_2$ | Alumina + Titania $Al_2O_3 + TiO_2$ | Iron Oxide $Fe_2O_3$ | Lime CaO | Magnesia MgO | Alkalies |
|---|---|---|---|---|---|---|
| Penna. Clays | 52.9 | 43.0 | 2.1 | 0.2 | 0.3 | 1.5 |
| Ohio Clays | 55.4 | 38.9 | 2.8 | .4 | .4 | 1.4 |
| Kentucky Clays | 54.6 | 41.4 | 2.3 | .2 | .3 | 1.1 |
| Missouri Clays | 54.6 | 41.7 | 2.0 | trace | .7 | 0.9 |
| Alabama Clays | 58.7 | 38.6 | 1.5 | .2 | .2 | 1.0 |
| Maryland Clays | 57.4 | 38.9 | 2.5 | .4 | .4 | 1.4 |
| New Jersey Clays | 81.1 | 17.2 | 1.4 | .15 | .2 | 1.2 |
| Colorado Clays | 62.4 | 34.2 | 1.9 | .3 | .4 | 0.95 |
| California Clays | 68.2 | 29.0 | 1.5 | .4 | .54 | 0.32 |
| Texas Clays | 75.05 | 21.94 | 2.30 | .4 | .62 | ..... |

**Table 4.—Typical Analyses of Intermediate Heat Duty Fireclay Brick**

(P.C.E. = Cone 28 to 31)

| Raw Materials | $SiO_2 + TiO_2$ | $Al_2O_3$ | $Fe_2O_3$ | CaO | MgO | Alkalies |
|---|---|---|---|---|---|---|
| Penna. Clays | 59.6 | 34.9 | 2.6 | 0.4 | 0.6 | 2.0 |
| Ohio Clays | 63.0 | 31.5 | 3.1 | .4 | .5 | 1.3 |
| Missouri Clays | 60.6 | 35.2 | 2.6 | trace | .7 | 1.1 |
| Alabama Clays | 61.1 | 32.5 | 3.8 | .5 | .4 | 1.8 |

The manufacture of fireclay brick consists in blending the raw materials, flint clay, plastic clay, and calcined clay or grog, either by grinding together in a wet pan, or by grinding together or separately in a dry pan, then mixing and tempering with the addition of water. The brick are molded by hand or mechanically, dried, either on hot floors or in dryers, and fired in periodic down-draft kilns or in continuous tunnel kilns. Table 5 gives pyrometric cone equivalents.

**HIGH-ALUMINA BRICK.**—In many parts of furnaces where the requirements are severe, high-alumina brick may be used to decided advantage. The equilibrium diagram (Fig. 1) of the alumina-silica system is of interest in considering alumina refractories.

High-alumina raw materials include bauxite, diaspore, cyanite, andalusite, sillimanite, and crystalline alumina or corundum. In manufacturing high-alumina brick, *i.e.*, brick containing 50% or more of alumina, these materials are used individually, collectively, or blended with flint and plastic clays according to the properties and percentage of alumina desired in the finished product. High-alumina brick are manufactured by methods similar to those used in manufacturing fireclay brick.

**Table 5.—Pyrometric Cone Equivalents (P.C.E.) for Various Types of Refractory Brick**

| Type | P.C.E. | Temperature Deg. F. | Temperature Deg. C. |
|---|---|---|---|
| Fireclay | | | |
| Super-duty | 33–34 | 3173–3200 | 1745–1760 |
| High heat duty (Aluminous) | 31–33 | 3056–3173 | 1680–1745 |
| Intermediate heat duty | 28–31 | 2939–3056 | 1615–1680 |
| Moderate heat duty | 26–28 | 2903–2939 | 1595–1615 |
| High heat duty (Siliceous) | 28–31 | 2939–3056 | 1615–1680 |
| High-alumina | | | |
| 50% Alumina | 34–35 (av.) | 3200–3245 | 1760–1785 |
| 60% Alumina | 36 (av.) | 3290 | 1810 |
| 70% Alumina | 37 (av.) | 3308 | 1820 |
| 80% Alumina | 38 (av.) | 3335 | 1835 |
| Silica | 31–32 | 3092 | 1700 |
| Magnesite | Above 38 | 3335+ | 1835+ |
| Chrome | Above 38 | 3335+ | 1835+ |

**Table 6.—Typical Analysis of High-alumina Raw Materials**

| Raw Material | $SiO_2$ | $Al_2O_3$ $+TiO_2$ | $Fe_2O_3$ | CaO | MgO | Alkalies | Loss on Ignition |
|---|---|---|---|---|---|---|---|
| Bauxite* | 5.5 | 61.9 | 1.7 | | | | 30.6 |
| Diaspore† | 28.36 | 56.44 | 0.56 | 0.12 | 0.26 | 0.97 | 13.46 |
| Diaspore† | 9.40 | 75.15 | 0.59 | .31 | .23 | .93 | 14.32 |
| Diaspore* | 16.90 | 66.70 | 1.20 | .30 | .80 | | 14.00 |
| Cyanite‡ | 40.5 | 57.2 | 1.5 | .1 | .2 | | 0.1 |
| Andalusite§ | 33.78 | 56.89 | ← | —5.37— | — | → | 3.67 |
| Sillimanite¶ | 33.42 | 63.80 | 1.31 | .46 | .13 | .48 | 0.62 |
| Crystalline Alumina (electrically fused) | 2.50 | 97.00 | 0.50 | | | | 0.00 |

* Analysis furnished by Harbison-Walker Refractories Co., Pittsburgh, Pa.

† A study of the Shrinkage of Diaspore Clays, Pt. 1, S. M. Phelps, *Jour.* Am. Ceramic Soc., ix, No. 10, Oct., 1926.

‡ Notes on Cyanite and Diaspore Refractories, E. J. Vachuska and G. A. Bole, *Jour.* Am. Ceramic Soc., x, No. 10, p. 761, Oct., 1927.

§ Progress Report on Use of Andalusite as a Refractory, Robt. Twells, Jr., *Jour.* Am. Ceramic Soc., viii, No. 8, p. 485, Aug., 1925.

¶ Some Notes on the Use of Indian Sillimanite, W. J. Rees, *Trans.* Cer. Soc., xxvi, p. 132 (1926–27).

**Table 7.—Typical Analyses of High-alumina Brick**

| Alumina Class | $SiO_2$ | $Al_2O_3$ | $Fe_2O_3$ | CaO | MgO | Alkalies | $TiO_2$ |
|---|---|---|---|---|---|---|---|
| 50 percent | 41.8 | 51.8 | 1.8 | 0.1 | 0.5 | 1.3 | 2.7 |
| 60 percent | 32.8 | 60.6 | 1.8 | .2 | .4 | 1.2 | 3.2 |
| 70 percent | 22.7 | 70.8 | 1.8 | .1 | .3 | 0.9 | 3.7 |
| 80 percent | 15.7 | 79.0 | 1.8 | .2 | .3 | 1.0 | 2.2 |

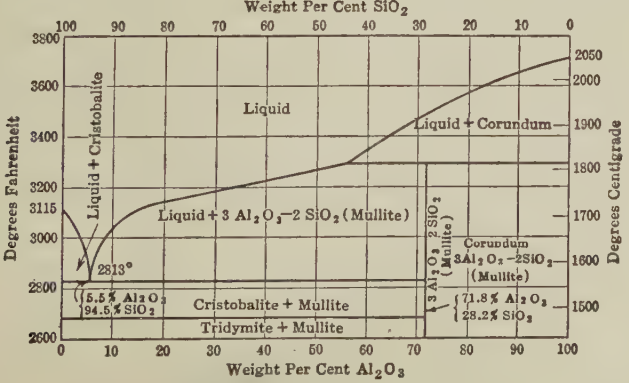


FIG. 1. Equilibrium Diagram of Alumina-silica System

**SILICA BRICK.**—(Abstracted from A Study of the Silica Refractories, J. Spotts McDowell, *Trans.* A.I.M. & M.E., lvii, p. 3, 1917.) The facts and data presented by the author have been subsequently confirmed by other investigators. Table 8 gives typical analyses of quartzites or ganisters from which silica brick are made.

**Table 8.—Typical Analyses of Quartzite**

| | $SiO_2$ | $Al_2O_3$ | $Fe_2O_3$ | CaO | MgO | Alkalies |
|---|---|---|---|---|---|---|
| Medina quartzite (Penna.) | 97.80 | 0.90 | 0.85 | 0.10 | 0.15 | 0.40 |
| Baraboo quartzite (Wisconsin) | 97.15 | 1.00 | 1.05 | 0.10 | 0.25 | 0.10 |
| Alabama quartzite | 97.70 | 0.96 | 0.80 | 0.05 | 0.30 | 0.31 |
| Colorado quartzite | 96.50 | 0.42 | 0.96 | 0.90 | 0.16 | 0.92 |
| Cambrian quartzite (Utah) | 98.0 | 0.7 | 0.25 | 0.3 | trace | 0.2 |

For refractory purposes, the quartzite, to produce a physically strong brick should be hard, dense, and crush to splintery angular fragments of varied size and shape. The strength of the brick depends on the silica content of the rock; generally the silica content should not be less than 96% and the alkalies should not exceed 0.5%. $Al_2O_3$ and $Fe_2O_3$ are necessary to form a bond; their combined percentage is usually about 1.75% and should not exceed 2.5%. The melting interval of quartzite depends upon the contents of the bases and for good quartzite is cone 35 to 36 (1755°–1775° C.; 3191°–3227° F.).

Chemical analysis alone does not fully indicate the behavior of quartzite on heating. Material suitable for refractory purposes will not weaken or crack under high temperature. It shows perceptible expansion after the first firing, which is essentially unchanged by subsequent firings. Expansion is due to: 1. Expansion of the mineral itself; 2. Increase in porosity. Table 9 shows the volume increase due only to the first factor, calculated from the specific gravities of 3 samples each, of good quartzite and quartzite unsuitable for refractory purposes, heated repeatedly to 1450° C.

**Table 9.—Volume Increase, Percent, of Quartzite After Repeated Burnings**

| No. of firings | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Good quartzite | 9.4 | 10.7 | 11.0 | 11.3 | 11.5 |
| Quartzite unsuitable for refractory purposes | 5.3 | 6.5 | 6.5 | 9.7 | 10.7 |

**Table 10.—Typical Analyses of Silica Brick**

| Raw Material | $SiO_2$ | $Al_2O_3$ | $Fe_2O_3$ | CaO | MgO | Alkalies |
|---|---|---|---|---|---|---|
| Medina quartzite (Penna.) | 95.9 | 1.0 | 1.0 | 2.0 | 0.10 | 0.10 |
| Baraboo quartzite (Wisconsin) | 95.1 | 1.3 | 1.2 | 2.1 | .20 | .20 |
| Weisner quartzite (Alabama) | 95.6 | 1.2 | 1.0 | 2.1 | .10 | .30 |
| Cambrian quartzite (Utah) | 96.35 | 0.6 | 0.5 | 2.2 | trace | ......... |

A paper by S. S. Cole, Effect of Firing at 1500° C. on Porosity and Specific Gravity of Quartzites (Am. Ceramic Soc., Refractories Div. Feb. 1931) contains a table giving descriptions and data covering quartzites in the U. S. and other countries.

The density of silica brick depends both upon the character of the quartzite used in its manufacture, and the temperature and duration of firing. The specific gravity of silica brick should lie between 2.28 and 2.40.

**EXPANSION OF SILICA BRICK.**—In manufacturing silica brick, permanent expansion occurs from the green brick size to the fired size. In service, under heat there is a further temporary expansion (often called thermal expansion) corresponding to the coefficient of expansion of silica, which disappears upon cooling. It is desirable that the permanent expansion be completed in the firing, but as the final increments of permanent expansion take place slowly under high temperature, silica brick will show a slight permanent expansion during reheating in service. (See Table 9.) The linear expansion in firing from green brick size to finished brick size ranges from 3.1% to 3.6%. The temporary expansion depends upon the temperature. The temporary expansion of Star silica brick tested at Mass. Inst. of Tech. at various temperatures was as follows:

| Temperature, deg. C. | 600 | 900 | 1200 |
|---|---|---|---|
| Temporary expansion | 0.7% | 0.9% | 1.1% |

The thermal expansion of well-burned silica brick amounts to 1.6% to 2.1%, the greater part of which occurs below 600° C.

In laying silica brick, it is important to provide for expansion at the rate of 1/8 to 3/16 in. per ft. Expansion joints should be kept free from mortar by packing to a depth of about 1 1/2 in., with a filler which will burn out, as sawdust, cardboard, or strips of wood.

Silica brick will not withstand rapid temperature changes. In both manufacture and service, heating and cooling must occur slowly and evenly, otherwise the brick will crack or spall.

**MAGNESITE BRICK.**—(Abstracted from "Magnesite Refractories," *Jour.* Am. Ceramic Soc., iii, No. 3, March, 1920, and from Basic Refractories for the Open Hearth, both by J. Spotts McDowell and Raymond M. Howe, *Bull.* No. 146, Feb., 1919, Am. Inst. of Min. Eng.) The chief use of magnesite is in the manufacture of refractory materials. Its chemical properties and high refractoriness make it highly suitable for furnace linings for metallurgical operations. For refractory purposes, it is sold in the form of bricks, of finely ground furnace magnesite for bricklaying, and of dead-burned grains for making and repairing furnace bottoms. The grains are a mixture of granules varying from 5/8 in. diameter to very fine but gritty particles.

**Table 11.—Typical Analyses of Crude Magnesites**

| | Crystalline Magnesite | | | Cryptocrystalline or Amorphous Magnesite | |
|---|---|---|---|---|---|
| | Austrian | Washington | Canadian | Grecian | Californian |
| $MgCO_3$ | 83–94 | 93–97† | ......... | 93.7–96.6† | 87.6–97.2† |
| $CaCO_3$ | 0.5–4.5 | 0.9–3.1† | 12.5–18 | 1.4–2.9† | trace—5.97† |
| $Fe_2O_3$ | 4–12 | ......... | ......... | ............ | ............ |
| $Al_2O_3$ | Up to 1 | ......... | ......... | ............ | ............ |
| $R_2O_3$* | ......... | 0.8–1.2 | ......... | 0.2–1.4 | 0.4–2.2 |
| $SiO_2$ | Up to 3 | 0.6–4.3 | ......... | 0.46–2 28 | 1.1–5.2 |
| $Mn_2O_3$ | Up to 0.5 | trace | ......... | ............ | ............ |

* $R_2O_3 = Fe_2O_3 + Al_2O_3$. † Calculated from MgO and CaO content.

**Table 12.—Typical Analyses of Dead-burned Magnesite**

| Type | MgO | CaO | $SiO_2$ | $Al_2O_3$ | $Fe_2O_3$ | $Mn_2O_3$ |
|---|---|---|---|---|---|---|
| Austrian | 85.6 | 2.9 | 2.8 | 0.6 | 7.1 | 0.6 |
| Canadian | 67.5 | 17.7 | 5.5 | 2.0 | 7.1 | ......... |
| Washington | 83.1 | 3.1 | 6.8 | 2.0 | 5.0 | trace |

**Metalkase Magnesite Brick.**—Metalkase brick consist of round or square soft steel containers packed with ground dead-burned magnesite. They are used for the back walls, bulkheads and ends of basic open-hearth furnaces and side walls of electric furnaces. Their special advantage lies in the fact that the metal containers fuse for one or two inches back from the surface exposed to the heat, impregnate the magnesite and bind the entire face of the wall. Joints, which form channels for cutting action of the flame are thus eliminated and the life of the wall greatly extended. For dimensions, shapes, and other details, consult catalog of Harbison-Walker Refractories Co., Pittsburgh.

**CHROME BRICK.**—(Abstracted from Chrome Refractories, by J. Spotts McDowell and H. S. Robertson, *Jour.* Am. Ceramic Soc., v. No. 12, Dec., 1922.) Chromite ($FeO \cdot Cr_2O_3$) is the most nearly neutral chemically of all the common refractories. It is acted upon by either strong acids or by strong bases. Because of its neutral chemical character and high refractoriness it is an important refractory in metallurgical furnaces. Chromite deposits occur in Rhodesia, the Transvaal, New Caledonia, Turkey, India, Greece, Canada, Cuba, and Brazil, and in the U. S., in California, Oregon, Wyoming, Alaska, North Carolina, Maryland, and Pennsylvania. (See Table 11.) Ores of the approximate analyses in Table 12 are used in the U. S. as refractory materials.

**Table 13.—Typical Analyses of Chromite**

1, 2 Asiatic Turkey; 3, 4, New Caledonia; 5, 6, 7, Quebec; 8, Greece; 9, 10, Cuba; 11, 12, California; 13, The Transvaal; 14, Rhodesia, South Africa.

| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $Cr_2O_3$ | 56.8 | 48.1 | 54.5 | 58.1 | 42.6 | 53.1 | 32.9 | 40.7 | 33.2 | 39.5 | 52.7 | 34.5 | 44.1 | 48.4 |
| FeO | 12.1 | 14.8 | 19.5 | 27.6 | 13.8 | 15.3 | 13.0 | 16.6 | 14.6 | 15.8 | 14.2 | 14.2 | 27.3 | 14.4 |
| $Al_2O_3$ | 15.0 | 12.5 | 11.0 | 3.8 | 8.3 | 8.0 | 8.3 | 24.3 | 28.8 | 26.2 | 12.5 | 19.0 | 12.8 | 12.1 |
| MgO | 14.0 | 16.5 | 8.0 | 3.2 | 21.1 | 16.1 | 24.6 | 14.9 | n. d. | 15.8 | 15.5 | 20.5 | 11.8 | 14.2 |
| $SiO_2$ | 1.5 | 6.2 | 3.1 | 5.1 | 9.5 | 6.4 | 14.0 | 3.0 | 6.0 | 3.0 | 4.0 | 11.0 | 3.8 | 5.9 |
| CaO | 0.7 | .... | 1.5 | .... | .... | 1.2 | .... | .... | .... | .... | 0.5 | 0.8 | trace | .... |

**Table 14.—Approximate Analyses of Chrome Ores Used in U. S.**

| | $Cr_2O_3$ | FeO | MgO | $Al_2O_3$ | $SiO_2$ |
|---|---|---|---|---|---|
| Grecian | 40.8 | 17.8 | 14.9 | 22.4 | 3.7 |
| South African | 44–48 | 14 | 15 | 12 | 5–8 |
| Cuban | 32.7 | 13.3 | 17.8 | 29.5 | 5.0 |

## 2. PROPERTIES OF REFRACTORIES

(Arranged Alphabetically)

**ABRASION (Resistance to).**—Tests for resistance to abrasion have been rather unsatisfactory. No method so far devised has been generally regarded as sufficiently reliable to warrant adoption as standard.

**COLD CRUSHING STRENGTH** of various types of refractories is given by J. Spotts McDowell (*Blast Furnace and Steel Plant*, Oct. and Nov., 1923). See Table 15.

**CONDUCTIVITY, THERMAL.**—Table 16 gives the approximate thermal conductivities of various refractories over a wide range of temperatures.

**EXPANSION, THERMAL.**—The linear expansion of refractory brick, for various types of refractories, is given in Table 17. The figures given represent average values for total expansion of brick from room temperature to the temperatures indicated.

**LOAD, RESISTANCE TO, AT HIGH TEMPERATURE.**—The behavior of refractory brick of various types under load at high temperature is indicated in Table 18.

**MODULUS OF RUPTURE.**—Average values of modulus of rupture of various types of refractory brick are given in Table 19.

**Table 15.—Cold Crushing Strength of Refractories**

Cold crushing strength in lb. per sq. in.; average values; bricks tested on flat

| Fireclay Brick | | | | High-alumina Brick 70% $Al_2O_3$ | Silica Brick | Magnesite Brick | Chrome Brick |
|---|---|---|---|---|---|---|---|
| Super-duty | High Heat Duty (Aluminous) | Moderate Heat Duty | High Heat Duty (Siliceous) | | | | |
| 4000 | 1000–4000 | Up to 5000 | 1500 | 4000 | 2000–4000 | 4500 | 4000 |

**Table 16.—Approximate Thermal Conductivity of Refractories**

From Modern Refractory Practice, Harbison-Walker Refractories Co., Pittsburgh

B.t.u. per hr. per sq. ft. per deg. F. per 1 in. thickness

| Temp., deg. F. | Kind of Brick | | | | Temp., deg. F. | Kind of Brick | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Fire-clay | Silica | Mag-nesite | Chrome | | Fire-clay | Silica | Mag-nesite | Chrome |
| 0 | 5.0 | 6.0 | 39.5 | 9.0 | 1400 | 9.0 | 11.3 | 26.9 | 12.5 |
| 200 | 5.6 | 6.8 | 37.2 | 9.5 | 1600 | 9.6 | 12.0 | 25.9 | 13.0 |
| 400 | 6.1 | 7.5 | 35.4 | 10.0 | 1800 | 10.1 | 12.8 | 25.0 | 13.5 |
| 600 | 6.7 | 8.3 | 33.3 | 10.5 | 2000 | 10.7 | 13.5 | 24.0 | 14.0 |
| 800 | 7.3 | 9.0 | 31.5 | 11.0 | 2200 | 11.3 | 14.3 | 23.5 | 14.5 |
| 1000 | 7.8 | 9.8 | 30.0 | 11.5 | 2400 | 11.8 | 15.0 | 23.2 | 15.0 |
| 1200 | 8.4 | 10.5 | 28.3 | 12.0 | 2600 | 12.4 | 15.8 | 23.0 | 15.5 |

**Table 17.—Thermal Expansion of Various Types of Refractory Brick**

(Expressed in linear percent)

| Temp., deg. F. | Type of Brick | | | | Temp., deg. F. | Type of Brick | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Fireclay 40% $Al_2O_3$ | Silica | Mag-nesite | Chrome | | Fireclay 40% $Al_2O_3$ | Silica | Mag-nesite | Chrome |
| 400 | 0.10 | 0.54 | 0.28 | 0.23 | 1600 | 0.47 | 1.34 | 1.18 | 0.96 |
| 800 | 0.23 | 1.12 | 0.56 | 0.47 | 2000 | 0.57 | 1.38 | 1.59 | 1.33 |
| 1200 | 0.34 | 1.28 | 0.86 | 0.70 | 2400 | 0.50 | 1.38 | 1.97 | 1.71 |

**Table 18.—Behavior of Refractory Brick Under Load at High Temperature**

| Type of Brick | Linear subsidence, percent* | Temperature of failure, deg. F.† |
|---|---|---|
| Fireclay—Super-duty | 4–5 | ........ |
| Fireclay—High heat duty (Aluminous) | 3–10 | ........ |
| Fireclay—High heat duty (Siliceous) | 2–4 | ........ |
| High-alumina—70% Alumina | 1–3 | ........ |
| Silica | ......... | 2800–2900 |
| Magnesite | ......... | 2400–2800 |
| Chrome | ......... | 2400–2550 |

* Standard Load Test, A.S.T.M. Designation C16–20. Under a load of 25 lb. per sq. in. on end, the specimen is heated according to a prescribed heating schedule to 1350° C. (2462° F.) and held at that temperature for 90 min.

† Standard Load Test, A.S.T.M. Designation C16–20. Under a load of 25 lb. per sq. in. on end, the specimen is heated according to a prescribed heating schedule until failure occurs.

**Table 19.—Modulus of Rupture at Atmospheric Temperature**

(A.S.T.M. Method C67–31, 7-in. span on flat)

| Type of Brick | Modulus of Rupture, lb. per sq. in. | Type of Brick | Modulus of Rupture, lb. per sq. in. |
|---|---|---|---|
| Fireclay | | High-alumina | |
| Super-duty | 400 | 50% Alumina | 750 |
| High heat duty (Aluminous) | 600–1200 | 70% Alumina | 1200 |
| Intermediate heat duty | 800–1400 | Silica | 500–1000 |
| High heat duty (Siliceous) | 400 | Magnesite | 1500 |
| | | Chrome | 1600–2000 |

**POROSITY.**—The porosity of refractories depends upon various factors, among them the character and quality of raw material, the grind, the water content at time of molding, the pressure of molding, and the time and temperature of firing. The porosity in turn affects the resistance of the brick to temperature change, the rigidity under load at high temperatures, the thermal conductivity, and the resistance of the brick to abrasion and slag action. For a method of test for porosity and permanent volume changes in refractory materials, adopted as standard, see A.S.T.M. Designation C20–33. Porosities of various types of refractory brick are given in Table 20.

**Table 20.—Porosities by Volume, Average Values**

| Type of Brick | Process of Manufacture | Porosity, percent | Type of Brick | Process of Manufacture | Porosity, percent |
|---|---|---|---|---|---|
| Fireclay | | | High-alumina | | |
| Super-duty | Machine | 12–16 | 50% Alumina | Machine | 20–25 |
| High heat duty (Aluminous) | Hand | 20–30 | 70% Alumina | Machine | 28–31 |
| High heat duty (Aluminous) | Machine | 15–24 | Silica | Hand or machine | 24–32 |
| Intermediate heat duty | Machine | 17–21 | Magnesite | Machine | 20–30 |
| High heat duty (Siliceous) | Hand or machine | 30 | Chrome | Machine | 20–30 |

**PYROMETRIC CONE EQUIVALENT** (P.C.E.).—See p. 5–03.

**REFRACTORINESS.**—(See Pyrometric Cone Equivalent.)

**RESISTIVITY (ELECTRICAL).**—Fig. 2 showing the approximate electrical resistivity of refractory brick was prepared from the results of Hartman, Sullivan and Allen (*Trans.* Am. Electrochem. Soc., vol. xxxviii, 279, 1920).

**RESISTIVITY (THERMAL).**—Thermal resistivity is the reciprocal of Thermal Conductivity. See Report of the Committee on Heat Transmission of the National Research Council on Definitions, Nomenclature, Symbols and Units for Heat Transmission.

**SLAGGING (RESISTANCE TO).**—This property is of prime importance in selection of refractory brick for service in furnaces where they are subjected to chemical corrosion by molten slags or fluxes. The resistance of a refractory brick to slag action depends on its chemical composition, density and porosity, the prevailing temperature and the chemical characteristics of the slag or flux. To resist acid slags or fluxes, fireclay brick, high-alumina brick or silica brick are serviceable. Where basic slags or fluxes are present, either fireclay, high-alumina, magnesite, or chrome brick are serviceable, the selection depending upon conditions stated above. Various simulative service tests have been devised, but as slagging depends on many factors no test as yet developed (1936) has proved sufficiently reliable to warrant its adoption as standard.

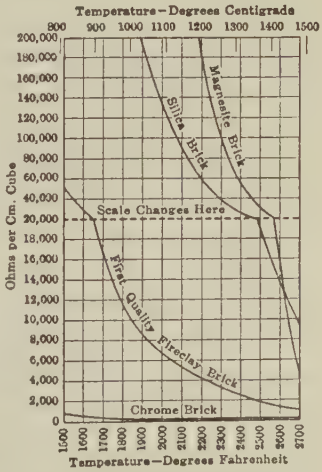


Fig. 2. Electrical Resistivity of Refractory Brick

**SPALLING, RESISTANCE TO.**—Spalling is defined as breaking or cracking of refractory brick in service to such an extent that pieces are separated or fall away, leaving new surfaces of the brick exposed. Spalling is of four types: 1. *Thermal Spalling*, caused by rapid expansion or contraction of the brick, due to rapid temperature changes. 2. *Textural Spalling* or changes in the texture or mineral constitution of the hot end of a brick, due to heat or the absorption of slags or fluxes, resulting in the formation of different zones between inner and outer brick surfaces. The physical properties of the zones may differ from each other and from the original brick, causing the inner end of the brick to crack and fall off, due to shrinkage cracks, unequal thermal expansion between zones, increased sensitiveness to temperature change, or other causes. 3. *Mechanical Spalling* or shattering of the brick by abuse in removal of slag or clinker, or by pinching due to thermal expansion or other causes. Pinch spalling is frequent in spring arches. 4. *Structural Spalling* due to improper design of brick shapes, faulty design or construction of the furnace, or excessive insulation that increases the severity of furnace conditions. Any of these may cause spalling of the first three types.

Of the five general types of refractory brick most widely used, super-duty fireclay brick have the greatest resistance to thermal spalling. High-alumina brick are only slightly less resistant. High-alumina brick may be considerably more resistant to textural spalling than other fireclay brick in the presence of certain types of slag. This is due to their high refractoriness and chemical composition which enable them to resist change in texture due to high temperature or slag action. Silica brick are extremely sensitive to temperature change in the range of 390° to 1110° F. Magnesite and chrome brick also are very sensitive to temperature change, although changes in manufacturing methods are improving this characteristic. No single method for determining resistance to spalling can be applied to all types of refractory brick. Various methods have been devised but none has been generally accepted as satisfactory. A method of control test for resistance of fireclay brick to thermal spalling is covered by A.S.T.M. Designation C38–36.

**SPECIFIC GRAVITY** of various types of refractories is given in Table 21.

For information concerning the determination of specific gravity see A.S.T.M. Designation C20–33T.

**SPECIFIC HEAT.**—The data in Table 22 are from Modern Refractory Practice, published by Harbison-Walker Refractories Co. Additional information is given in The Heat Required to Fire Ceramic Bodies by A. Ernest MacGee, *Jour.* Am. Ceramic Soc., vol. ix, p. 206 (1926) and in a supplementary article of the same title by H. E. Simpson, *Jour.* Am. Ceramic Soc., vol. x, p. 897, 1927.

## 3. MORTARS FOR LAYING REFRACTORY BRICK

Until comparatively recent years all fireclay brick were laid with fireclay mortar. It was the general opinion that the raw or unfired clay used for the purpose should have a refractoriness one to two cones lower than that of the brick with which it was used, in order that the joint material would be partially vitrified and thus bond the brick firmly together. Silica clay served for laying silica brick, finely ground dead-burned magnesite, known as furnace magnesite, for magnesite brick, and finely ground chrome ore, known as furnace chrome, for chrome brick.

**Table 21.—Specific Gravity of Various Types of Refractories**

| Type of Brick | Specific Gravity | | | Type of Brick | Specific Gravity | | |
|---|---|---|---|---|---|---|---|
| | High | Low | Average | | High | Low | Average |
| Fireclay | | | | High-alumina | Somewhat higher than for super-duty fireclay | | |
| Super-duty | 2.80 | 2.70 | 2.75 | Silica | 2.40 | 2.28 | 2.35 |
| High heat duty (Aluminous) | 2.75 | 2.65 | 2.70 | Magnesite | 3.60 | 3.44 | 3.52 |
| High heat duty (Siliceous) | Somewhat lower than for aluminous fireclay | | | Chrome | 4.4 | 3.7 | 4.05 |

**Table 22.—Specific Heats of Refractory Materials**

| Temp., deg. C. | 0 | 100 | 200 | 300 | 400 | 500 | 600 | 700 | 800 | 900 | 1000 | 1100 | 1200 | 1300 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Fireclay brick | 0.193 | 0.199 | 0.205 | 0.211 | 0.218 | 0.225 | 0.231 | 0.238 | 0.244 | 0.249 | 0.254 | 0.258 | 0.261 | 0.265 |
| Silica brick | .169 | .189 | .210 | .230 | .236 | .245 | .248 | .253 | .258 | .262 | .266 | .269 | .272 | .275 |
| Magnesite brick | .208 | .220 | .232 | .241 | .248 | .254 | .260 | .265 | .269 | .274 | .278 | .282 | .287 | .292 |
| Chrome brick | .170 | .177 | .182 | .188 | .193 | .198 | .202 | .206 | .210 | .213 | .217 | .220 | .222 | .224 |

**FIRECLAY MORTARS.**—(Raymond M. Howe, *Chem. & Met. Engg.*, Aug. 11, 1920.) Common practice in the making of firebrick mortars is to add certain materials to ground fireclay to make the mortar set firmly without the application of heat, or to cause it to burn to a dense structure by reason of decreased refractoriness. Ground silica and alumina are often added, and since clay itself is composed almost entirely of these materials, this is legitimate, and the general properties of plastic clay are often improved thereby. The addition of other materials is not justified and their effect is shown in Table 23.

The figures in this table were obtained from a series of experiments on columns and arches subjected to varying degrees of heat. The brick forming these columns and arches were laid in mortars of the compositions indicated in the table. The point of fusion given in the table is the point at which the mortar flowed between the joints.

In addition to the substances given in the table, ground bats were added to the mortar in another series of experiments, and their influence upon the fusion-point of the mortar was as follows:

| | | | | |
|---|---|---|---|---|
| Percentage of ground bats in mortar.... | 0 | 25 | 50 | 75 |
| Fusion-point, Cone No............. | 30 | 30–31 | 31 | 31–32 |
| Fusion-point, deg. C............... | 1730 | 1740 | 1750 | 1760 |

The addition of these ground bats is, in effect, an addition of clay of the same quality as that forming the mortar, but which has previously been shrunk. It accomplishes the same result in avoiding shrinkage of mortar as the addition of the substances in Table 23, but without their harmful effects.

**BONDING MORTARS.**—Until recently (1936) the term high-temperature cement designated various refractory products now known as high-temperature bonding mortars used in furnace masonry to bond refractory brick. Some of these will set and develop great strength merely upon air drying. Others do not develop any considerable strength until heated. Still others require very high temperatures to fully develop their bond. There is also considerable variation in the refractoriness of bonding mortars now on the market. They include a wide range of base materials. Consequently, their properties vary considerably. Care is necessary in selection to obtain satisfactory results.

Certain advantages over fireclay are claimed for high-temperature bonding mortars, as follows: 1. Increased resistance to erosion by mechanical action of molten metal, slag, or dust-laden gases moving at high velocity. 2. Greater resistance to chemical corrosion by slags and fluxes. 3. Increased resistance to abrasion. 4. Decreased permeability to gases, liquid metal, and slag. 5. More rigid construction. 6. Greater resistance to spalling.

## 4. COMMERCIAL FIREBRICK

A straight 9-inch firebrick weighs 7 lb.; a silica brick, 6.2 lb.; a magnesia brick, 9 lb.; a chrome brick, 10 lb.

Clay brick expand or shrink, depending upon the proportion of silica to alumina contained in the brick; but most fireclay brick contain alumina sufficient to show some shrinkage.

**Table 23.—Effect of Addition of Various Substances on Fusion-point of Fireclay Mortar**

| Clay, per-cent | Added Material, per-cent | Fusion Points | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cement | | Lime | | Asbestos | | Water Glass | | Salt | | Carborundum | |
| | | Cone No. | Deg. C. | Cone No. | Deg. C. | Cone No. | Deg. C. | Cone No. | Deg. C. | Cone No. | Deg. C. | Cone No. | Deg. C. |
| 100 | 0 | 30 | 1730 | 30 | 1730 | 30 | 1730 | 30 | 1730 | 30 | 1730 | 30 | 1730 |
| 97 | 3 | ...... | ...... | ...... | ...... | 28–29 | 1700 | ...... | ...... | ...... | ...... | ...... | ...... |
| 96 | 4 | 27 | 1670 | 20–26 | 1590 | ...... | ...... | 29–30 | 1720 | ...... | ...... | ...... | ...... |
| 95 | 5 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 26 | 1650 | 29 | 1710 |
| 94 | 6 | 20 | 1530 | ...... | ...... | 19–20 | 1520 | ...... | ...... | ...... | ...... | ...... | ...... |
| 92 | 8 | 19 | 1510 | 17–18 | 1480 | ...... | ...... | 29 | 1710 | ...... | ...... | ...... | ...... |
| 91 | 9 | ...... | ...... | ...... | ...... | 18–19 | 1500 | ...... | ...... | ...... | ...... | ...... | ...... |
| 90 | 10 | 15 | 1430 | ...... | ...... | ...... | ...... | ...... | ...... | 14 | 1410 | 29 | 1710 |
| 88 | 12 | ...... | ...... | 11 | 1350 | ...... | ...... | 28–29 | 1700 | ...... | ...... | ...... | ...... |
| 85 | 15 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 5 | 1230 | 29 | 1710 |
| 84 | 16 | ...... | ...... | 10 | 1330 | ...... | ...... | 26–27 | 1660 | ...... | ...... | ...... | ...... |
| 80 | 20 | 13 | 1390 | ...... | ...... | ...... | ...... | 26 | 1650 | ...... | ...... | 29 | 1710 |
| 76 | 24 | ...... | ...... | ...... | ...... | ...... | ...... | 20 | 1530 | ...... | ...... | ...... | ...... |
| 70 | 30 | 11 | 1350 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 29 | 1710 |
| 60 | 40 | 8 | 1290 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 29 | 1710 |
| 50 | 50 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 29 | 1710 |

One cubic foot of wall requires 17, 9-in. bricks and weighs from 125 to 140 lb. One square foot of wall requires the following bricks: 4 1/2-in. wall, 7 9-in. straight bricks; 9-in. wall, 14 bricks; 13 1/2-in. wall, 21 bricks. Closely stacked, 1000 bricks occupy 56 cu. ft.; loosely stacked, 72 cu. ft. Where keys, wedges, and other "shapes" are used, add 10%, in estimating the number required.

To obtain the best results, firebrick should be laid in the same clay from which they are manufactured. From 400 to 600 lb. of fireclay should be sufficient to lay 1000 ordinary bricks. It should be used as a thin paste and not as mortar. The thinner the joint the better the furnace wall. Silica brick should be laid in silica cement and with the smallest joint possible.

Tables 24–28 give data on the commercial sizes and applications of fireclay and silica brick and block.

## 5. MISCELLANEOUS REFRACTORIES

**SILICON CARBIDE REFRACTORIES.**—(Data supplied by the Carborundum Co., Refractory Div., Perth Amboy, N. J.) Silicon carbide is composed of carbon and silicon, the chemical formula being SiC. It is made in the electric resistance furnace and sold under trade names of Carborundum, Crystolon and Carbolon. The crystalline product forms at 3344° F. and dissociates into its elements at 4064° F. No softening or fusion occurs below the dissociation temperature. Its hardness is 9.6 on the Moh scale, giving a remarkable resistance to abrasion. It is inert to all acids, liquid or gaseous, excepting a mixture of hydrofluoric and nitric acids, but may be decomposed by alkali when subject to high temperature. Silicon carbide reacts with basic slags and molten iron and steel. It is made into refractory brick and shapes of two types known as bonded and recrystallized.

Bonded SiC refractories comprise silicon carbide crystals bonded with small percentages of permanent binders, depending on the type of service. The standard 9 × 4 1/2 × 2 1/2 in. brick weighs approximately 9.25 lb. A wide variety of special shapes are made for use in boiler, heat treating, malleable iron annealing, forge and melting furnaces, soaking pits, recuperators, electric furnaces, zinc retorts, coking and calcining retorts, gas

**Table 24.—Sizes of Straight Fireclay and Silica Brick**

(American Refractories Inst., Pittsburgh, Pa.)

| Name of Brick or Tile | Material* | Dimensions, in.† | | |
|---|---|---|---|---|
| | | Length | Width | Thickness |
| 9-in. Straight | F, S | 9 | 4 1/2 | 2 1/2, 3‡ |
| Large 9-in. | F, S | 9 | 6 3/4 | 2 1/2 |
| Small 9-in. | F, S | 9 | 3 1/2 | 2 1/2 |
| Split | F, S | 9 | 4 1/2 | 1 1/4 |
| 2-in. | F, S | 9 | 4 1/2 | 2 |
| Soap | F, S | 9 | 2 1/2 | 2 1/4 |
| Checker | F | 9 | 2 3/4 | 2 3/4 |
| " | S | 10, 10 1/2 | 4 1/2 | 4 1/2 |
| Flat Back | F | 9 | 6 | 2 1/2, 3 |
| " " Splits | F | 9 | 6 | 1 1/4 |
| Open Hearth Checker | F | 10 | 4 1/2 | 4 1/2 |
| Square Edge Tile | F | 12 | 12 | 3 |
| 12-in. Straight | S | 12 | 4 1/2, 6, 9 | 3 |
| 13 1/2-in. Straight | F, S | 13 1/2 | 6 | 2 1/2, 3 |
| " " | S | 13 1/2 | 9 | 3 |
| Bridge Block | F | 13 1/2 | 6 | 3 |
| Square Bung | F | 13 | 4 1/2 | 3 |
| 15-in. Straight | S | 15 | 6, 9 | 3 |
| Stock Hole Tile | F | 18 | 9 | 4 1/2 |
| 18-in. Straight | S | 18 | 6, 9 | 3 |
| 20-in. Straight | S | 20 | 6, 9 | 3 |
| Regenerator Tile | F | 18 | 6, 9 | 3 |
| " " | F | 18 | 9, 12 | 4 |
| " " | F | 22 1/2 | 6, 9 | 3 |
| " " | F | 22 1/2 | 9, 12 | 4 |
| " " | F | 27 | 9 | 3, 4 |
| " " | F | 27 | 12 | 4 |
| " " | F | 31 1/2 | 12 | 4 |
| " " | F | 36 | 12 | 4 |

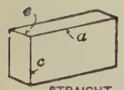
* F = Fireclay; S = Silica.

† Where two or more dimensions are given, each additional dimension indicates an additional size of brick.

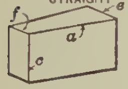
‡ Fireclay only.

producers, water gas generators, porcelain enameling furnaces, periodic and tunnel kilns, glass lehrs, rotary kilns, roasting furnaces, etc. SiC refractories are especially resistant to the destructive action of clinker formed from coal ash. Thermal conductivity of the higher grades of SiC refractories averages 108 B.t.u. per sq. ft. per 1 in. thickness per deg. F. per hr. in the temperature range 1200°–2800° F., making them suitable for indirect heating, as in muffles, retorts, recuperators, etc. Other average physical properties are: Modulus of rupture, 900 lb. per sq. in. at 2462° F.; crushing strength at 2462° F., 10,000

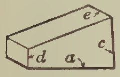
**Table 25.—Sizes and Shape of Wedge-shape and Taper Fireclay and Silica Brick**
(American Refractories Institute, Pittsburgh, Pa.)



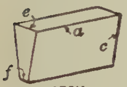
STRAIGHT



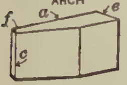
WEDGE



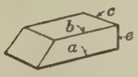
KEY



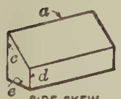
ARCH



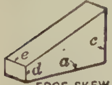
NECK



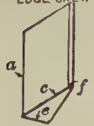
END SKEW



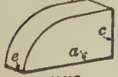
SIDE SKEW



EDGE SKEW



FEATHER EDGE



JAMB

| Name of Brick or Tile | Material* | Length, in.† | | Width, in.† | | Thickness, in. | |
|---|---|---|---|---|---|---|---|
| | | Long Edge, *a* | Short Edge, *b* | Long Edge, *c* | Short Edge, *d* | Max. *e* | Min. *f* |
| Wedge No. 1 | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 1 7/8 |
| " " 2 | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 1 1/2 |
| " " 3 | F, S | 9 | ........ | 4 1/2 | ........ | 3 | 2 |
| " Large 9-in., No. 1 | F, S | 9 | ........ | 6 3/4 | ........ | 2 1/2 | 1 7/8 |
| " " " " 2 | F, S | 9 | ........ | 6 3/4 | ........ | 2 1/2 | 1 1/2 |
| " 12-in., No. 1 | S | 12 | ........ | 4 1/2, 6‡, 9 | ........ | 3 | 2 3/4 |
| " " " 2 | S | 12 | ........ | 4 1/2, 6‡, 9 | ........ | 3 | 2 1/2 |
| " " " 3 | S | 12 | ........ | 4 1/2, 6‡, 9 | ........ | 3 | 2 |
| " 13 1/2-in., No. 1 | F, S | 13 1/2 | ........ | 6 | ........ | 3 | 2 3/4 |
| " " " 2 | F, S | 13 1/2 | ........ | 6 | ........ | 3 | 2 1/2 |
| " " " 3 | F, S | 13 1/2 | ........ | 6 | ........ | 3 | 2 |
| " " " 1 | S | 13 1/2 | ........ | 9 | ........ | 3 | 2 3/4 |
| " " " 2 | S | 13 1/2 | ........ | 9 | ........ | 3 | 2 1/2 |
| " " " 3 | S | 13 1/2 | ........ | 9 | ........ | 3 | 2 |
| " 15-in. No. 1 | S | 15 | ........ | 6, 9 | ........ | 3 | 2 3/4 |
| " " " 2 | S | 15 | ........ | 6, 9 | ........ | 3 | 2 1/2 |
| " " " 3 | S | 15 | ........ | 6, 9 | ........ | 3 | 2 |
| " 18-in. " 1 | S | 18 | ........ | 6, 9 | ........ | 3 | 2 3/4 |
| " " " 2 | S | 18 | ........ | 6, 9 | ........ | 3 | 2 1/2 |
| " " " 3 | S | 18 | ........ | 6, 9 | ........ | 3 | 2 |
| " 20-in. " 1 | S | 20 | ........ | 6, 9 | ........ | 3 | 2 3/4 |
| " " " 2 | S | 20 | ........ | 6, 9 | ........ | 3 | 2 1/2 |
| " " " 3 | S | 20 | ........ | 6, 9 | ........ | 3 | 2 |
| Arch, No. 1 | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 2 1/8 |
| " " 2 | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 1 3/4 |
| " " 3 | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 1 |
| " 12-inch No. 1 | S | 12 | ........ | 6 | ........ | 3 | 2 1/2 |
| " " " 2 | S | 12 | ........ | 6 | ........ | 3 | 2 |
| " 13 1/2-in. No. 1 | S | 13 1/2 | ........ | 6 | ........ | 3 | 2 1/2 |
| " " " 2 | S | 13 1/2 | ........ | 6 | ........ | 3 | 2 |
| Key, No. 1 | F. S | 9 | ........ | 4 1/2 | 4 | 2 1/2 | ........ |
| " " 2 | F, S | 9 | ........ | 4 1/2 | 3 1/2 | 2 1/2 | ........ |
| " " 3 | F, S | 9 | ........ | 4 1/2 | 3 | 2 1/2 | ........ |
| " " 4 | F, S | 9 | ........ | 4 1/2 | 2 1/4 | 2 1/2 | ........ |
| " 12-in., No. 1 | S | 12 | ........ | 6 | 5 1/2 | 3 | ........ |
| " " " 2 | S | 12 | ........ | 6 | 5 | 3 | ........ |
| " 9 × 6-in. No. 1 | F | 9 | ........ | 6 | 5 3/8 | 2 1/2, 3 | ........ |
| " " " 2 | F | 9 | ........ | 6 | 4 13/16 | 2 1/2, 3 | ........ |
| " 13 1/2-in. No. 1 | F | 13 1/2 | ........ | 6 | 5 | 2 1/2, 3 | ........ |
| " " " 2 | F | 13 1/2 | ........ | 6 | 4 3/8 | 2 1/2, 3 | ........ |
| Neck, No. 1 | F | 9 | 3 1/2 | 4 1/2 | ........ | 2 1/2 | 5/8 |
| " " 2 | F | 9 | 1 1/2 | 4 1/2 | ........ | 2 1/2 | 5/8 |
| " " 3 | F, S | 9 | 0 | 4 1/2 | ........ | 2 1/2 | 5/8 |
| Edge Skew | F, S | 9 | ........ | 4 1/2 | 1 1/2 | 2 1/2 | ........ |
| End " | F, S | 9 | 6 3/4 | 4 1/2 | ........ | 2 1/2 | ........ |
| Side " | F, S | 9 | ........ | 4 1/2 | 2 1/4 | 2 1/2 | ........ |
| " " 3-in. | S | 9 | ........ | 4 1/2 | 3 | 2 1/2 | ........ |
| Feather Edge | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 1/8 |
| Jamb | F, S | 9 | ........ | 4 1/2 | ........ | 2 1/2 | ........ |
| Bung Arch | F | 9 | ........ | 4 1/2 | ........ | 2 1/2 | 2 3/8 |
| Flat Back Arch No. 1 | F | 9 | ........ | 6 | ........ | 3 1/2 | 2 1/2 |
| " " " " 2 | F | 9 | ........ | 6 | ........ | 3 1/2 | 2 |
| Arch Bung No. 103 | F | 13 | ........ | 4 1/2 | ........ | 3 | 2 5/8 |
| " " " 105 | F | 13 | ........ | 4 1/2 | ........ | 3 | 2 7/8 |
| Angle Bung No. 102 | F | 11 3/8 | 12 3/4 | 4 1/2 | ........ | 3 | ........ |
| Arch Angle Bung No. 104 | F | 11 3/8 | 12 3/4 | 4 1/2 | ........ | 3 | 2 5/8 |

---

* F = Fireclay; S = Silica.

† Where two or more dimensions are given, additional dimension indicates an additional size of firebrick.

‡ Fireclay only.

lb. per sq. in.; at 68° F., 15,000 lb. per sq. in. Specific heat, 0.18. Spall resistance, higher than any other refractory.

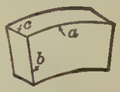
Recrystallized refractories consist of a dense mass of SiC crystals bonded together by a recrystallization process. Since these bricks contain no added bonds they have essentially the same refractoriness as silicon carbide itself. They have little tendency to crack and spall and are used in extremely high temperature ranges where great load-carrying capacity is essential.

**ACID-RESISTING BRICK.**—Chemical process industries, and other industries where various acids are manufactured or are used in large quantities, require storage tanks, towers, and other types of acid equipment for which linings and packing material with acid-resisting properties are necessary. Lead, wood, asphalt, ordinary vitrified brick, chemical stoneware and special acid-resisting refractory brick have wide application, especially acid-resisting refractory brick. Such brick are serviceable both in contact with

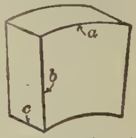
**Table 26.—Sizes and Shapes of Circular Firebrick and Blocks**

(American Refractories Institute)

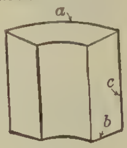
| Size and Kind | Length a, in. | Width b, in. | Thickness, c, in. | Diameter of Circle, in. | | No. of Brick to Circle |
|---|---|---|---|---|---|---|
| | | | | Outside | Inside | |
| 24-in Circle | 9 | 4 1/2 | 2 1/2 | 33 | 24 | 12 |
| 36-in. " | 9 | 4 1/2 | 2 1/2 | 45 | 36 | 16 |
| 48-in. " | 9 | 4 1/2 | 2 1/2 | 57 | 48 | 20 |
| 60-in. " | 9 | 4 1/2 | 2 1/2 | 69 | 60 | 24 |
| 72-in. " | 9 | 4 1/2 | 2 1/2 | 81 | 72 | 28 |
| 84-in. " | 9 | 4 1/2 | 2 1/2 | 93 | 84 | 32 |
| 120-in. " | 9 | 4 1/2 | 2 1/2 | 129 | 120 | 45 |
| 48-in. Rotary Kiln Block | 9 | 9 | 4 | 66 | 48 | 23 |
| 54-in. " " " | 9 | 9 | 4 | 72 | 54 | 26 |
| 60-in. " " " | 9 | 9 | 4 | 78 | 60 | 28 |
| 66-in. " " " | 9 | 9 | 4 | 84 | 66 | 30 |
| 72-in. " " " | 9 | 9 | 4 | 90 | 72 | 32 |
| 78-in. " " " | 9 | 9 | 4 | 96 | 78 | 34 |
| 84-in. " " " | 9 | 9 | 4 | 102 | 84 | 36 |
| 90-in. " " " | 9 | 9 | 4 | 108 | 90 | 38 |
| 96-in. " " " | 9 | 9 | 4 | 114 | 96 | 40 |
| 102-in. " " " | 9 | 9 | 4 | 120 | 102 | 42 |
| 108-in. " " " | 9 | 9 | 4 | 126 | 108 | 44 |
| 114-in. " " " | 9 | 9 | 4 | 132 | 114 | 46 |
| 117-in. " " " | 9 | 9 | 4 | 135 | 117 | 48 |
| 120-in. " " " | 9 | 9 | 4 | 138 | 120 | 49 |
| 123-in. " " " | 9 | 9 | 4 | 141 | 123 | 50 |
| 126-in. " " " | 9 | 9 | 4 | 144 | 126 | 51 |
| 132-in. " " " | 9 | 9 | 4 | 150 | 132 | 53 |
| 138-in. " " " | 9 | 9 | 4 | 156 | 138 | 55 |
| 144-in. " " " | 9 | 9 | 4 | 162 | 144 | 57 |
| 150-in. " " " | 9 | 9 | 4 | 168 | 150 | 59 |
| 9-in. Cupola Block A | 9 | 4 1/2 | 9 | 25 | 16 | 9 |
| " " " B | 9 | 4 1/2 | 9 | 30 | 21 | 11 |
| " " " C | 9 | 4 1/2 | 9 | 36 | 27 | 13 |
| " " " D | 9 | 4 1/2 | 9 | 39 | 30 | 14 |
| " " " E | 9 | 4 1/2 | 9 | 49 | 40 | 17 |
| " " " F | 9 | 4 1/2 | 9 | 60 | 51 | 21 |
| " " " G | 9 | 4 1/2 | 9 | 69 | 60 | 24 |
| " " " H | 9 | 4 1/2 | 9 | 82 | 73 | 29 |
| 6-in. " " 6–30 | 9 | 6 | 4 | 42 | 30 | 15 |
| " " " 6–36 | 9 | 6 | 4 | 48 | 36 | 17 |
| " " " 6–42 | 9 | 6 | 4 | 54 | 42 | 19 |
| " " " 6–48 | 9 | 6 | 4 | 60 | 48 | 21 |
| " " " 6–54 | 9 | 6 | 4 | 66 | 54 | 23 |
| " " " 6–60 | 9 | 6 | 4 | 72 | 60 | 26 |
| " " " 6–66 | 9 | 6 | 4 | 78 | 66 | 28 |
| " " " 6–72 | 9 | 6 | 4 | 84 | 72 | 30 |
| " " " 6–78 | 9 | 6 | 4 | 90 | 78 | 32 |
| " " " 6–84 | 9 | 6 | 4 | 96 | 84 | 34 |
| " " " 6–90 | 9 | 6 | 4 | 102 | 90 | 36 |
| " " " 6–96 | 9 | 6 | 4 | 108 | 96 | 38 |
| " " " 6–102 | 9 | 6 | 4 | 114 | 102 | 40 |
| " " " 6–108 | 9 | 6 | 4 | 120 | 108 | 42 |
| " " " 6–114 | 9 | 6 | 4 | 126 | 114 | 44 |
| " " " 6–120 | 9 | 6 | 4 | 132 | 120 | 46 |
| " " " 6–123 | 9 | 6 | 4 | 135 | 123 | 48 |



CIRCLE BRICK



ROTARY KILN BLOCK



CUPOLA BLOCK

### Table 27.—Number of Firebrick Required for Various Circles

| Size and Kind of Brick | Inside Diameter of Circle, Inches | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 6 | 12 | 18 | 21 | 24 | 27 | 30 | 33 | 36 | 39 | 42 | 45 | 48 | 51 | 54 | 57 | 60 | 63 | 66 | 69 | 72 | 78 | 84 | 90 | 96 | 102 | 108 | 114 | 120 | 126 | 132 | 138 | 144 | 150 |
| 24″ Circle | ... | ... | ... | ... | 12 | 9 | 6 | 3 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 36″ " | ... | ... | ... | ... | ... | 4 | 8 | 12 | 16 | 11 | 7 | 3 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 48″ " | ... | ... | ... | ... | ... | ... | ... | ... | ... | 6 | 11 | 16 | 20 | 14 | 9 | 4 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 60″ " | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 7 | 13 | 19 | 24 | 17 | 11 | 5 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 72″ " | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 8 | 15 | 22 | 28 | 21 | 14 | 7 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 84″ " | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 8 | 16 | 24 | 32 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| Total | ... | ... | ... | ... | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 9″ Arch No. 3 | 19 | 12 | 4 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 2 | ... | 15 | 30 | 38 | 34 | ... | 26 | ... | 19 | ... | 11 | ... | 4 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 1 | ... | ... | ... | ... | 8 | ... | 23 | ... | 38 | ... | 53 | ... | 68 | 76 | 76 | ... | 76 | ... | 76 | ... | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | ... |
| 9″ Straight | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 4 | ... | 11 | ... | 19 | ... | 27 | 34 | 42 | 49 | 57 | 64 | 72 | 79 | 87 | 94 | 102 | 109 | 117 | ... |
| Total | 19 | 27 | 34 | 38 | 42 | ... | 49 | ... | 57 | ... | 64 | ... | 72 | 76 | 80 | ... | 87 | ... | 95 | ... | 103 | 110 | 118 | 125 | 133 | 140 | 148 | 155 | 163 | 170 | 178 | 185 | 193 | ... |
| 9″ Key No. 4 | ... | ... | 25 | ... | 16 | ... | 9 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 3 | ... | ... | ... | ... | 13 | ... | 25 | ... | 38 | ... | 29 | ... | 21 | ... | 12 | ... | 5 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 2 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 13 | ... | 25 | ... | 38 | ... | 50 | 57 | 55 | ... | 50 | 46 | 42 | 38 | 34 | 29 | 25 | 21 | 17 | 13 | 9 | 4 | ... | ... |
| " No. 1 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 4 | ... | 13 | 21 | 29 | 38 | 46 | 55 | 63 | 71 | 80 | 88 | 96 | 105 | 113 | 113 |
| 9″ Straight | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 4 |
| Total | ... | ... | 25 | ... | 29 | ... | 34 | ... | 38 | ... | 42 | ... | 46 | ... | 50 | ... | 55 | 57 | 59 | ... | 63 | 67 | 71 | 76 | 80 | 84 | 88 | 92 | 97 | 101 | 105 | 109 | 113 | 117 |
| 9″ Wedge No. 2 | ... | ... | ... | ... | ... | 57 | 49 | ... | 38 | ... | 26 | ... | 12 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 1 | ... | ... | ... | ... | ... | ... | 11 | ... | 30 | ... | 50 | ... | 71 | ... | 91 | ... | 91 | ... | 91 | ... | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 | 91 |
| 9″ Straight | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 8 | ... | 15 | ... | 23 | 30 | 38 | 45 | 53 | 60 | 68 | 76 | 83 | 91 | 98 | 106 | 113 | 121 |
| Total | ... | ... | ... | ... | ... | 57 | 60 | ... | 68 | ... | 76 | ... | 83 | ... | 91 | ... | 99 | ... | 106 | ... | 114 | 121 | 129 | 136 | 144 | 151 | 159 | 167 | 174 | 182 | 189 | 197 | 204 | 212 |
| | Inside Diameter of Circle, Inches | | | | | | | | | | | | | | | | | | Inside Diameter of Circle, Feet | | | | | | | | | | | | | | | |
| | 48 | 54 | 60 | 66 | 72 | 78 | 84 | 90 | 96 | 102 | 108 | 114 | 120 | 126 | 132 | 135 | 138 | 144 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | ... | ... | ... |
| 13 1/2″ Key No. 2 | ... | ... | ... | ... | 52 | 48 | 42 | 37 | 33 | 28 | 23 | 18 | 12 | 7 | 2 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 1 | ... | ... | ... | ... | ... | 7 | 16 | 24 | 32 | 40 | 48 | 56 | 65 | 73 | 81 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | ... | ... | ... |
| 13 1/2″ Key Straight | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 2 | 5 | 11 | 17 | 24 | 30 | 36 | 43 | 49 | 55 | 61 | 68 | 74 | 80 | 87 | ... | ... | ... |
| Total | ... | ... | ... | ... | 52 | 55 | 58 | 61 | 65 | 68 | 71 | 74 | 77 | 80 | 83 | 85 | 87 | 90 | 96 | 102 | 109 | 115 | 121 | 128 | 134 | 140 | 146 | 153 | 159 | 165 | 172 | ... | ... | ... |
| 12″ Wedge No. 3 | 75 | 69 | 63 | 56 | 51 | 44 | 38 | 31 | 25 | 19 | 12 | 7 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 2 | ... | 13 | 25 | 38 | 50 | 63 | 75 | 88 | 101 | 113 | 126 | 138 | 151 | 144 | 138 | ... | 133 | 126 | 113 | 101 | 88 | 75 | 63 | 50 | 38 | 25 | 13 | ... | ... | ... | ... | ... | ... | ... |
| " No. 1 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 13 | 25 | ... | 37 | 50 | 75 | 100 | 126 | 151 | 176 | 201 | 226 | 251 | 276 | 302 | 302 | 302 | 302 | ... | ... | ... |
| 12″ Straight | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 12 | 25 | 37 | ... | ... | ... |
| Total | 75 | 82 | 88 | 94 | 101 | 107 | 113 | 119 | 126 | 132 | 138 | 145 | 151 | 157 | 163 | ... | 170 | 176 | 188 | 201 | 214 | 226 | 239 | 251 | 264 | 276 | 289 | 302 | 314 | 327 | 339 | ... | ... | ... |
| 13 1/2″ Wedge No. 3 | ... | 85 | 79 | 73 | 66 | 60 | 54 | 47 | 41 | 35 | 29 | 22 | 16 | 10 | 3 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| " No. 2 | ... | ... | 13 | 25 | 38 | 50 | 63 | 76 | 88 | 101 | 113 | 126 | 138 | 151 | 164 | 170 | 167 | 160 | 148 | 135 | 123 | 110 | 97 | 85 | 72 | 60 | 47 | 35 | 22 | 10 | ... | ... | ... | ... |
| " No. 1 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 6 | 19 | 44 | 69 | 94 | 120 | 145 | 170 | 195 | 220 | 245 | 270 | 295 | 320 | 340 | ... | ... | ... |
| 13 1/2″ Straight | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | 3 | ... | ... | ... |
| Total | ... | 85 | 92 | 98 | 104 | 110 | 117 | 123 | 129 | 136 | 142 | 148 | 154 | 161 | 167 | 170 | 173 | 179 | 192 | 204 | 217 | 230 | 242 | 255 | 267 | 280 | 292 | 305 | 317 | 330 | 343 | ... | ... | ... |

acids and other liquid solvents, and in conjunction with nascent gases and acid fumes at high temperatures.

**Duro**, an acid-resisting refractory brick, manufactured by Harbison-Walker Refractories Company, Pittsburgh, Pa., is dense, vitreous, non-absorbent, of low porosity, and strongly resistant to the action of solvents. It is manufactured from materials of extremely low flux content. The brick are fired at exceedingly high temperatures, which convert the constituents largely into crystalline minerals, which are insoluble in acids and corrosive acid chemical solutions, and highly resistant to concentrated alkali solutions. Duro is highly insoluble in mineral and organic acids of any degree of concentration, and at practically all working temperatures. It is well adapted for service in processes with temperatures up to 2400° F. in conjunction with nascent gases and acid fumes, and it has high mechanical strength.

**DIGESTER BRICK.**—The largest source of raw materials for the manufacture of paper pulp is soft wood, although other materials are used such as rags, straw, waste fibers, bamboo, jute, bagasse (waste sugar cane) and esparto (wild grass). Wood is transformed into paper pulp by either mechanical or chemical methods. One of the latter, known as the sulphite process, comprises boiling wood chips in sulphite liquor, which consists mainly of calcium and magnesium bisulphates dissolved in sulphurous acid. This process is accomplished in steel tanks called digesters. Formerly, digesters were lead lined, but in present practice the lining is a 2-in. thickness of Portland cement and sand against the shell and a double lining of acid-resisting brick, with a cement-sand mixture, 1-in. thick, between.

Widely used types of digester brick are such as are made by Fiske Brick and Granule Company, Boston, Mass. One type is exceedingly dense and impervious; another is of a slightly more open texture. The dense vitreous brick often is preferred for the back lining, with the more open texture brick for the face. The face brick readily becomes impregnated with solids and after a short time in service is quite impervious. The open texture type is more resistant to spalling conditions, but depending upon the practice, the dense impervious brick frequently is used to good advantage for both the face and back linings. For certain digester conditions, a still more open texture brick is made. This is exceptionally resistant to spalling but can be used to advantage only in operations where the pores become filled with organic constituents and their salts.

**FORSTERITE BRICK.**—Forsterite refractories are made from olivine rock to which magnesia is added in manufacture. Their chief constituent is forsterite ($2MgO \cdot SiO_2$), the brick analyzing about 57% MgO, 32% $SiO_2$, 6% FeO, 5% other oxides. Forsterite refractories are considered to be semi-basic, and are stable in volume at high temperatures. Their thermal conductivity is less than that of magnesite brick and approximates that of silica brick. Steam or aqueous acids will attack them.

Forsterite brick are used in ports, bridge walls, division walls, uptakes and bulkheads of basic open-hearth steel furnaces, and in exposed sections of side walls in copper reverberatory and refining furnaces. Their ability to carry load at relatively high temperatures has made them suitable within a certain temperature range for suspended and sprung arch roofs on copper reverberatory and refining furnaces and on copper holding furnaces.

**INSULATING BRICK.**—Thermal insulation is used to increase the operating efficiency of industrial furnaces. The materials available for this purpose may be classified as (1) bulk insulation, (2) insulating brick, (3) insulating refractory brick.

1. Bulk insulation, in the form of diatomaceous earth powder or expanded vermiculite, is used as an insulating backing to refractory settings, especially for irregular surfaces where brick cannot be readily fitted.

2. Insulating brick have high insulating values but are not suitable for use where they may be exposed to flame or products of combustion. They are manufactured from diatomaceous earth or other materials having inherently low thermal conductivities and are used mainly as a backing for refractory brick settings to reduce heat losses.

3. Insulating refractory brick usually are made from selected fireclays, which are given open porous structure by "chemical bloating," or by the addition of organic materials which burn out during the firing process. Insulating refractory brick can be used in contact with flame or combustion gases, but will not withstand slag action or mechanical abrasion.

Insulation of a refractory setting reduces the heat flow and raises the mean temperature of the wall. This may necessitate the use of a more refractory material for the setting and always increases the amount of heat stored in it. Where high heat capacity is undesirable, a lining of insulating refractory brick may be used in place of a combination of refractory brick and an insulating backing. Insulating refractory brick ordinarily weigh only 30 to 40% as much as refractory fireclay brick of the usual density, and their heat capacity is only 30 to 40% as great. Therefore, when conditions permit their use, they promote maximum thermal efficiency in intermittently operated furnaces. For additional information on heat insulation see Kent's Mechanical Engineers' Handbook—Power, pp. 3–54 to 3–71, forming Vol. 2 of this series.

## 6. APPLICATIONS OF REFRACTORIES

The applications of the common types of refractory brick and of insulation in the construction of furnaces are given in Table 28. In some cases the refractories specified appear to be conflicting. However, since furnace design and operating conditions vary widely, the requirements of the refractories also will vary. The recommendations given in the table are for general guidance, and manufacturers of refractories should be consulted for recommendations for specific conditions.

**Table 28.—Applications of Refractory Brick and Insulation**

Types of Brick

| | | |
|---|---|---|
| 1. Fireclay Brick | 2. High-alumina Brick | 3. Silica Brick |
| 1–A Super-duty | 2–A 50% alumina class | 4. Magnesite Brick |
| 1–B High heat duty | 2–B 60% alumina class | 5. Chrome Brick |
| 1–C Intermediate heat duty | 2–C 70% alumina class | 6. Insulation |
| 1–D Moderate heat duty | 2–D 80% alumina class | 6–A Insulating refractory brick |
| 1–E Low heat duty | | 6–B Insulating brick |

| Type of Furnace | Application | Kinds of Brick Used (alone or in combination) |
|---|---|---|
| **Air furnace** | Bottom | 1–A, 1–B |
| | Side walls | 1–A, 1–B |
| | Roof | 1–A, 1–B |
| | Stack base | 1–B |
| | Stack | 1–C, 1–D, 1–E |
| **Annealing furnace** | Lining | 1–B, 1–C, 1–D, 6–A |
| | Combustion chamber | 1–A, 1–B, 2 |
| | Backing | 1–C, 1–D, 1–E, 6–B |
| **Arches, sprung** | Complete | 1, 2, 3, 6–A |
| **Arches, suspended** | Complete | 1–A, 1–B, 2, 3, 4, 6–A |
| **Bake oven** | Combustion chamber | 1–A, 1–B, 2 |
| | Lining | 1–B, 1–C |
| | Backing | 1–C, 1–D, 1–E |
| **Blast furnace** | | |
| Copper | Crucible | 1–B, 1–C, 1–D, 4, 5 |
| | Shaft | 1–B, 1–C, 1–D |
| | Settlers | 4, 5 |
| Iron | Hearth and bosh | 1–B |
| | Inwall | 1–B |
| | Top | 1–B |
| | Piping | 1–C |
| Lead | Crucible and shaft | 1–B, 1–C, 1–D |
| **Blast furnace stove** | | |
| Iron | Lining | 1–B |
| | Well wall | 1–A, 1–B, 2 |
| | Checkers | 1–A, 1–B, 2 |
| **Boiler setting** | Combustion chamber | 1–A, 1–B, 2 |
| | Arches and side walls | 1–A, 1–B, 2 |
| | Bridge walls | 1–A, 1–B, 2 |
| | Baffles | 1–B |
| | Second pass | 1–B, 1–C, 1–D |
| | Backing | 1–C, 1–D, 1–E |
| **Carbon baking furnace** | Lining | 1–B |
| **Carbureter** | | |
| Oil gas | Lining | 1–B |
| | Checkers | 1–A, 1–B, 1–C |
| Water gas | Lining | 1–B |
| | Checkers | 1–A, 1–B, 1–C, 2 |
| **Coal gas benches** | Retorts | 3 |
| | Settings | 1–B |

Table 28.—Applications of Refractory Brick and Insulation.—*Continued*

| Type of Furnace | Application | Kinds of Brick Used (alone or in combination) |
|---|---|---|
| Coke oven | | |
| By-product | Above floor | 3, 6–A, 6–B |
| | Below floor | 1–B |
| | Regenerators | 1–B |
| | Checkers | 1–B, 3 |
| | Flues | 1–B, 1–C, 1–D |
| | Doors | 1–B, 1–C, 1–D |
| | Backing | 1–D, 6–B |
| Beehive | Floors and doors | 1–B |
| | Walls and crown | 3 |
| Converter | | |
| Copper | Lining | 4 |
| Steel | Lining | 3 |
| Crucible furnace | Lining | 1–A, 1–B, 2 |
| Cupola | Lining (except melting zone) | 1–B, 1–C |
| | Melting zone | 1–A, 1–B, 1–C, 3 |
| | Backing | 1–C, 1–D |
| Distillation furnace | | |
| Zinc | Outer walls | 1–A, 1–B |
| | Center walls | 1–B, 3 |
| | Roof | 1–A, 1–B |
| Dross furnace (non-ferrous) | Bottoms | 1–A, 1–B |
| | Side walls | 1–A, 1–B, 2 |
| | Side walls (at slag line) | 2, 4 |
| | Roof | 1–A, 1–B, 2 |
| Electric furnace | | |
| Brass | Complete lining | 1–A, 1–B, 2 |
| Steel (acid) | Bottom | 3 |
| | Side walls | 3 |
| | Door jambs | 1–A, 1–B, 3 |
| | Roof | 1–A, 1–B, 3 |
| Steel (basic) | Bottoms | 4 |
| | Side walls | 3, 4 |
| | Door jambs | 1–A, 1–B, 3 |
| | Roof | 1–A, 1–B, 3 |
| Enameling furnace | Complete lining | 1–A, 1–B |
| | Piers | 1–A, 1–B, 2 |
| Enamel frit furnace | Complete lining | 1–A, 1–B |
| Forge furnace | Lining | 1–A, 1–B, 2–A, 2–B |
| Generator | | |
| Oil gas | Complete lining | 1–A, 1–B |
| Producer gas | Complete lining | 1–B |
| Water gas | Complete lining | 1–B |
| | Wall (at clinker line) | 2–A |
| Glass lehrs | Complete lining | 1–B, 6–A |
| | Backing | 6–B |
| Glass pot furnace | Lining | 1–A, 1–B, 2 |
| | Crowns | 1–B, 3 |
| Glass tank (continuous and day) | Regenerators | 1–A, 1–B |
| | Checkers | 1–A, 1–B, 2 |
| | Ports | 1–A, 1–B, 2, 3 |
| | Breast walls | 3 |
| | Roof | 3 |
| | Bottoms and side walls | Flux blocks |
| Gypsum kettle | Complete lining | 1–A, 1–B |
| | Backing | 1–C, 1–D, 6–A, 6–B |
| Heating furnace | Bottoms and several courses in side walls | 1–A, 1–B, 4, 5 |
| | Side walls | 1–A, 1–B, 2, 6–A |
| | Roof | 1–A, 1–B, 2, 6–A |

**Table 28.**—Applications of Refractory Brick and Insulation—*Continued*

| Type of Furnace | Application | Kinds of Brick Used (alone or in combination) |
|---|---|---|
| Incinerator | | |
| Municipal | Complete lining | 1–A, 1–B |
| | Backing | 1–C, 1–D, 1–E |
| Paper mill | Lining | 1–B |
| Kilns | | |
| Cement, rotary | Burning zone | 1–B, 2, 4 |
| | Intermediate zone | 1–B |
| | Cold zone | 1–B, 1–C, 1–D |
| | Coolers | 1–C, 1–D |
| Ceramic, periodic | Lining | 1–A, 1–B |
| | Combustion chambers | 1–A, 1–B, 2 |
| | Backing | 1–C, 1–D, 1–E, 6–A, 6–B |
| Ceramic, continuous | Lining | 1–A, 1–B |
| | Burning zone (walls and arch) | 1–A, 1–B, 2, 3 |
| | Backing | 1–C, 1–D, 1–E, 6–A, 6–B |
| Dolomite, rotary | Burning zone | 1–B, 2, 4 |
| | Intermediate zone | 1–B |
| | Cold zone | 1–B, 1–C, 1–D |
| | Coolers | 1–C, 1–D |
| Lime, rotary | Burning zone | 1–B, 2, 4 |
| | Intermediate zone | 1–B |
| | Cold zone | 1–B, 1–C, 1–D |
| | Coolers | 1–C, 1–D |
| Lime, vertical shaft and pot type | Lining | 1–B |
| | Top lining | 1–B, 1–C, 1–D |
| | Hot zone and arches | 1–A, 1–B, 2, 3 |
| | Piers | 1–A, 1–B, 2, 3 |
| Ladles | | |
| Steel, tapping | Complete lining | 1–B, 1–C, 1–D, 1–E |
| Treadwell | Complete lining | 1–B |
| Litharge furnace | Bottom | 4 |
| | Lower walls | 4 |
| | Upper walls | 1–B |
| | Roof | 1–B |
| Locomotive firebox | Complete lining | 1–B |
| Malleable-iron furnace | Bottom | 1–A, 1–B |
| | Side walls | 1–A, 1–B |
| | Bungs | 1–A, 1–B |
| | Stack base | 1–B |
| | Stack | 1–C, 1–D, 1–E |
| Metal mixer | Complete lining | 1–B |
| Muffle furnace | Complete lining | 1–B |
| | Muffle | 3 |
| Open-hearth furnace | | |
| Steel (acid) | Furnace lining | 3 |
| | Checker chamber lining | 1–A, 1–B |
| | Checkers | 1–A, 1–B |
| | Top course checkers | 1–A, 1–B, 2 |
| | Backing checker chambers | 1–C, 1–D, 1–E, 6–A, 6–B |
| | Stack | 1–C, 1–D, 1–E |
| Steel (basic) | Bottom | 4, 5 |
| | Bottom walls to slag line | 4 |
| | Slopes | 4, 5 |
| | Upper front walls | 3, 4, 5 |
| | Upper back walls | 3, 4, 5 |
| | Upper end walls | 3, 4, 5 |
| | Corner walls | 4, 5 |
| | Ports | 4, 5 |
| | Water-cooled ports | 2 |
| | Bulkheads | 3, 4, 5 |
| | Checker chamber lining | 1–A, 1–B |
| | Checkers | 1–A, 1–B |
| | Top courses checkers | 1–A, 1–B, 2, 3 |
| | Backing checker chambers | 1–C, 1–D, 1–E, 6–A, 6–B |
| | Stack | 1–C, 1–D, 1–E |

**Table 28.—Applications of Refractory Brick and Insulation**—*Continued*

| Type of Furnace | Application | Kinds of Brick Used (alone or in combination) |
|---|---|---|
| Puddle furnace | Complete lining | 1–A, 1–B |
| Recuperators | Complete lining | 1–B |
| Refining furnace (non-ferrous) | Bottom | 1–B, 4 |
| | Side walls to slag line | 4 |
| | Side walls, upper | 1–A, 1–B, 2, 3 |
| | Roof | 1–A, 1–B, 2, 3 |
| Regenerators | Complete lining | 1–A, 1–B |
| | Checkers | 1–A, 1–B, 2, 3 |
| | Backing | 1–C, 1–D, 1–E, 6–A, 6–B |
| Reverberatory furnace | | |
| Copper | Side walls | 3, 4 |
| | Roof | 3, 4 |
| | Taps | 4, 5 |
| Lead | Bottom | 1–B |
| | Side walls | 1–B, 2 |
| | Side walls at slag line | 4 |
| | Roof | 1–B, 2 |
| Roasters | Complete lining | 1–B |
| Silicate of soda furnace | Bottom | 1–B |
| | Side walls | 1–A, 1–B |
| | Roof | 1–A, 1–B, 2, 3 |
| Soaking pits (steel) | Bottom | 4, 5 |
| | Lower side walls | 4, 5 |
| | Side walls | 1–A, 1–B |
| | Roof | 1–A, 1–B |
| | Backing | 1–C, 1–D, 1–E, 6–A, 6–B |
| Soda recovery furnace (Paper mill) | Bottom | 5 |
| | Walls | 1–B, 5 |
| | Backing | 1–C, 1–D, 1–E, 6–A, 6–B |
| Softening furnace. *See* Reverberatory furnace, lead. | | |
| Stacks, chimneys | Stack base | 1–B, 1–C, 1–D |
| | Lining | 1–C, 1–D, 1–E |
| Stills (oil refining) | Lining | 1–A, 1–B, 6–A, 6–B |
| | Combustion chamber, particularly at slag line | 1–A, 1–B, 2 |
| | Bridgewall | 1–A, 1–B, 2 |
| | Backing | 1–C, 1–D, 1–E |
| Superheater (water gas) | Complete lining | 1–B |
| | Checkers | 1–B, 1–C, 1–D |
| Welding furnace | Complete lining | 1–A, 1–B, 2 |
| | Bottom | 1–B, 4, 5 |
| Zinc distillation furnace. *See* Distillation furnace, zinc. | | |
| Zinc oxide furnace | Complete lining | 1–A, 1–B, 3 |

## WOOD

**PROPERTIES OF WOOD.**—The composition of absolutely dry wood is given in Bull. 73 of the U. S. Dept. of Agriculture as: Carbon, 49%; oxygen, 44%; hydrogen, 6%; ash, 1%. The proportions of these several elements vary slightly in the different varieties, and the presence of various oils and pigments will cause a variation in the heating values of the several species of as much as 22%. Heating values range from 5400 to 6800 B.t.u. per lb. For more detailed information on the composition and heating value of wood see Kent's Mechanical Engineers' Handbook—Power, p. 4–41, forming Vol. 2 of this series. Table 1 gives the specific gravity and weight of the various varieties of wood.

Table 2 gives the approximate commercial measures and weights of wood. For data on the strength of wood see pp. 7–28 and 7–37 of this book.

**CHARACTERISTICS AND USES OF THE VARIOUS WOODS.**—The woods most generally used in industry are as follows:

**Ash.**—A heavy, hard, coarse-grained wood. The white ash is compact and elastic, and is used for interior and cabinet work, but is not suitable for structural work. The red and green varieties are brittle, and are used as a substitute for the white ash.

**Balsa.**—An extremely light wood, used for heat insulation. When waterproofed, it is used for life preservers.

**Table 1.—Specific Gravity and Weight of Wood**

| Kind of Wood | Specific Gravity | | Wt., lb. per cu. ft. | Kind of Wood | Specific Gravity | | Wt., lb. per cu. ft. |
|---|---|---|---|---|---|---|---|
| | Range | Aver. | | | Range | Aver. | |
| Acacia | 0.58–0.85 | 0.72 | 45. | Hornbeam | 0.75 | 0.75 | 47. |
| Alder | .42– .68 | .54 | 34. | Juniper | .56 | .56 | 35. |
| Apple | .66– .84 | .75 | 47. | Lancewood | 0.68–1.00 | .84 | 52.4 |
| Arbor Vitæ | ...... | .55 | 34.3 | Larch | .50– .56 | .54 | 33.6 |
| Ash | .65– .85 | .75 | 47. | Laurel | .58– .72 | .65 | 40.5 |
| Aspen | .40– .42 | .41 | 25.6 | Lignum Vitæ | 1.17–1.33 | 1.25 | 78. |
| Bamboo | .31– .40 | .36 | 22.4 | Linden | 0.32–0.59 | 0.45 | 28. |
| Beech | .70– .90 | .80 | 50. | Locust | .63– .70 | .66 | 41.2 |
| Birch | .51– .77 | .64 | 40. | Mahogany | .66– .85 | .75 | 47. |
| Box | .95–1.16 | 1.05 | 65.5 | Maple | .53– .78 | .65 | 40.5 |
| Butternut | 0.38 | 0.38 | 23.7 | Mulberry | .56– .90 | .73 | 45.5 |
| Cedar | 0.31–0.45 | .38 | 23.7 | Oak, live | 0.82 | .82 | 51.2 |
| Cherry | .41– .51 | .46 | 28.7 | " white | 0.69–0.86 | .77 | 48. |
| Chestnut | .44– .48 | .46 | 28.7 | " red | 0.63 | .63 | 39.3 |
| Cork | .22– .26 | .24 | 14.9 | Osage Orange | ...... | .76 | 47.4 |
| Cypress | .43– .44 | .43 | 26.8 | Pear | 0.61–0.73 | .67 | 41.8 |
| Dogwood | .68– .77 | .72 | 45.9 | Pine, white | .39– .43 | .41 | 25.6 |
| Ebony | 1.11–1.33 | 1.22 | 71.1 | " yellow | .41– .66 | .54 | 33.6 |
| Elm | 0.51–0.66 | 0.58 | 36.2 | Poplar | .35– .50 | .42 | 26.2 |
| Fir, Balsam | 0.38 | .38 | 23.7 | Redwood | ...... | .33 | 20.6 |
| " California, white | 0.38 | .38 | 23.7 | Satinwood | 0.95 | .95 | 59.2 |
| " Douglas | 0.44–0.51 | .47 | 29.3 | Spruce | 0.32–0.48 | .40 | 25. |
| Gum | .84–1.00 | .92 | 57.4 | Sycamore | .40– .60 | .50 | 31.2 |
| Hackmatack | ...... | .59 | 36.8 | Tamarack | 0.53 | .53 | 33. |
| Hemlock | .42–0.46 | .44 | 27.4 | Teak | 0.66–0.98 | .82 | 51.2 |
| Hickory | .62– .78 | .70 | 43.7 | Walnut | .50– .67 | .64 | 40. |
| Holly | 0.60 | .60 | 37.4 | Willow | .40– .60 | .50 | 31.2 |

**Table 2.—Approximate Commercial Weights and Measures of Woods**

| Kind of Wood | Pounds of Green Logs to Scale 1000 ft., Board Measure | Lb. per 1000 ft. of Lumber, Board Measure | | Pounds per Cord (128 cu. ft.) of Seasoned Wood |
|---|---|---|---|---|
| | | Green | Dry | |
| Ash | .......... | 4600 | 3800 | .......... |
| Basswood | .......... | 4200 | 2500 | .......... |
| Beech | .......... | 5750 | 4000 | 3250 |
| Birch | .......... | 5500 | 4000 | .......... |
| Chestnut | .......... | 5000 | 2800 | 2350 |
| Elm, rock | .......... | 5400 | 4000 | 4500 |
| " soft | .......... | 4750 | 4100 | .......... |
| Hemlock | 6000– 7000 | .......... | .......... | .......... |
| " bark | .......... | .......... | .......... | 2200 |
| Hickory | .......... | 6000 | 4500 | 4500 |
| Maple, hard | .......... | 5400 | 3900 | 4500 |
| " soft | .......... | 5000 | 3300 | .......... |
| Oak, red | .......... | 5000 | 4000 | 3250 |
| " white | .......... | 5700 | 4100 | 3850 |
| Pine, Norway | 7000– 8000 | 5000 | 3000 | 2000 |
| " yellow | 8000–10000 | 5000 | 3000 | 2000 |
| " white, Michigan | 7000– 8000 | 4000 | 2500 | 2000 |
| " " Pennsylvania | 5000– 6000 | .......... | .......... | 2000 |
| Poplar | .......... | 3900 | 2800 | 2350 |
| Sycamore | .......... | 4750 | 3000 | .......... |
| Walnut | .......... | 4900 | 3800 | .......... |

**Butternut.**—A light, coarse-grained, soft, compact wood that is easily worked and takes a high polish. Used for interior finish.

**Cedar.**—The red variety is compact and brittle, but easily worked. Its characteristic odor is repellent to insects, and it, therefore, is used as a lining for vermin-proof closets and chests. The white variety is soft, compact and resistant to decay when in contact with the soil, but it is low in strength and toughness. Both varieties are used for posts, sills, fencing, and shingles.

**Chestnut.**—A coarse-grained, soft, light wood, easily worked, and weather resistant. Used for heavy construction, interior finish, railroad ties and telegraph poles.

**Cypress.**—A close-grained, durable, light, hard, brittle wood, that is easily worked and takes a high polish. Resistant to moisture. Used for trestles, docks, building construction, shingles, interior finish and cabinet work.

**Douglas Fir.**—A hard, strong, durable wood, but difficult to work. Used for all kinds of construction.

**Elm.**—A close-grained, hard, heavy, tough wood, capable of high polish. Difficult to work, and warps badly in drying. Used for car, wagon, boat and shipbuilding, and for bridge timbers, sills, ties, and furniture.

**Gum.**—A close-grained, heavy, hard, tough wood, that tends to warp and shrink in seasoning. Not resistant to weather. Takes a high polish. Is used in furniture manufacture.

**Hemlock.**—A coarse-grained, light, soft, brittle wood that splits easily, and which is not durable under exposure. Used principally for crates and packing cases.

**Hickory.**—The heaviest, hardest, strongest and toughest of all American woods. Extremely flexible, and used for bent wood work. Not suitable for building construction.

**Lignum Vitæ.**—A very heavy, hard and resinous wood, difficult to work. Used for small turnings, tool handles and pulley sheaves.

**Locust.**—A close-grained, strong, heavy, hard wood, used for posts and turnings.

**Mahogany.**—A wood that in the green condition is strong, durable and flexible, but which is brittle when dry. Resistant to dry rot and worms. Used for interior finish, furniture, pattern-work and handrails.

**Maple.**—Both white and hard maple are fine-grained, heavy, hard, strong woods, used for flooring, interior finish and furniture.

**Oak.**—The white oak is heavy, hard, strong, tough and close-grained. Will take a high polish. Is used for framed structures, interior finish, furniture, shipbuilding and railroad ties. The chestnut oak is durable in contact with soil and is used for railroad ties. Red and black oak are used for interior finish, furniture and railroad ties.

**Palmetto.**—A light wood, durable under water and resistant to teredo attacks. Used principally for piling.

**Pine.**—The white pine is soft, straight grained, light, but of relatively low strength. It is easily worked and is used for interior finish and patternwork. Yellow pine, long-leaf variety, is a heavy, hard, coarse-grain, strong wood, durable when dry and well ventilated. It is used for heavy framing timbers and floors, but is subject to rapid decay in contact with ground. The short-leaf variety varies widely in quality and is used as a substitute for the long-leaf kind. The red, or Norway pine, is a coarse-grained, compact, light, hard wood, used for all purposes of construction.

**Poplar.**—A soft, close, straight grain, brittle wood. It shrinks excessively in drying, and warps and twists. It is easily worked and is used for general carpentry work.

**Redwood.**—A coarse-grained, light, soft, brittle, and easily worked wood, which is durable in contact with soil. Used for railroad ties, fence posts, telegraph poles, etc.

**Spruce.**—The Sitka spruce is light, soft, and of medium strength. It is used for structural purposes and in airplane framework. Red spruce is a close straight-grained, light, soft wood used for general construction and framing purposes; also for piles and other submerged construction. White spruce is similar to red; is used as construction lumber.

**Tamarack.**—A wood that is similar to pine in appearance and quality, and which is used for telegraph poles, railway ties, and shipbuilding.

**Walnut (Black).**—A coarse-grained, hard, heavy, strong wood used for interior finish and cabinet work.

**PRESERVATION OF TIMBER.**—Timber is treated to inhibit decay, usually by impregnating it with creosote. The creosote can be applied by soaking the wood in an open tank of creosote, or by subjecting it to a vacuum and pressure process which causes the creosote to penetrate the fibers more thoroughly. The quantity of creosote required to preserve timbers is approximately as follows: Hardwood railroad ties, 6–14 lb. per cu. ft.; softwood railroad ties, 13–20 lb. per cu. ft.; dimension timbers, 15–20 lb. per cu. ft.; piles, 20–30 lb. per cu. ft.

Zinc chloride sometimes is used as a substitute for creosote. Timbers treated with zinc chloride cannot be used in exposed locations or for submerged construction.

**Table 3.—Properties of American Standard Yard Lumber and Timber Sizes**

| Nominal Size in inches | American Standard Dressed Size, inches | Area of Section, sq. in. | Weight per lin. ft.,* lb. | Moment of Inertia, $I$, in.$^4$ | Section-Modulus, $S$, in.$^3$ |
|---|---|---|---|---|---|
| 2 × 4 | 1 5/8 × 3 5/8 | 5.89 | 1.6 | 6.45 | 3.56 |
| 2 × 6 | 1 5/8 × 5 5/8 | 9.14 | 2.5 | 24.10 | 8.57 |
| 2 × 8 | 1 5/8 × 7 1/2 | 12.19 | 3.4 | 57.13 | 15.32 |
| 2 × 10 | 1 5/8 × 9 1/2 | 15.44 | 4.3 | 116.09 | 24.44 |
| 2 × 12 | 1 5/8 × 11 1/2 | 18.69 | 5.2 | 205.94 | 35.82 |
| 2 × 14 | 1 5/8 × 13 1/2 | 23.62 | 6.5 | 333.15 | 49.36 |
| 2 × 16 | 1 5/8 × 15 1/2 | 25.18 | 7.0 | 504.24 | 65.07 |
| 2 × 18 | 1 5/8 × 17 1/2 | 28.43 | 7.9 | 725.71 | 82.94 |
| 2 × 20 | 1 5/8 × 19 1/2 | 31.69 | 8.8 | 1,004.05 | 102.98 |
| 3 × 4 | 2 5/8 × 3 5/8 | 9.51 | 2.6 | 10.42 | 5.75 |
| 3 × 6 | 2 5/8 × 5 5/8 | 14.76 | 4.2 | 38.93 | 13.84 |
| 3 × 8 | 2 5/8 × 7 1/2 | 19.68 | 5.7 | 92.28 | 24.60 |
| 3 × 10 | 2 5/8 × 9 1/2 | 24.93 | 7.2 | 187.55 | 39.48 |
| 3 × 12 | 2 5/8 × 11 1/2 | 30.18 | 8.8 | 332.69 | 57.86 |
| 3 × 14 | 2 5/8 × 13 1/2 | 35.43 | 10.3 | 538.21 | 79.73 |
| 3 × 16 | 2 5/8 × 15 1/2 | 40.68 | 11.3 | 814.60 | 105.11 |
| 3 × 18 | 2 5/8 × 17 1/2 | 45.94 | 12.8 | 1,172.36 | 133.98 |
| 3 × 20 | 2 5/8 × 19 1/2 | 51.19 | 14.21 | 1,622.00 | 166.36 |
| 4 × 4 | 3 5/8 × 3 5/8 | 13.14 | 3.6 | 14.38 | 7.94 |
| 4 × 6 | 3 5/8 × 5 5/8 | 20.39 | 5.7 | 53.76 | 19.11 |
| 4 × 8 | 3 5/8 × 7 1/2 | 27.18 | 7.5 | 127.44 | 33.98 |
| 4 × 10 | 3 5/8 × 9 1/2 | 34.43 | 9.6 | 258.99 | 54.52 |
| 4 × 12 | 3 5/8 × 11 1/2 | 41.68 | 11.6 | 459.42 | 79.90 |
| 4 × 14 | 3 5/8 × 13 1/2 | 48.93 | 13.6 | 743.23 | 110.11 |
| 4 × 16 | 3 5/8 × 15 1/2 | 56.18 | 15.6 | 1,124.90 | 145.15 |
| 4 × 18 | 3 5/8 × 17 1/2 | 63.43 | 17.6 | 1,618.96 | 185.02 |
| 4 × 20 | 3 5/8 × 19 1/2 | 70.69 | 19.6 | 2,239.88 | 229.73 |
| 6 × 6 | 5 1/2 × 5 1/2 | 30.25 | 8.4 | 76.25 | 27.73 |
| 6 × 8 | 5 1/2 × 7 1/2 | 41.25 | 11.4 | 193.35 | 51.56 |
| 6 × 10 | 5 1/2 × 9 1/2 | 52.25 | 14.5 | 329.96 | 82.73 |
| 6 × 12 | 5 1/2 × 11 1/2 | 63.25 | 17.5 | 697.06 | 121.23 |
| 6 × 14 | 5 1/2 × 13 1/2 | 74.25 | 20.6 | 1,127.66 | 167.06 |
| 6 × 16 | 5 1/2 × 15 1/2 | 85.25 | 23.6 | 1,706.76 | 220.22 |
| 6 × 18 | 5 1/2 × 17 1/2 | 96.25 | 26.7 | 2,456.36 | 280.73 |
| 6 × 20 | 5 1/2 × 19 1/2 | 107.25 | 29.8 | 3,398.46 | 348.53 |
| 6 × 22 | 5 1/2 × 21 1/2 | 118.25 | 32.8 | 4,555.05 | 423.76 |
| 8 × 8 | 7 1/2 × 7 1/2 | 56.25 | 15.6 | 263.67 | 70.31 |
| 8 × 10 | 7 1/2 × 9 1/2 | 71.25 | 19.8 | 535.85 | 112.81 |
| 8 × 12 | 7 1/2 × 11 1/2 | 86.25 | 23.9 | 950.55 | 165.31 |
| 8 × 14 | 7 1/2 × 13 1/2 | 101.25 | 28.0 | 1,537.73 | 227.81 |
| 8 × 16 | 7 1/2 × 15 1/2 | 116.25 | 32.0 | 2,327.42 | 300.31 |
| 8 × 18 | 7 1/2 × 17 1/2 | 131.25 | 36.4 | 3,349.60 | 382.81 |
| 8 × 20 | 7 1/2 × 19 1/2 | 146.25 | 40.6 | 4,634.30 | 475.31 |
| 8 × 22 | 7 1/2 × 21 1/2 | 161.25 | 44.8 | 6,211.48 | 577.81 |
| 8 × 24 | 7 1/2 × 23 1/2 | 176.25 | 48.9 | 8,111.17 | 690.31 |
| 10 × 10 | 9 1/2 × 9 1/2 | 90.25 | 25.0 | 678.75 | 142.89 |
| 10 × 12 | 9 1/2 × 11 1/2 | 109.25 | 30.3 | 1,204.01 | 209.39 |
| 10 × 14 | 9 1/2 × 13 1/2 | 128.25 | 35.6 | 1,947.78 | 288.56 |
| 10 × 16 | 9 1/2 × 15 1/2 | 147.25 | 40.9 | 2,948.04 | 380.39 |
| 10 × 18 | 9 1/2 × 17 1/2 | 166.25 | 46.1 | 4,242.80 | 484.89 |

* Based on assumed average weight of 40 lb. per cu. ft.

*Continued on page 5–24*

**Table 3.—Properties of American Standard Yard Lumber and Timber Sizes**—*Continued*

| Nominal Size in inches | American Standard Dressed Size, inches | Area of Section, sq. in. | Weight per lin. ft.,* lb. | Moment of Inertia, $I$, in.$^4$ | Section-Modulus, $S$, in.$^3$ |
|---|---|---|---|---|---|
| 10 × 20 | 9 1/2 × 19 1/2 | 185.25 | 51.4 | 5,870.05 | 602.06 |
| 10 × 22 | 9 1/2 × 21 1/2 | 204.25 | 56.7 | 7,867.81 | 731.89 |
| 10 × 24 | 9 1/2 × 23 1/2 | 223.25 | 62.0 | 10,274.06 | 874.39 |
| 10 × 26 | 9 1/2 × 25 1/2 | 242.25 | 67.3 | 13,126.81 | 1029.56 |
| 10 × 28 | 9 1/2 × 27 1/2 | 261.25 | 72.5 | 16,465.24 | 1197.39 |
| 10 × 30 | 9 1/2 × 29 1/2 | 280.25 | 77.8 | 20,323.79 | 1377.89 |
| 12 × 12 | 11 1/2 × 11 1/2 | 132.25 | 36.7 | 1,457.50 | 253.47 |
| 12 × 14 | 11 1/2 × 13 1/2 | 155.25 | 43.1 | 2,357.85 | 349.31 |
| 12 × 16 | 11 1/2 × 15 1/2 | 178.25 | 49.5 | 3,568.70 | 460.48 |
| 12 × 18 | 11 1/2 × 17 1/2 | 201.25 | 55.9 | 5,136.49 | 586.98 |
| 12 × 20 | 11 1/2 × 19 1/2 | 224.25 | 62.3 | 7,105.90 | 728.81 |
| 12 × 22 | 11 1/2 × 21 1/2 | 247.25 | 68.7 | 9,524.24 | 885.98 |
| 12 × 24 | 11 1/2 × 23 1/2 | 270.25 | 75.0 | 12,437.08 | 1058.47 |
| 12 × 26 | 11 1/2 × 25 1/2 | 293.25 | 81.4 | 15,890.42 | 1246.31 |
| 12 × 28 | 11 1/2 × 27 1/2 | 316.25 | 87.8 | 19,932.58 | 1449.47 |
| 12 × 30 | 11 1/2 × 29 1/2 | 339.25 | 94.2 | 24,602.61 | 1667.97 |
| 14 × 14 | 13 1/2 × 13 1/2 | 182.25 | 50.6 | 2,767.92 | 410.06 |
| 14 × 16 | 13 1/2 × 15 1/2 | 209.25 | 58.1 | 4,189.36 | 540.56 |
| 14 × 18 | 13 1/2 × 17 1/2 | 236.25 | 65.6 | 6,029.29 | 689.06 |
| 14 × 20 | 13 1/2 × 19 1/2 | 263.25 | 73.1 | 8,341.73 | 855.56 |
| 14 × 22 | 13 1/2 × 21 1/2 | 290.25 | 80.6 | 11,180.67 | 1040.06 |
| 14 × 24 | 13 1/2 × 23 1/2 | 317.25 | 88.1 | 14,600.10 | 1242.56 |
| 14 × 26 | 13 1/2 × 25 1/2 | 344.25 | 95.6 | 18,654.04 | 1463.06 |
| 14 × 28 | 13 1/2 × 27 1/2 | 371.25 | 103.1 | 23,398.73 | 1701.56 |
| 14 × 30 | 13 1/2 × 29 1/2 | 398.25 | 110.6 | 28,881.42 | 1958.06 |
| 16 × 16 | 15 1/2 × 15 1/2 | 240.25 | 66.7 | 4,809.98 | 620.64 |
| 16 × 18 | 15 1/2 × 17 1/2 | 271.25 | 75.3 | 6,922.49 | 791.14 |
| 16 × 20 | 15 1/2 × 19 1/2 | 302.25 | 83.9 | 9,577.50 | 982.31 |
| 16 × 22 | 15 1/2 × 21 1/2 | 333.25 | 92.5 | 12,837.00 | 1194.14 |
| 16 × 24 | 15 1/2 × 23 1/2 | 364.25 | 101.2 | 16,763.00 | 1426.64 |
| 16 × 26 | 15 1/2 × 25 1/2 | 395.25 | 109.8 | 21,417.50 | 1679.81 |
| 16 × 28 | 15 1/2 × 27 1/2 | 426.25 | 118.4 | 26,863.78 | 1953.64 |
| 16 × 30 | 15 1/2 × 29 1/2 | 457.25 | 127.0 | 33,159.98 | 2248.14 |
| 18 × 18 | 17 1/2 × 17 1/2 | 306.25 | 85.0 | 7,815.73 | 893.23 |
| 18 × 20 | 17 1/2 × 19 1/2 | 341.25 | 94.8 | 10,813.33 | 1109.06 |
| 18 × 22 | 17 1/2 × 21 1/2 | 376.25 | 104.5 | 14,493.43 | 1348.23 |
| 18 × 24 | 17 1/2 × 23 1/2 | 411.25 | 114.2 | 18,926.02 | 1610.72 |
| 18 × 26 | 17 1/2 × 25 1/2 | 446.25 | 123.9 | 24,181.11 | 1896.56 |
| 18 × 28 | 17 1/2 × 27 1/2 | 481.25 | 133.7 | 30,331.62 | 2205.72 |
| 18 × 30 | 17 1/2 × 29 1/2 | 516.25 | 143.4 | 37,438.79 | 2538.22 |
| 20 × 20 | 19 1/2 × 19 1/2 | 380.25 | 105.6 | 12,049.49 | 1235.81 |
| 20 × 22 | 19 1/2 × 21 1/2 | 419.25 | 116.4 | 16,149.86 | 1502.31 |
| 20 × 24 | 19 1/2 × 23 1/2 | 458.25 | 127.3 | 21,089.04 | 1794.81 |
| 20 × 26 | 19 1/2 × 25 1/2 | 497.25 | 138.1 | 26,944.73 | 2113.31 |
| 20 × 28 | 19 1/2 × 27 1/2 | 536.25 | 148.9 | 33,798.17 | 2457.81 |
| 20 × 30 | 19 1/2 × 29 1/2 | 575.25 | 159.8 | 41,717.61 | 2828.31 |
| 24 × 24 | 23 1/2 × 23 1/2 | 552.25 | 153.4 | 25,414.96 | 2162.97 |
| 24 × 26 | 23 1/2 × 25 1/2 | 599.25 | 166.4 | 32,471.80 | 2546.81 |
| 24 × 28 | 23 1/2 × 27 1/2 | 646.25 | 179.5 | 40,731.06 | 2916.97 |
| 24 × 30 | 23 1/2 × 29 1/2 | 693.25 | 192.5 | 50,274.98 | 3408.47 |

*Based on assumed average weight of 40 lb. per cu. ft.

**KILN DRYING OF LUMBER.**—See Kent's Mechanical Engineers' Handbook—Power, p. 3–52, forming Vol. 2 of this series and also pp. 22–17 to 22–21 of this book.

**YARD LUMBER AND TIMBER SIZES.**—Table 3 gives the dressed size of American standard yard lumber and timber, together with some of the properties.

**PLYWOOD** consists of an odd number of thin layers of wood glued together, the grain in the alternate layers being at right angles. A characteristic of plywood is that the strength is approximately equal in all directions, and is approximately the mean of the strength with and across the grain. See Table 4 for the tensile strength of plywood. Plywood is made usually from basswood, birch, maple, poplar, redwood and red gum.

**Table 4.—Tensile Strength of Plywood**

(U. S. Forest Products Laboratory, Madison, Wis.)

| Species | Weight of Plywood, Kiln-dry, lb. per cu. ft. | Moisture at Test, percent | Tensile Strength of 3-ply Plywood Parallel to Grain of Faces lb. per sq. in. | Tensile Strength of Single-ply Veneer, Parallel to Grain, lb. per sq. in. |
|---|---|---|---|---|
| Basswood | 28 | 9.2 | 6,900 | 10,300 |
| Yellow Birch | 45 | 8.5 | 13,200 | 19,800 |
| Maple, soft | 38 | 8.9 | 8,200 | 12,300 |
| " hard | 46 | 8.0 | 10,200 | 15,300 |
| Yellow poplar | 34 | 3.4 | 7,400 | 11,100 |
| Red gum | 36 | 8.7 | 7,800 | 11,800 |
| Redwood | 28 | 9.7 | 4,800 | 7,200 |

## CORK

Cork is the spongy outer bark of the cork oak tree, which grows principally in Spain, Portugal and Northern Africa. It is of cellular structure, each cell comprising a strong, impenetrable skin, enclosing air.

**NATURAL CORK.**—The uses of natural cork are limited by the variability of the raw material, by structural flaws, which are uncontrollable in a natural material, and by the irregularity in size, thickness and shape. These limitations can be overcome by laminating the corkwood to produce a material of greater uniformity and structural strength.

**CORK COMPOSITIONS.**—Finely ground cork combined with various binders form materials having all the desirable basic qualities of natural cork. Composition can be made to provide for expansion or contraction, and to be fabricated into sheets, discs, tubes, blocks, rings, strips, balls, wheels, rods and irregularly shaped articles. It also can be molded directly to the desired pattern. Composition is laminated with cloth, vulcanized fiber, or other material to increase the tensile strength. Cork also has been combined with rubber and synthetic resins.

**PROPERTIES OF CORK.—1. Resistance to Liquid Penetration.**—Cork is highly resistant to penetration by liquids. Since it is not fibrous, liquids are not drawn through it by capillarity.

**Compressibility.**—Prolonged compression, under heavy loads, will cause little permanent deformation. Eight small pieces of cork composition $^7/_8$ in. thick, used as cushions between journal box and the car body of railway cars, supporting approximately 120,000 lb., after one year were still resilient, with a reduction of total thickness of only $^1/_8$ in.

**Resilience.**—Long subjection to pressure does not destroy the structure or resiliency of cork. It will not harden or deteriorate under normal conditions.

**Buoyancy.**—Natural cork, with a specific gravity of 0.15 to 0.20, is generally used for floats or other devices requiring buoyancy.

**Thermal Conductivity.**—The air-cell structure of cork forms an excellent insulator. It is highly resistant to charring and ignition. At about 250° F., slow distillation begins and the cork discolors slightly. It does not ignite except in contact with flame, and it burns only as long as flame is applied from an external source. Combustion produces a carbonized coating which retards further burning.

**Chemical Inertness.**—Cork resists chemical action better than most natural materials, and can be safely used in contact with most chemicals. In industrial processes it does not corrode, discolor, or deteriorate and is not affected by oil, grease, gasoline, or other petroleum products. Alkalis, certain acids and organic solvents have a destructive action, and, as a rule, cork should not be used in contact with them. Iron solutions sometimes react with the natural tannin in the cork to cause the surface discoloration, which does not affect the properties of the cork.

**Stability.**—Cork retains its initial properties practically unimpaired under all conditions of temperature, humidity or other atmospheric conditions. Its compressibility and resilience enable it to withstand abrasive wear.

**Frictional Properties.**—The coefficient of friction of natural cork and compositions is higher than that of most materials used for friction. This coefficient is not materially decreased by the presence of moisture, oil or grease. Compositions are more extensively used, as it is possible to control, within reasonable limits, their coefficient of friction. The softer compositions have the higher coefficients of friction, but dense materials for severe conditions have a sufficiently high coefficient for ordinary work. See Table 1 for coefficient of friction of cork and cork compositions.

**Table 1.–Angle of Repose and Coefficient of Friction of Leather and Cork**

(Armstrong Cork Co.)

| | Angle of Repose, $\theta$ | Coefficient of Friction, $f$ | $1 \div f$ |
|---|---|---|---|
| Leather on oak | 15°–19 1/2° | 0.27–0.38 | 3.7–2.86 |
| " on steel (dry) | 29 1/2° | 0.56 | 1.79 |
| " on steel (wet) | 20° | .36 | 2.78 |
| " on steel (greasy) | 13° | .23 | 4.35 |
| " on steel (oily) | 8 1/2° | .15 | 6.67 |
| 243 fine cork composition on cork | 40° | 0.838 | 1.19 |
| " " " on pine (with grain) | 30 1/2° | .584 | 1.71 |
| " " " on glass | 44 1/2° | .987 | 1.01 |
| " " " on steel (dry) | 27° | .514 | 1.95 |
| " " " on steel (wet) | 38 1/2° | .794 | 1.26 |
| " " " on steel (cork water-soaked) | 40 1/2° | .894 | 1.18 |
| " " " on steel (greasy—steel oiled) | 29° | .549 | 1.82 |
| " " " on steel (oily—cork oil-soaked) | 28° | .527 | 1.90 |
| " " " on steel (so hot as to char cork) | 32 1/2° | .640 | 1.56 |
| Acco cork composition on pine (with grain) | 23 1/2° | 0.430 | 2.32 |
| " " " on glass | 50° | 1.172 | 0.85 |
| " " " on steel (dry) | 22 1/2° | 0.417 | 2.40 |
| " " " on steel (wet) | 30 1/2° | .584 | 1.71 |
| " " " on steel (cork water-soaked) | 36° | .723 | 1.38 |
| " " " on steel (greasy—steel oiled) | 24 1/2° | .451 | 2.22 |
| " " " on steel (oily — cork oil-soaked) | 25° | .458 | 2.18 |
| " " " on steel (so hot as to char cork) | 32 1/2° | .637 | 1.57 |
| Natural cork on cork | 30 1/2° | 0.592 | 1.69 |
| " " on pine (with grain) | 26° | .487 | 2.06 |
| " " on glass | 27 1/2° | .517 | 1.93 |
| " " on steel (dry) | 24 1/2° | .452 | 2.22 |
| " " on steel (wet) | 35° | .694 | 1.44 |
| " " on steel (cork water-soaked) | 29 1/2° | .565 | 1.77 |
| " " on steel (greasy—steel oiled) | 24° | .447 | 2.24 |
| " " on steel (oily—cork oil-soaked) | 23° | .419 | 2.38 |
| " " on steel (so hot as to char cork) | 33° | .645 | 1.55 |

**USES.**—The following paragraphs give the principal uses of cork and cork products:

**Seals.**—Gaskets for sealing and cushioning, and for sealing anti-friction bearings. While cork closures are sometimes made as split rings, the continuous ring generally is used. Cork seals should not be distorted either in assembling or in use. They are not recommended for assembly in internal grooves and must be lubricated before assembly, with graphite grease thinned with lubricating oil to the consistency of vaseline. Cork gaskets should not be used on steam or other high-temperature connections or in contact with strong alkaline solutions, such as ammonia.

**Cushioning** between adjoining surfaces of metal and glass; metal and porcelain or other similar combinations; automobile bodies; to deaden vibration in machinery.

**Friction.**—In the textile industry for drawing yarn and cloth through machines; friction drives; clutches; feed rolls for machines handling paper, cloth, foil, etc.

**Floats.**—Gage mechanisms; carburetors; dispensing pumps for oil and gasoline.

**Insulation.**—Under roofs to reduce heat losses and to lessen condensation and eliminate ceiling drip; pipe covering for drinking water, brine and ammonia lines, and cold lines in the oil refining and chemical industries.

**Natural Cork Products.**—Bottle corks; wheels for polishing glass; floats for carburetors and gages; handles for tools and instruments; discs, washers, and gaskets for various applications; life preservers; yacht fenders; cork balls; paper.

## RUBBER *

Rubber is used in industry for machinery mountings, for bumpers and springs, for the absorption of vibration, for the lining of tanks, and many other purposes.

**PHYSICAL PROPERTIES.**—The physical properties of various grades of rubber are given in Table 1. The heat of combustion of crude rubber is 19,300 B.t.u. per lb. The electrical resistivity depends on temperature and the character of the rubber stock, decreasing perhaps several thousand times with increase of temperature from room temperature to 212° F. Depending on the rubber stock, it may range from $1 \times 10^8$ to $200 \times 10^8$ megohm per sq. cm. per cm. length. The dielectric constant (air = 1) ranges usually from 2.5 to 4.5. The electric breakdown stress ranges from 1 to 60 kilovolts per millimeter thickness. Rubber should not be mechanically worked at temperatures above 180° F.

**Table 1.—Physical Properties of Rubber**

All values based on room temperature

| Stock Number | 5140 | 5488 | 5638 | 5255 | 5691 | 5623 |
|---|---|---|---|---|---|---|
| Property | | | | | | |
| Hardness (Shore Durometer) | 30 | 40 | 50 | 60 | 70 | 80 |
| Elastic modulus for static compression, lb. per sq. in. | 185 | 270 | 375 | 550 | 750 | 1200 |
| Shear modulus (static), lb. per sq. in. | 55 | 83 | 110 | 132 | 190 | .... |
| Volume compression at 1000 lb. per sq. in. pressure, percent | 0.34 | 0.31 | 0.29 | 0.27 | 0.24 | 0.22 |
| Specific gravity | 1.01 | 1.06 | 1.11 | 1.18 | 1.25 | 1.31 |
| Specific heat | 0.47 | 0.43 | 0.40 | 0.38 | 0.35 | 0.33 |
| Thermal conductivity in B.t.u. per sq. ft. per sec. for a temp. gradient of 1° F. per in. | 0.00027 | 0.00029 | 0.00030 | 0.00032 | 0.00035 | 0.00037 |
| Cubical expansion with temperature, percent change in unit volume per deg. F. | 0.00033 | 0.00029 | 0.00026 | 0.00024 | 0.00022 | 0.00020 |
| Velocity of sound in rubber rods, ft. per sec. | 115 | 165 | 210 | 345 | 750 | .... |
| Shrinkage in linear dimensions of the cured rubber piece* | 2.2 | 2.0 | 1.7 | 1.5 | 1.4 | 1.3 |

* With respect to its mold cavity, both at room temperature.

## 1. ACTION OF RUBBER UNDER STRESS

In determining the action of rubber under stress, the fatigue characteristics and working temperature must be considered as well as the load. In general, compression strain† should not exceed 15 to 20%, and with hard stock should not exceed 5%.

**ELASTIC MODULUS.**—Rubber does not exactly follow Hooke's law (see p. 7–04). The elastic modulus in compression may be approximately expressed as

$$E = (F/A_o)\{h/(h_o - h)\} = (F/A)\{h_o/(h_o - h)\} \qquad [1]$$

where $E$ = elastic modulus in compression, lb. per sq. in.; $F$ = compression load, lb.; $A_o$ and $A$ = respectively, original no-load and compressed cross-sectional area, sq. in.; $h_o$ and $h$ = respectively original and compressed height, in., under load $F$. These relations apply only to blocks where the rubber can expand freely in a direction perpendicular to the direction of the applied load, *i.e.*, to blocks whose faces have been lubricated with glycerine or soap solution. For soft stocks (30 Shore durometer) these relations will hold up to 50% compression; for hard stocks (80 Shore durometer) they probably are incorrect beyond 15% compression. In determining $E$, the block must be subjected to at least five complete compression cycles before readings are taken.

**Relation of Elastic Modulus in Compression to Hardness.**—Hardness is measured by means of the Shore durometer, which gives the ratio of a constant indenting force to the indentation of a small flat-tipped tapering needle. The readings are given in terms of an arbitrary scale, running with increasing hardness from 0 to 100. The relation of durometer hardness to elastic modulus is only approximate, as the reading of the former often is a function of the observer and the size of test piece. Further, elastic modulus will vary with the calendar direction of the grain, which variation may be as large as 10%. Table 2 shows the approximate relation between durometer hardness and elastic modulus in compression and in shear for several grades of rubber.

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* The data herein given are abstracted, by permission, from a treatise entitled The Physical Properties of Rubber, issued by U. S. Rubber Products, Inc., Detroit.

† Strain, as here used, is the ratio of deflection under stress to the original thickness.

Table 2.—Relation between Durometer Hardness and Elastic Modulus

| Stock No. | 5140 | 5488 | 5638 | 5255 | 5691 | 5623 |
|---|---|---|---|---|---|---|
| Durometer hardness | 30 ± 3 | 40 ± 4 | 50 ± 4 | 60 ± 4 | 70 ± 4 | 80 ± 4 |
| Elastic modulus in compression, lb. per sq. in. | 185 ± 30 | 270 ± 40 | 375 ± 50 | 550 ± 60 | 750 ± 100 | 1200 ± 300 |
| Elastic modulus in shear, lb. per sq. in. | 55 ± 7 | 83 ± 10 | 110 ± 15 | 132 ± 20 | 190 ± 25 | ...... |

**DECREASE OF VOLUME OF RUBBER UNDER COMPRESSION** is shown in Fig. 1 together with the decrease in volume of water. The values shown are those at 77° F.

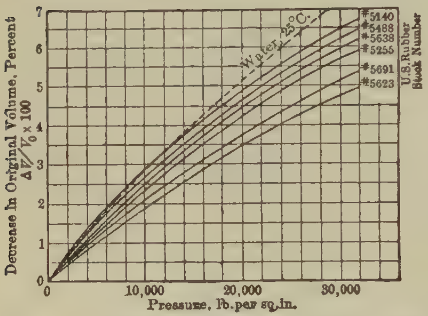

Fig. 1. Decrease of Volume of Rubber under Compression

**RUBBER SANDWICHES UNDER COMPRESSION.**—A rubber sandwich is a rubber block with two opposite faces adhered to metal plates. The compression under any load acting across the plates is a function of the dimensions and shape of the block, the elastic modulus of the rubber, the distribution of load over the surface of the block, the temperature, the amount of non-parallelism of the metal plate, the distance the plates extend beyond the side surfaces of the rubber and other variables. In general, sandwiches of the same relative dimensions and same elastic moduli are deflected the same percentage of their thickness by the same compression loads per unit area. Usually, it is not feasible to load a sandwich with more than 750 lb. per sq. in. static load, or to apply a load producing more than 20% deflection. An approximate value of the amount of compression may be obtained by means of the nomographic chart, Fig. 2.

**Use of the Nomographic Chart.**—Let $L$ = length, $W$ = width, $T$ = thickness, $F$ = total load, lb., and $P$ = unit pressure, lb. per sq. in., on a rubber sandwich. Draw a line from the point on the deflection scale representing the desired deflection, to the point on the pounds per square inch scale, corresponding to $P = F/A$, intersecting the line XX at $y$. Connect the points on the $L/W$ and $W/T$ scales corresponding to the ratios $L/W$ and $W/T$ of the sandwich under consideration, and project the line to the $Q/T$ scale. From the point so found on the $Q/T$ scale, draw a line through $y$ to intersect the $E$ scale, to find the modulus of elasticity in compression.

$Q/T$ is the ratio of the length of side of the sandwich under consideration to the thickness of a square sandwich which will be compressed the same percentage of its thickness by the same load per unit area. $Q$ is the length in the nomogram of a non-rectangular sandwich corresponding to $L$. Values of $Q$ for various shapes of sandwich are as follows.

**Solid Sandwiches.**—*Square* sandwich, $Q = W$; *rectangular* sandwich, $Q = L$; *cylindrical* sandwich of *circular* cross-section, $Q = 0.93 \times$ diam.; *cylindrical* sandwich of *elliptical* cross-section, major axis corresponds to nomographic $L$ and minor axis to nomographic $W$; *triangular* sandwich, $Q$ = diam. of inscribed circle.

**Doughnut Sandwiches** (sandwiches with hole).—*Square doughnut*, $Q = S - s$, where $S$, $s$ = respectively, length of a side of the sandwich, and the side or diam. of the hole; this relation holds for a circular hole whose diameter is small compared to the size of the sandwich.

*Circular doughnut* with circular hole, $Q = D - d$, where $D$, $d$ = respectively, diameters of sandwich and of hole.

*Rectangular* or *elliptical* sandwiches with central hole,

$$Q = Q_n - (W/L)d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

where $Q_n$ = value of $Q$ for a solid sandwich and $d$ = diam. or side of hole.

For a hole not in the center of the sandwich,

$$Q = Q_n - (W/L)\{r/(r + y)\}d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where $r$ = distance from center of hole to edge of sandwich on a line passing through center of sandwich, and $y$ = distance from center of hole to center of sandwich, other notation as before. Relation [3] holds only for values of $d < (W/2)$.

The nomographic chart will give values of compression loads for a given stock number that will be accurate to within 25% of the deflection for values of $Q/T < 12$. For values of $Q/T > 12$, it is accurate only to within 50%.

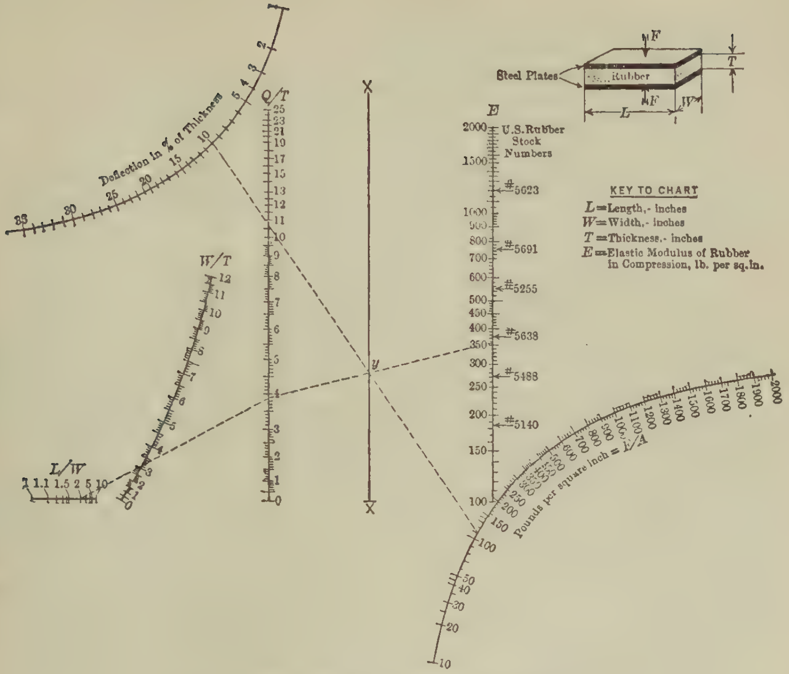

FIG. 2. Nomographic Chart for Rubber Sandwiches

EXAMPLE.—Determine the rubber stock number and durometer hardness of a rectangular rubber sandwich 1 in. thick, 6 in. long, 3 in. wide, to support a load of 2000 lb. with a deflection of approximately 10%. *Solution.*—$F/A = 2000/(6 \times 3) = 111.1$ lb. per sq. in. From the point corresponding to 111.1 on the pounds per square inch scale, Fig. 2, draw a line to point 10 on the deflection scale, intersecting XX at $y$. From the point $L/W = 6/3 = 2$ on the $L/W$ scale, draw a line to the point $W/T = 3/1 = 3$ on the $W/T$ scale, and extend it until it intersects the $Q/T$ scale at 3.9. From this point draw a line through $y$ and project it to intersect the $E$ scale at 360, which shows that the stock number to be used is 5638. Reference to Table 2 shows that stock No. 5638 corresponds to a durometer hardness of 50 ± 4.

**RUBBER IN SHEAR.**—Rubber sandwiches may be used in shear as a resilient mounting for heavy loads. Usually a double rubber sandwich is employed to balance out the rotational torque that would exist with a single rubber sandwich. The stress-strain relations in shear are given by the following formulas.

$$F/A = S(d/T), \text{ for values of } d/T = 0 \text{ to } 0.3 \qquad [4]$$

$$F/A = 0.9\,S(d/T), \text{ for values of } d/T = 0.3 \text{ to } 0.8 \qquad [5]$$

$$F/A = 0.833\,S(d/T), \text{ for values of } d/T = 0.8 \text{ to } 1.2 \qquad [6]$$

where $F$ = applied load producing shear, lb.; $A$ = area of sandwich, sq. in. = $L \times W$; $T$ = thickness of sandwich, in.; $d$ = deflection, in.; and $S$ = shear modulus (see Table 1). A more exact relation good up to 80% deflection in terms of $T$ is

$$F/A = S(\theta/57.3) = S\{\tan^{-1}(d/T)/57.3\} \qquad [7]$$

where $\theta$ = average angle of shear, deg. The shear in equations [4] to [7] may be used for sandwiches in shear with faces other than rectangular by replacing $(L \times W)$ with the area of one of the faces. If the two faces are not equal, $(L \times W)$ should be replaced with a value of one-half the sum of the area of the two faces. If lateral forces of tension or compression exist, equations [4] to [7] are approximately correct if $T$ is taken as the thickness when no lateral forces are applied. In general, metal-rubber bond should not be subjected to static stresses in shear greater than 25 lb. per sq. in.

**Angular Shear.**—Let $R$ = outer radius, $r$ = inner radius of a circular doughnut sandwich, in.; $T$ = thickness of sandwich, in.; $d_o$ = linear deflection, in., along the arc, of a point on the periphery at a distance $R$ from the center. $\theta$ = angular deflection of rotation, deg.; $S$ = shear modulus for the rubber, lb. per sq. in. $\tau$ = moment of couple or torque, lb.-in., producing angular deflection $\theta$. Then

$$\tau = \frac{\pi}{2}\,\frac{Sd_o}{T}\left[\frac{R^4 - r^4}{R}\right] = 0.0274\,(S\theta/T)/(R^4 - r^4) \quad . \; . \; . \; . \; . \; . \quad [8]$$

Maximum stress $f$ in the rubber will occur at radius $R$ where stress lb. per sq. in. is

$$f = S(d_o/T) = (SR/T)\,(\theta/57.3) \quad . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

For a given torque $\tau$ the dimensions of the sandwich must be such that $f$ is not too large. The most economical sandwich for a given torque and a maximum allowable value of $f$ is one in which the ratio $R/r = \sqrt{3}$.

## 2. GENERAL PROPERTIES OF RUBBER

**ABSORPTION OF LIQUID AND GASES.**—The amount and rate of absorption of water depends on the type of rubber, size of sample and its previous history. In general, amount of absorption increases from 1.2 to 2 times per 18° F. temperature rise, increases with amount of dissolved air and amount of aging. Repeated absorption with intermediate drying increases water-absorbing tendency. Water absorbed by thin rubber sheet samples ranges from 0.1% to 2%, by weight, in one day to 2% to 15% after one year. With thick samples the absorption is many times less.

The absorption of gas by rubber decreases, in general, with increase in temperature, and increases with increase in pressure. The absorption coefficient $C$ is the volume of gas, cu. cm. at 0° C. and atmospheric pressure, absorbed by one cu. cm. of rubber in equilibrium with the gas. Approximate values of $C$ for various gases, at room temperature, are:

| | | | | | |
|---|---|---|---|---|---|
| Hydrogen | $C$ = 0.01 | Air | $C$ = 0.12 | Nitrous oxide | $C$ = 1.6 |
| Nitrogen | = .04 | Methane | = .25 | Hydrogen sulphide | = 2.7 |
| Carbon monoxide | = .06 | Carbon dioxide | = 1.0 | Ammonia | = 10.0 |
| Oxygen | = .07 | Ethylene | = 1.4 | Sulphur dioxide | = 20.0 |

The value of $C$ for $CO_2$ will be halved, approximately, in going from room temperature to 120° F. above room temperature. A variation of pressure from 0.1 to 1 atmosphere will decrease the coefficient from 6 to 7 times.

**PERMEABILITY** is the amount of gas penetrating a unit area of rubber in unit time. It is proportional to the partial pressure of the gas on either side of the membrane and inversely proportional to its thickness. Permeability increases with increase in temperature. The specific permeability of hydrogen at normal temperature and pressure through rubber is approximately 0.000038 cu. ft. per day per sq. ft. per inch thickness per 1 atmosphere difference in pressure. The relative permeability of several gases is as follows, the permeability of hydrogen being taken as 1:

| | | | | | |
|---|---|---|---|---|---|
| Nitrogen | 0.16 | Oxygen | 0.44 | Ammonia | 8.0 |
| Air | 0.22 | Helium | 0.65 | Methyl chloride | 18.0 |
| Argon | 0.26 | Carbon dioxide | 2.9 | Water vapor | 50.0 |
| | | Nitrous oxide | 4.5 | | |

The variation with temperature of specific permeability $P$ of soft vulcanized rubber for hydrogen and carbon dioxide is as follows:

| Temperature 0° F. | 32 | 68 | 77 | 104 | 140 | 176 | 212 |
|---|---|---|---|---|---|---|---|
| Hydrogen, $P$ | = 0.12 | 0.30 | 0.34 | 0.57 | 1.02 | 1.68 | 2.60 |
| $CO_2$, $P$ | = 0.30 | 0.84 | 0.98 | 1.77 | 3.88 | 4.00 | 5.30 |

**ADHERENCE OF RUBBER TO METALS.**—Rubber may be bonded to metal with an ultimate bond strength of 400–500 lb. per sq. in. Such bonds are used in engine and motor mountings to absorb vibration, in linings of pipes or tanks conveying or holding acids or corrosive liquids, and where a positive fastening of rubber to adjoining metal parts is desired. Bonding may be effected by two processes. Large surfaces are cleaned and sand blasted and a vulcanizing cement applied. Uncured rubber sheets of the proper thickness then are laid on the metal and steam curing or vulcanizing effects the bond. Small molded parts are cleaned and electroplated with a suitable brass alloy. A special cement is applied, followed by the application of a piece of rubber of suitable size. The assembly is squeezed to shape in a mold in the presence of heat. The plating process can be applied to steel, nickel, zinc, aluminum, copper, brass, Monel metal and cast iron.

**EFFECT OF ORGANIC LIQUIDS AND VAPORS.**—In general, rubber is unaffected by weak acids, or weak caustic alkalies. Special hard rubber compounds are not attacked by concentrated hydrochloric or hydrofluoric acids, nor by sulphuric and nitric acids up to certain concentrations. Ozone cracks and soon destroys rubber, especially if it is under tension. Certain liquids and their vapors swell soft vulcanized rubber. The figures below show the absorption $Q$, in cu. cm. of liquid at room temperature, in one day by 1 cu. cm. of rubber. The values may be doubled or halved with different soft vulcanized rubber stocks. Special hard rubbers are only slightly, or not at all, affected by these liquids. In some cases the liquid will cause a slight softening of hard rubber.

| | $Q$ | | $Q$ | | $Q$ |
|---|---|---|---|---|---|
| Acetic acid | 0.1 | Carbon tetrachloride | 12.5 | Naphtha | 8.0 |
| Acetone | 0.2 | Castor oil | 0.01 | Nitrobenzine | 1.4 |
| Alcohol (ethyl) | 0.01 | Chloroform | 9.0 | Paraffin oil | 2.8 |
| " (methyl) | 0.02 | Cod liver oil | 1.3 | Petroleum | 6.0 |
| Aqueous ammonia | very small | Cottonseed oil (refined) | 1.0 | Rosin oil | 3.8 |
| Benzene | 5.8 | Creosote oil | 3.1 | Sperm oil | 1.8 |
| Benzyl chloride | 4.4 | Ethyl acetate | 0.4 | Tallow oil | 1.0 |
| Butter fat | 1.5 | Ethyl ether | 3.4 | Toluene | 7.4 |
| Butyric acid | very small | Gasoline | 4.0 | Turpentine | 5.5 |
| Carbon disulphide | 8.6 | Glycerine | 0.0 | Xylene | 6.3 |
| | | Linseed oil (raw) | 1.1 | | |

**DRIFT OF RUBBER.**—Rubber supporting a constant load over long periods undergoes a slowly increasing deformation called *drift*. Drift is distinguished from permanent set in that it is a measure of the deformation while the force is acting, not after the force is removed. Relative drift for $t$ days of constant load for a sandwich either in shear or compression is $(d_t - d_o)/d_o$, where $d_o$ = original deflection one minute after first application of load and $d_t$ = deflection $t$ days later. Relative drift depends, among other variables, on temperature, static load per unit area, operating media (oil and grease, which cause the rubber to deteriorate, appreciably change the drift), shape of rubber sample, etc. If the rubber sample is subjected to vibration, drift will be increased over that which would occur with purely static load.

**Shear Drift.**—The following are estimates of the approximate amount of drift to be expected under static loads causing 15% deflection in terms of sandwich thickness, after 24 hours constant loading at 100° F.:

| Stock No. | 5140 | 5488 | 5638 | 5255 | 5691 |
|---|---|---|---|---|---|
| Drift, percent | 2 | 5 | 8 | 12 | 15 |

After one year, these values of relative drift show a proportional increase, ranging from 20% for stock No. 5140 to 40% for stock No. 5691. That is, the deflection of stock No. 5140 used in shear to support a constant load corresponding to a 15% deflection will be, after one year, $15 \times \{1 + (20/100)\} = 18\%$.

**Compression Drift.**—Under static loads giving 15% initial compression, the relative drift after 24 hr. constant loading at 100° F. are approximately as follows:

| Stock No. | 5140 | 5488 | 5638 |
|---|---|---|---|
| Drift, percent | 2 | 4 | 8 |

The proportional increase after a period of one year ranges from 12% for No. 5140 stock to about 35% for No. 5638 stock. After two years, the relative drifts are 25% for No. 5140, 35% for No. 5488 and 50% for No. 5638. For rubber bushings, or rubber mountings, used under static loads and subject to continual vibration, stock No. 5140, from the standpoint of drift and vibration life, is better than the harder stocks.

## CARBON *

Essentially pure carbon occurs in widely varied forms from the diamond which is white, transparent and extremely hard, to graphite which is black, opaque and unctuous. Lampblack, charcoal and coke are practically pure carbon except for certain mineral or volatile impurities. Coal contains a very high percentage of carbon as well as a variety of tarry hydrocarbons from which the numerous coal tar products are derived. The peculiar physical and chemical characteristics of carbon adapt it to many uses for which no other material is available.

From a chemical standpoint carbon is very inactive, and resistant to most acids and other corrosive materials. It does not melt at incandescent temperatures, and the rate of oxidation is relatively slow. It can readily be formed in a variety of shapes during manufacturing operations, either in the initial plastic condition or by machining in solid form. In graphitic form, carbon is valuable as a lubricant where, because of temperature or other inhibiting factors, oil or grease lubrication is not feasible. The thermal conduc-

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* Compiled from data supplied by National Carbon Co., Cleveland.

### Table 1.—Specific Heat of Carbon

(National Carbon Co., Cleveland)

| Temp., deg. F. | 79–164 | 79–540 | 79–1300 | 97–1656 | 117–2179 | 133–2610 |
|---|---|---|---|---|---|---|
| Specific heat, carbon | 0.168 | 0.200 | 0.199 | 0.315 | 0.352 | 0.387 |
| " " graphite | 0.165 | 0.195 | 0.234 | 0.324 | 0.350 | 0.390 |
| " " diamond | 0.160 | 0.315 | 0.415 | .......... | .......... | .......... |

### Table 2.—Coefficient of Linear Expansion of Carbon per Degree F.*

(National Carbon Co., Cleveland)

| Form of Carbon | 68 to 212° F. | 68 to 392° F. | 68 to 572° F. | 68 to 662° F. | 68 to 752° F. | 68 to 1112° F. | 68 to 1472° F. | 68 to 1832° F. |
|---|---|---|---|---|---|---|---|---|
| Carbon, coke base | $0.0_{5}83$ | $0.0_{5}78$ | $0.0_{5}94$ | $0.0_{4}10$ | ........ | ........ | ........ | ........ |
| Carbon, lampblack base | $0.0_{4}21$ | $0.0_{4}23$ | $0.0_{4}25$ | $0.0_{4}25$ | ........ | ........ | ........ | ........ |
| Graphite, artificial | | | | | | | | |
| From 14-in. electrode | $0.0_{4}10$ | $0.0_{5}94$ | $0.0_{4}11$ | ........ | $0.0_{4}12$ | $0.0_{4}13$ | $0.0_{4}14$ | $0.0_{4}14$ |
| From 3-in. electrode | $0.0_{5}55$ | $0.0_{5}72$ | $0.0_{5}83$ | ........ | $0.0_{5}94$ | $0.0_{4}11$ | $0.0_{4}13$ | $0.0_{4}14$ |
| From 1-in. electrode | $0.0_{5}44$ | $0.0_{5}55$ | $0.0_{5}61$ | ........ | $0.0_{5}78$ | $0.0_{5}89$ | $0.0_{4}11$ | $0.0_{4}12$ |

* The subscript indicates the number of zeros following the decimal point; thus $0.0_{4}21 = 0.000021$.

tivity of carbon, especially non-graphitic, is low as compared with most metals, but high as compared with most other solid materials. Carbon is a good conductor of electricity, but of much higher electrical resistance than the metals.

The applications of carbon include: Lighting carbons; carbon brushes for motors and generators; welding carbon products; carbon and graphite electrodes and anodes for electrochemical and electro-metallurgical industries; carbon Raschig rings, tubes, brick, tile and special structural shapes used in handling and processing acids and other corrosive materials; dry cells; switch and circuit breaker contacts; resistance discs; steam turbine packing; thrust rings for automobile clutches; back plates, diaphragms and granular carbon for telephones; anodes and grids for electronic devices; and, in the form of pure graphite powder, as a lubricant and a constituent of lubricating greases.

**PHYSICAL PROPERTIES.**—The physical properties of carbon in its various forms are given below.

**Specific Gravity** varies with the type of carbon and the method of manufacture as follows: Gas calcined coke carbon, 1.98–2.10 *; gas calcined lampblack carbon, 1.80–1.85 *; graphitized coke carbon, 2.20–2.24; graphitized lampblack carbon, 1.98–2.08; graphite, pure, 2.25; diamond, 3.51.

**Melting and Volatilization.**—Carbon volatilizes without melting at ordinary pressures, the volatilization point being 6629° F.

**Specific Heat** of various forms of carbon at various temperatures is given in Table 1.

**Coefficient of Thermal Expansion** of carbon in various forms is given in Table 2.

**Thermal Conductivity** of carbon in B.t.u. per hr. per sq. ft. per deg. F. per in. thickness, is as follows: Electrographitic brush, 68–108° F., 84.187; natural graphite brush, 68–108° F., 113.217; coke electrode, 68–108° F., 22.934; 108–644° F., 4.645; 644–1112° F., 3.484; graphitized electrodes, 68–108° F., 165.471; 108–644° F., 1190.23; 644–1112° F., 870.90.

**Electrical Resistivity** in ohms per cubic centimeter at 20° C. is: Carbon, coke base, 0.0035–0.0046; carbon, lampblack base, 0.0046–0.0056; graphite, 0.0008–0.0013.

**Strength** of carbon varies with the form as follows, values being given in pounds per square inch.

| | Tensile Strength | Crushing Strength | Transverse Strength |
|---|---|---|---|
| Carbon, coke base | 600–1250 | 2500–9000 | 1500–6000 |
| Graphite, artificial | 700–1200 | 3000–4000 | 3000–5000 |

**CARBON BRUSHES** are divided into six classes on the basis of composition, as follows:

**Carbon.**—Practically all amorphous carbon.

**Carbon-Graphite.**—Amorphous carbon with a small percentage of graphite.

**Graphite-Carbon.**—Amorphous carbon with a large percentage of graphite.

**Graphite.**—Practically all graphite.

**Electro-graphitic.**—Amorphous carbon, wholly or in part converted to graphite in the electric furnace.

**Metal-graphite.**—Various proportions of metal and graphite.

Because of the wide range of operating conditions encountered and the many conflicting demands of service, a great variety of brush grades are manufactured with a wide range of physical characteristics. For specific information on the application of various grades of brushes, the brush manufacturers should be consulted.

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* Depends on source, character of bond, and degree of calcination.

**ELECTRODES.**—Carbon and graphite electrodes are important in the production of abrasives, aluminum, iron, steel, alloys, and carbide in the electric furnace, and in the electrolytic production of chlorine, caustic soda and other products. For the same physical size, graphite electrodes cost more than carbon electrodes but have lower electrical resistivity. For equivalent current-carrying capacity, graphite electrodes are much smaller than carbon electrodes.

Electrodes are made to specifications which vary somewhat with size, but, in general, are as follows: Permissible variation in diameter, ±2%; permissible curvature, $\frac{1}{2}$% of length, measured from the chord to arc of the electrode; minimum apparent density, carbon electrodes, 1.45, graphite electrodes, 1.51; electrodes must not show signs of coring or laminations, and surface must be clean and free from pits and blowholes.

Electrodes are made in many shapes and sizes. Round electrodes range from $\frac{1}{16}$ in. diam. and 12 in. long, to 18 in. diam. and 72 in. long for graphite, and up to 40 in. diam. and 110 in. long for carbon.

**Electrode Applications.**—Graphite electrodes are preferable to carbon for electrolytic operations. In electrothermic work, both carbon and graphite electrodes are used. Aluminum furnaces and carbide furnaces use carbon electrodes. Carbon electrodes are used for most ferroalloy furnaces with the exception of furnaces for ferrovanadium and some low-carbon steel alloys. Steel-making furnaces use both carbon and graphite electrodes, as do furnaces for making steel castings. Electric furnaces for gray iron castings use both carbon and graphite electrodes, the latter predominating. Furnaces for nonferrous materials use graphite electrodes almost exclusively. In the abrasive industry, both types are used, carbon electrodes predominating, although only graphite electrodes can be used for abrasives of very light color.

The principal factor governing choice of carbon or graphite electrodes is cost per ton of product. Where the electrodes are consumed slowly, as in the production of aluminum carbide and ferroalloys, carbon electrodes are preferred. Where the electrodes are consumed rapidly, as in iron and steel furnaces, graphite electrodes, which are but a little over half the size of the corresponding carbon electrodes for equivalent current-carrying capacity are preferable. The fact that in the steel and iron industry only about half as much graphite is used per ton of product offsets the higher cost of the carbon electrode. Graphite electrodes, because of their lighter weight for an equivalent current-carrying capacity, are more easily handled than carbon electrodes, control apparatus is more effective and the supporting structure can be made lighter. Further, the smaller size of the graphite electrode requires smaller ports in furnace walls and roofs with consequent less weakening of the structure. Many other factors enter into the correct choice of an electrode, and the manufacturer should be consulted in the selection of the electrode best suited to a given set of conditions. Table 3 gives current-carrying capacity of electrodes.

**Table 3.—Current-carrying Capacity of Carbon and Graphite Electrodes**
(National Carbon Co., Cleveland)

| Diam. or Side of Square, in. | Carbon Electrodes | | | | | Graphite Electrodes | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Round | | Square | | Current Density, amperes per sq. in. | Round | | Current Density, amperes per sq. in. | Square | | Current Density, amperes per sq. in. |
| | Area, sq. in. | Current-Carrying Capacity, amperes | Area, sq. in. | Current-Carrying Capacity, amperes | | Area, sq. in. | Current-Carrying Capacity, amperes | | Area, sq. in. | Current-Carrying Capacity, amperes | |
| 2 | ..... | ..... | ..... | ..... | ..... | 3.1416 | 600– 1,000 | 200–320 | 4 | 650– 1,000 | 160–250 |
| $2\frac{1}{2}$ | ..... | ..... | ..... | ..... | ..... | 4.9087 | 800– 1,500 | 160–310 | ..... | ..... | ..... |
| 3 | ..... | ..... | ..... | ..... | ..... | 7.0686 | 1,200– 2,100 | 170–300 | ..... | ..... | ..... |
| 4 | ..... | ..... | ..... | ..... | ..... | 12.566 | 1,800– 3,000 | 140–240 | 16 | 1,900– 2,900 | 120–180 |
| $5\frac{1}{8}$ | ..... | ..... | ..... | ..... | ..... | 20.629 | 2,300– 4,100 | 110–200 | ..... | ..... | ..... |
| 6 | 28 | 1,200– 1,700 | 36 | 1,500– 2,200 | 40–60 | 28.274 | 3,000– 4,300 | 110–190 | 36 | 3,800– 5,000 | 100–140 |
| 7 | ..... | ..... | ..... | ..... | ..... | 38.485 | 4,000– 7,000 | 110–180 | ..... | ..... | ..... |
| 8 | 50 | 2,000– 3,000 | 64 | 2,500– 3,800 | 40–60 | 50.265 | 5,500– 9,000 | 110–180 | 64 | 6,500– 8,500 | 100–130 |
| 9 | ..... | ..... | ..... | ..... | ..... | 63.617 | 6,500–10,500 | 100–170 | ..... | ..... | ..... |
| 10 | 79 | 3,000– 4,800 | 100 | 4,000– 6,000 | 40–60 | 78.540 | 8,500–12,500 | 100–160 | 100 | 9,000–13,000 | 90–130 |
| 12 | 113 | 4,500– 6,800 | 144 | 5,500– 8,500 | 40–60 | 113.10 | 11,500–17,000 | 100–150 | ..... | ..... | ..... |
| 14 | 154 | 5,400– 8,500 | 196 | 6,800–10,800 | 35–55 | 153.94 | 15,000–21,500 | 100–140 | ..... | ..... | ..... |
| 16 | 201 | 7,000–11,000 | 256 | 9,000–14,000 | 35–55 | 201.06 | 20,000–26,000 | 100–130 | ..... | ..... | ..... |
| 17 | 227 | 7,900–12,500 | ..... | ..... | 35–55 | ..... | ..... | ..... | ..... | ..... | ..... |
| 18 | 254 | 8,900–14,000 | ..... | ..... | 35–55 | 254.47 | 26,000–30,000 | 100–120 | ..... | ..... | ..... |
| 20 | 314 | 11,000–17,300 | 400 | 14,000–20,000 | 35–55 | ..... | ..... | ..... | ..... | ..... | ..... |
| 24 | 452 | 15,000–25,000 | ..... | ..... | 35–55 | ..... | ..... | ..... | ..... | ..... | ..... |
| 30 | 707 | 24,700–35,300 | ..... | ..... | 35–50 | ..... | ..... | ..... | ..... | ..... | ..... |
| 35 | 962 | 28,800–38,400 | ..... | ..... | 30–40 | ..... | ..... | ..... | ..... | ..... | ..... |
| 40 | 1257 | 37,000–50,000 | ..... | ..... | 30–40 | ..... | ..... | ..... | ..... | ..... | ..... |

**LIGHTING CARBONS.**—The original commercial application of carbon on an extensive scale was in the electric arc lamp. The principal types of carbons now made to meet specific applications of the carbon arc are motion picture projector carbons, motion picture studio carbons, photographic carbons, both white flame and panchromatic, searchlight carbons, enclosed arc carbons for blue printing, etc., therapeutic carbons, and industrial carbons. The three basic types of carbon arcs are the low-intensity neutral cored carbon arcs, the flame arc, and the high intensity arc. The intrinsic brilliancy of these types of carbon arcs in comparison with other light sources is given in Table 4.

**SPECTROSCOPIC CARBONS** are artificial graphite electrodes of exceptional purity used for spectroscopic analysis. The ash content is below 0.001%. This slight trace of impurity defies identification by chemical analysis, and in some cases is too small to produce visible lines in the spectrum other than those of the element carbon itself. In some cases, however, faint lines of other elements as Si, Mg, Ca, Fe, Al, Cu or Ag may be visible. Never are all of these impurities found in the same electrode and usually only one or two, and these in very minute quantity. Copper appears to be the most prevalent impurity. Spectroscopic electrodes are available in 6 in. and 12 in. lengths and in diameters $^3/_{16}$, $^1/_4$, $^5/_{16}$ and $^3/_8$ in.

**CHEMICAL CARBON PRODUCTS.**—Carbon is immune to reaction with most of the products encountered in chemical manufacturing plants at the concentrations and temperatures ordinarily employed. It is unaffected by all acids, alkalies, and salt solutions, except hot concentrated solutions of highly oxidizing character. Further, because of its high thermal conductivity and low coefficient of expansion, carbon is highly resistant to thermal shock. Carbon Raschig rings, tubes, bricks, tile and special structural shapes are the forms in which it is most extensively used in the chemical industries. Table 5 gives physical data on carbon Raschig rings.

**Carbon Bricks** are made in the same forms as standard refractory brick, but to slightly different dimensions. The approximate dimensions and weights per 100 brick of the

**Table 4.—Intrinsic Brilliancy of Light Sources**

(National Carbon Co., Cleveland)

| Light Source | Candle Power per sq. mm. |
|---|---|
| Magnetite arc stream | 6.2 |
| Flame arc stream | 8.0 |
| 900-watt, special tungsten filament, clear gas-filled bulb | 26.6 |
| Positive crater of cored D.C. low-intensity carbon arc | 155–175 |
| Crater of A.C. high-intensity carbon arc | 280 |
| Positive crater of non-rotating D.C. high-intensity carbon arc | 380 |
| Positive crater of rotating D.C. high-intensity carbon arc | 400–800 |
| Sun at zenith | 920–950 |

**Table 5.—Dimensions and Specifications for Raschig Rings**

(National Carbon Co., Cleveland)

| Specifications | Size Number | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
| Outside diameter, in. | 1/4 | 1/2 | 3/4 | 1 | 1 1/4 | 1 1/2 | 2 | 3 |
| Inside diameter, in. | 1/8 | 3/8 | 1/2 | 3/4 | 7/8 | 1 | 1 1/2 | 2 3/8 |
| Length, in. | 1/4 | 1/2 | 3/4 | 1 | 1 1/4 | 1 1/2 | 2 | 3 |
| Approximate number of rings per cu. ft. | 85,000 | 10,600 | 3,140 | 1,325 | 678 | 392 | 166 | 49 |
| Sq. ft. of absorption surface per 1,000 rings | 2.50 | 10.75 | 24 | 43 | 67 | 96 | 172 | 388 |
| Sq. ft. of absorption surface per cu. ft. of tower space, $F$ | 212 | 114 | 75 | 57 | 45 | 37.5 | 28.5 | 19 |
| Pounds of rings per cu. ft. of tower space | 46 | 27 | 34 | 27 | 31 | 34 | 27 | 23 |
| Minimum crushing strength across diameter, lb. | 17 | 13 | 42 | 44 | 89 | 138 | 157 | 236 |
| Load on each bottom ring per 10 ft. of tower height, lb. (dry rings) | 0.25 | 0.56 | 1.6 | 2.25 | 4.0 | 6.4 | 9.0 | 17.5 |
| Percent free gas space, $J$ | 55 | 74 | 67 | 74 | 69 | 67 | 74 | 78 |
| Sq. ft. absorption surface per cu. ft of free gas space ($F \div J$) | 387 | 155 | 113 | 77 | 65 | 56 | 39 | 25 |
| Relative scrubbing capacity ($F \times J$) | 116 | 84 | 50 | 42 | 31 | 25 | 21 | 15 |

various shapes are as follows: 9-in. straight, 9 × 4 1/2 × 2 1/2 in., 580 lb.; split, 9 × 4 1/2 × 1 1/16 in., 250 lb.; soap, 9 × 2 1/2 × 2 1/16 in., 270 lb.; No. 1 arch, 9 × 4 1/2 × (2 1/2–2 1/8) in., 540 lb.; No. 2 arch, 9 × 4 1/2 × (2 1/2 1 3/4) in., 495 lb.; No. 3 arch, 9 × 4 1/2 × (2 1/2–1) in., 405 lb.; No. 1 key, 9 × (4 1/2–4) × 2 1/2 in., 550 lb.; No. 2 key, 9 × (4 1/2–3) × 2 1/2 in., 520 lb.; No. 3 key, 9 × (4 1/2–3) × 2 1/2 in., 490 lb.; No. 1 wedge, 9 × 4 1/2 × (2 1/2–1 7/8) in., 510 lb.; No. 2 wedge, 9 × 4 1/2 × (2 1/2 1 1/2) in., 470 lb. For illustrations showing the shape of these bricks, see Table 25, p. 5–13. Table 6 gives data on carbon bricks.

**Table 6.—Physical Characteristics of Standard Carbon Brick**

(National Carbon Co., Cleveland)

| | |
|---|---|
| Tensile strength | 750–1250 lb per sq in. |
| Crushing strength | 2500–4000 lb. per sq. in. |
| Transverse strength | 1500–2500 lb. per sq. in. |
| Coefficient of thermal expansion per deg. F. (79–390° F.): | |
| Longitudinal | 0.000 000 889 |
| Transverse | 0.000 002 22 |
| Thermal conductivity, average, up to 1250° F. | 46,448 B.t.u. per hr. per sq. ft. per in. per deg. F. |
| Max. safe temperature under oxidizing conditions | 705° F. |
| Apparent density, average | 1.56 |
| Porosity, approximate | 25% |
| Weight per cu. ft., approximate | 100 lb. |

**Table 7.—Number of Carbon Brick Required to Form Various Circles**

(National Carbon Co., Cleveland)

| Inside Diameter | 9-in. Arch Brick | | | | | 9-in. Key Brick | | | | | 9-in. Wedge Brick | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | No. 3 Arch | No. 2 Arch | No. 1 Arch | Straight | Total | No. 3 Key | No. 2 Key | No. 1 Key | Straight | Total | No. 2 Wedge | No. 1 Wedge | Straight | Total |
| 0′ 6″ | 19 | ..... | ..... | ..... | 19 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 0′ 9″ | 18 | 5 | ..... | ..... | 23 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 1′ 0″ | 16 | 11 | ..... | ..... | 27 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 1′ 3″ | 12 | 19 | ..... | ..... | 31 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 1′ 6″ | 8 | 26 | ..... | ..... | 34 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 1′ 9″ | 4 | 34 | ..... | ..... | 38 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2′ 0″ | ..... | 42 | ..... | ..... | 42 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2′ 6″ | ..... | 33 | 16 | ..... | 49 | 34 | ..... | ..... | ..... | 34 | 60 | ..... | ..... | 60 |
| 3′ 0″ | ..... | 27 | 30 | ..... | 57 | 38 | ..... | ..... | ..... | 38 | 51 | 17 | ..... | 68 |
| 3′ 6″ | ..... | 20 | 44 | ..... | 64 | 34 | 8 | ..... | ..... | 42 | 42 | 33 | ..... | 75 |
| 4′ 0″ | ..... | 11 | 61 | ..... | 72 | 26 | 20 | ..... | ..... | 46 | 29 | 54 | ..... | 83 |
| 4′ 6″ | ..... | 5 | 75 | ..... | 80 | 18 | 32 | ..... | ..... | 50 | 16 | 75 | ..... | 91 |
| 5′ 0″ | ..... | ..... | 83 | 4 | 87 | 10 | 45 | ..... | ..... | 55 | 3 | 95 | ..... | 98 |
| 5′ 6″ | ..... | ..... | 83 | 12 | 95 | 2 | 57 | ..... | ..... | 59 | ..... | 100 | 6 | 106 |
| 6′ 0″ | ..... | ..... | 83 | 20 | 103 | ..... | 55 | 8 | ..... | 63 | ..... | 100 | 13 | 113 |
| 6′ 6″ | ..... | ..... | 83 | 27 | 110 | ..... | 51 | 16 | ..... | 67 | ..... | 100 | 21 | 121 |
| 7′ 0″ | ..... | ..... | 83 | 35 | 118 | ..... | 47 | 24 | ..... | 71 | ..... | 100 | 28 | 128 |
| 7′ 6″ | ..... | ..... | 83 | 42 | 125 | ..... | 43 | 33 | ..... | 76 | ..... | 100 | 36 | 136 |
| 8′ 0″ | ..... | ..... | 83 | 50 | 133 | ..... | 39 | 41 | ..... | 80 | ..... | 100 | 43 | 143 |
| 8′ 6″ | ..... | ..... | 83 | 57 | 140 | ..... | 35 | 49 | ..... | 84 | ..... | 100 | 51 | 151 |
| 9′ 0″ | ..... | ..... | 83 | 65 | 148 | ..... | 31 | 57 | ..... | 88 | ..... | 100 | 58 | 158 |
| 9′ 6″ | ..... | ..... | 83 | 72 | 155 | ..... | 26 | 66 | ..... | 92 | ..... | 100 | 66 | 166 |
| 10′ 0″ | ..... | ..... | 83 | 80 | 163 | ..... | 22 | 75 | ..... | 97 | ..... | 100 | 74 | 174 |
| 10′ 6″ | ..... | ..... | 83 | 87 | 170 | ..... | 18 | 83 | ..... | 101 | ..... | 100 | 81 | 181 |
| 11′ 0″ | ..... | ..... | 83 | 95 | 178 | ..... | 15 | 90 | ..... | 105 | ..... | 100 | 89 | 189 |
| 11′ 6″ | ..... | ..... | 83 | 102 | 185 | ..... | 11 | 98 | ..... | 109 | ..... | 100 | 96 | 196 |
| 12′ 0″ | ..... | ..... | 83 | 110 | 193 | ..... | 7 | 106 | ..... | 113 | ..... | 100 | 104 | 204 |
| 12′ 6″ | ..... | ..... | 83 | 117 | 200 | ..... | 3 | 114 | ..... | 117 | ..... | 100 | 111 | 211 |
| 13′ 0″ | ..... | ..... | 83 | 125 | 208 | ..... | ..... | 119 | 3 | 122 | ..... | 100 | 119 | 219 |
| 13′ 6″ | ..... | ..... | 83 | 132 | 215 | ..... | ..... | 119 | 7 | 126 | ..... | 100 | 126 | 226 |
| 14′ 0″ | ..... | ..... | 83 | 140 | 223 | ..... | ..... | 119 | 11 | 130 | ..... | 100 | 134 | 234 |
| 14′ 6″ | ..... | ..... | 83 | 147 | 230 | ..... | ..... | 119 | 15 | 134 | ..... | 100 | 141 | 241 |
| 15′ 0″ | ..... | ..... | 83 | 155 | 238 | ..... | ..... | 119 | 19 | 138 | ..... | 100 | 149 | 249 |
| 15′ 6″ | ..... | ..... | 83 | 162 | 245 | ..... | ..... | 119 | 24 | 143 | ..... | 100 | 156 | 256 |
| 16′ 0″ | ..... | ..... | 83 | 170 | 253 | ..... | ..... | 119 | 28 | 147 | ..... | 100 | 164 | 264 |
| 16′ 6″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 32 | 151 | ..... | 100 | 172 | 272 |
| 17′ 0″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 36 | 155 | ..... | 100 | 179 | 279 |
| 17′ 6″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 40 | 159 | ..... | 100 | 187 | 287 |
| 18′ 0″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 44 | 163 | ..... | 100 | 194 | 294 |
| 18′ 6″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 49 | 168 | ..... | ..... | ..... | ..... |
| 19′ 0″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 53 | 172 | ..... | ..... | ..... | ..... |
| 19′ 6″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 57 | 176 | ..... | ..... | ..... | ..... |
| 20′ 0″ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 119 | 61 | 180 | ..... | ..... | ..... | ..... |

**WELDING CARBON PRODUCTS** include carbon and graphite welding electrodes, carbon rods, plates and paste. The latter are valuable aids in all classes of gas or electric welding operations. They are used for molds, cores and dams to restrict the flow of molten metal and to protect adjacent parts against excessive heating. They are of assistance in making vertical welds, in welding a break without dismantling the parts to be welded, in making forms for building up broken parts, and in supporting thin sections or light parts during welding.

In all arc welding, arc stability is an important factor, both as regards speed and the quality of the weld. In carbon arc welding this stability is best preserved by maintaining a small point approximately 1/8 in. diameter at the tip of the electrode. Most economical operation will be obtained when conditions are so adjusted that the combined effect of surface oxidation and reduction in length of electrode tends to form a taper from this small tip to the full diameter of the electrode in a length of 6 to 8 times the diameter. Factors affecting the form of the electrode tip are type of electrode, current density in electrode, distance from holder to tip of electrode, period of continuous operation, and method of cooling electrode. The operator should seek to establish conditions which will maintain the long tapered point described above. The exact current which will maintain this point cannot be specified, as it depends on many factors, but for average hand welding operations it probably will be within the following range:

| Electrode diam., in. | 1/8 | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 | 7/8 | 1 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Welding current, amperes | | | | | | | | | | | |
| Max. | 0 | 25 | 50 | 80 | 110 | 140 | 170 | 230 | 290 | 350 | 400 |
| Min. | 35 | 60 | 90 | 125 | 165 | 210 | 260 | 370 | 490 | 615 | 750 |
| Max. current density, amperes per sq. in. | 2890 | 2200 | 1855 | 1650 | 1510 | 1420 | 1340 | 1220 | 1125 | 1035 | 965 |

# MISCELLANEOUS MATERIALS

**BRICKWORK.**—Brickwork is estimated by the thousand. The number of bricks for various thicknesses of wall is as follows:

| Thickness of wall, in. | 8 1/4 | 12 3/4 | 17 | 21 1/2 |
|---|---|---|---|---|
| Number of bricks thick | 1 | 1 1/2 | 2 | 2 1/2 |
| No. of bricks per superficial foot | 14 | 21 | 28 | 35 |

An ordinary brick is 8 1/4 × 4 × 2 in. = 66 cu. in. = 26.2 bricks per cu. ft. Its average weight is 4 1/2 lb.

**Clay and Sand-lime Brick.**—Table 1 gives the standard sizes of clay and sand-lime brick, and Table 2 the grades of brick according to strength. The compressive tests of Table 2 are made with a half-brick, flatwise, bedded in plaster-of-paris to provide an even bearing. The bending test is made with the brick flatwise, with the load applied at the middle of a 7-in. span. The average sand-lime brick is about equal in strength to a medium clay brick.

**Table 1.—Standard Sizes for Clay and Sand-lime Building Brick**

(A.S.T.M. Designations C62–30, C73–30)

| Types | Standard Sizes, in. | | | Permissible Variations, in. | | |
|---|---|---|---|---|---|---|
| | Depth | Width | Length | Depth | Width | Length |
| Common clay brick | 2 1/4 | 3 3/4 | 8 | ± 1/16 | ± 1/8 | ± 1/4 |
| Rough-face clay brick | 2 1/4 | 3 3/4 | 8 | ± 1/16 | ± 1/8 | ± 1/4 |
| Smooth-face clay brick | 2 1/4 | 3 7/8 | 8 | ± 1/16 | ± 1/8 | ± 1/4 |
| Sand-lime brick | 2 1/4 | 3 3/4 | 8 | ± 1/16 | ± 1/8 | ± 1/4 |

**Table 2.—Grades of Clay or Sand-lime Building Brick**

(A.S.T.M. Designations C62–30, C73–30)

| Name of Grade | Compressive Strength (bricks flatwise), lb. per sq. in., mean gross area | | Modulus of Rupture (bricks flatwise), lb. per sq. in., gross area | |
|---|---|---|---|---|
| | Mean of 5 tests | Individual minimum | Mean of 5 tests | Individual minimum |
| Grade A | 4500 or over | 3500 | 600 or over | 400 |
| Grade B | 2500–4500 | 2000 | 450 or over | 300 |
| Grade C | 1250–2500 | 1000 | 300 or over | 200 |

The above classifications are based on strength and do not necessarily measure weather resistance.

### Table 3.—Measures and Weights of Miscellaneous Materials

| Material | Cu. ft. per Ton (2240 lb.) | Lb. per cu. ft. | Lb. per Bushel | Bushels per Ton (2240 lb.) | Cu. in. per Bushel |
|---|---|---|---|---|---|
| FUELS: | | | | | |
| Anthracite, broken | 34–41 | 55–66 | ......... | ......... | ......... |
| Bituminous coal, broken§ | 41–45 | 50–55 | 70–76 | 29.47 | 2688 |
| Cannel coal | 44.6 | 50.3 | ......... | ......... | ......... |
| Charcoal | 123 | 18–18.5 | 20 | 100* | 2748‡ |
| Coke | 83–100 | 22–27 | 35–42 | 31–37 | 2688 |
| Semi-bituminous coal | 40.3 | 53 | 80 | ......... | ......... |
| GRAIN: | | | | | |
| Barley | ......... | 37–40 | 48† | ......... | 2150.4† |
| Corn | ......... | 45 | 56† | ......... | 2150.4† |
| Oats | ......... | 28–31 | 30† | ......... | 2150.4† |
| Rye | ......... | 44–50 | 60† | ......... | 2150.4† |
| Wheat | ......... | 44–50 | 60† | ......... | 2150.4† |
| EARTH, ETC.: | | | | | |
| Clay | 17 | 133 | ......... | ......... | ......... |
| Earth in bank | 18 | 124 | ......... | ......... | ......... |
| Earth, dry | 27 | 83 | ......... | ......... | ......... |
| Gravel, in bank | 18 | 124 | ......... | ......... | ......... |
| Gravel, dry | 27 | 83 | ......... | ......... | ......... |
| Quartz, broken | 20* | 100 | ......... | ......... | ......... |
| Sand | 25 | 90 | ......... | ......... | ......... |
| ORES: | | | | | |
| Copper | 14.6–5.3 | 132.7–374.1 | ......... | ......... | ......... |
| " chalcocite | 5.6 | 355.4 | ......... | ......... | ......... |
| " chalcopyrite | 7.6 | 262.0 | ......... | ......... | ......... |
| " cuprite | 5.3 | 374.1 | ......... | ......... | ......... |
| " malachite | 8.2 | 243.1 | ......... | ......... | ......... |
| Gold | 13* | 153.8 | ......... | ......... | ......... |
| Iron, pyrite | 6.3* | 318.0 | ......... | ......... | ......... |
| " hematite | 6.4* | 311.8 | ......... | ......... | ......... |
| " magnetite | 6.4* | 311.8 | ......... | ......... | ......... |
| " limonite | 8.4* | 236.9 | ......... | ......... | ......... |
| Lead | 5.3–4.4* | 374.1–455.1 | ......... | ......... | ......... |
| " galena | 4.4* | 455.1 | ......... | ......... | ......... |
| Manganese | 9.1–6.7* | 218.2–299.3 | ......... | ......... | ......... |
| Silver | 6.0–3.6* | 336.7–561.1 | ......... | ......... | ......... |
| Tin | 7.1–4.7* | 280.5–424.0 | ......... | ......... | ......... |
| Tungsten | 5.3–4.4* | 374.2–455.1 | ......... | ......... | ......... |
| Zinc | 16.0–5.6 | 124.7–355.4 | ......... | ......... | ......... |
| SALT: | | | | | |
| Syracuse, N. Y. | ......... | 45 | 56† | ......... | 2150.4† |
| Turks Island | ......... | 61–64 | 76–80 | ......... | 2150.4† |

* Per ton of 2000 lb.

† Per struck bushel.

‡ Standard bushel of American Charcoal Iron Worker's Assoc.

§ Bituminous coal runs about 1600 tons (2240 lb.) per acre, 1 ft. thick. Waste in mining is 15–25%.

**ASBESTOS.**—The following analyses of asbestos are given by J. T. Donald, *Eng. and Min. Jour.*, June 27, 1891:

| | | Canadian | |
|---|---|---|---|
| | Italian | Broughton | Templeton |
| Silica | 40.30% | 40.57% | 40.52% |
| Magnesia | 43.37 | 41.50 | 42.05 |
| Ferrous oxide | 0.87 | 2.81 | 1.97 |
| Alumina | 2.27 | 0.90 | 2.10 |
| Water | 13.72 | 13.55 | 13.46 |
| | 100.53 | 99.33 | 100.10 |

Chemical analysis throws light upon an important point in connection with asbestos, *i.e.*, the cause of the harshness of the fiber of some varieties. Asbestos is principally a hydrous silicate of magnesia, *i.e.*, silicate of magnesia combined with water. When harsh fiber is analyzed it is found to contain less water than the soft fiber. In fiber of very fine quality from Black Lake, analysis showed 14.38% of water, while a harsh-fibered sample gave only 11.70%. If soft fiber be heated to a temperature that will drive off a portion of the combined water, there results a substance so brittle that it may be crumbled between thumb and finger. There is evidently a connection between the consistency of the fiber and the amount of water in its composition.

**Asbestos Yarns** are numbered according to the number of yards per pound. Thus No. 1 yarn contains 100 yd. per lb.; No. 2 yarn, 200 yd. per lb.; No. 10 yarn, 1000 yd. per lb., etc. In multi-ply yarns, the yardage multiplied by the number of plies gives the yarn number. A three-ply yarn, of 600 lb. per yd., is equivalent to 600 × 3 = 1800 yd. of single-ply No. 18 yarn, and is known as three-ply, No. 18.

**CEMENT.**—Weighs from 90 to 118 lb. per cu. ft. The following are the legal commercial measures:

| | Net Wt. per Barrel, lb. | Net Wt. per Bag, lb. |
|---|---|---|
| Portland cement | 376 | 94 |
| Natural cement | 282 | 94 |

**ZIRCONIA.**—Zirconia ore (84.1 $ZrO_2$; 7.74 $SiO_2$; 3.10 $Fe_2O_3$; 1.21 $TiO_2$; 0.06 $Al_2O_3$; loss on ignition 2.72) vitrifies slightly at 1830° C. (3326° F.). Mixed with different percentages of clay and molded into cones it vitrifies at somewhat lower temperatures. A zirconia brick containing 5% clay became plastic on its face at 1800° C. (3272° F.). (H. Conrad Meyer, *Met. & Chem. Engg.*, vol. xii, No. 12, 1914; vol. xiii, No. 4, 1915; Circular of Foote Mineral Co., Philadelphia.)

## Section 6

# FABRICATED MATERIALS

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# FABRICATED MATERIALS

## PLATES, SHEETS, BARS, RODS AND SHAPES

**FORMULAS FOR WEIGHT OF PLATE STEEL.**—Based on a unit weight of steel of 0.284 lb. per cu. in., the weight of plate steel, either rectangular or circular, can be found by the formulas

$$\text{Rectangular plate, } W = B \times L \times C; \text{ Circular plate, } W = D \times K$$

where $W$ = weight, lb.; $B$ = width of plate, in.; $L$ = length of plate, in.; $D$ = diameter of plate, in.; $C$ and $K$ = constants whose values are given in Table 1.

**Table 1.—Values of $C$ and $K$**

$T$ = thickness of plate, in.

| $T =$ | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 3/4 | 7/8 | 1 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $C =$ | 0.0533 | 0.0710 | 0.0888 | 0.1065 | 0.1243 | 0.142 | 0.160 | 0.178 | 0.215 | 0.2485 | 0.284 |
| $K =$ | 0.0419 | 0.0558 | 0.0697 | 0.0839 | 0.0965 | 0.1122 | 0.1255 | 0.1394 | 0.1673 | 0.1952 | 0.2238 |

**WEIGHT OF COMMERCIAL PLATE AND SHEET.**—Table 3 gives the weight of plate iron, in thicknesses up to 1 in. Table 3 can be used for steel, by adding 2% to the values given for iron. Table 4 gives the weight of rectangular sections of steel, and Table 5 the cross-sectional areas of the sections given in Table 4.

**Non-ferrous Sheet and Plate.**—Tables 6 to 10 and 14 give the weight of plate and sheet of brass and zinc. Approximate weights of plate and sheet of other metals can be found by the application of the data in Table 2. For data on aluminum plates, see p. 6–08.

**BARS, RODS AND SHAPES.**—Table 2 gives constants by which the weight of bars, rods, plates, tubes and spheres of various materials can be calculated. The table is based on the following formulas:

**Notation.**—$b$ = breadth; $t$ = thickness; $s$ = side of square; $D$ = external diameter; $d$ = internal diameter, all in inches. $v$ = volume of 1 ft. length, cu. in., $w$ = weight, lb. per cu. in.

**Square Bars.**—Sectional area = $s^2$; volume of 1 ft. length = $v = 12s^2$; weight = $w \times v$.

**Round Bars.**—Sectional area = $0.7854\ D^2$; volume of 1 ft. length = $v = 9.4248\ D^2$; weight of 1 foot length = $w \times v = w \times 9.4248\ D^2$.

**Flats.**—Sectional area = $bt$; volume of 1 ft. length = $v = 12bt$; weight of 1 ft. length = $wv = 12\ wbt$.

**Tubes.**—Sectional area = $0.7854\ (D^2 - d^2) = 3.1416\ (Dt - t^2)$; volume of 1 ft.

(*Continued on p. 6–08*)

**Table 2.—Weight of Rods, Bars, Plates, Tubes and Spheres of Various Materials**

| Material | Specific Gravity | Weight per cu. ft., lb. | Weight of Plates 1 in. Thick, per sq. ft., lb. | Weight per ft. of Square Bars, lb. | Weight per ft. of Flat Bars, lb. | Weight per cu. in., lb. | Relative Weight. Wrought Iron = 1 | Weight per ft. of Round Rod, lb. | Weight of Spheres, lb. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | $s^2 \times$ | $bt \times$ | | | $D^2 \times$ | $D^3 \times$ |
| Cast iron | 7.218 | 450. | 37.5 | 3 1/8 | 3 1/8 | 0.2604 | 15/16 | 2.454 | 0.1363 |
| Wrought iron | 7.7 | 480. | 40. | 3 1/3 | 3 1/3 | .2779 | 1. | 2.618 | .1455 |
| Steel | 7.854 | 489.6 | 40.8 | 3.4 | 3.4 | .2833 | 1.02 | 2.670 | .1484 |
| Copper and Bronze (copper and tin) | 8.855 | 552. | 46. | 3.833 | 3.833 | .3195 | 1.15 | 3.011 | .1673 |
| Brass {Cu 65, Zn 35} | 8.393 | 523.2 | 43.6 | 3.633 | 3.633 | .3029 | 1.09 | 2.854 | .1586 |
| Monel metal, rolled | 8.95 | 558. | 46.5 | 3.87 | 3.87 | .323 | 1.16 | 3.043 | .1691 |
| Lead | 11.38 | 709.6 | 59.1 | 4.93 | 4.93 | .4106 | 1.48 | 3.870 | .2150 |
| Aluminum | 2.67 | 166.5 | 13.9 | 1.16 | 1.16 | .0963 | 0.347 | 0.908 | .0504 |
| Glass | 2.62 | 163.4 | 13.6 | 1.13 | 1.13 | .0945 | 0.34 | 0.891 | .0495 |
| Pine wood, dry | 0.481 | 30.0 | 2.5 | 0.21 | 0.21 | .0174 | 1/16 | 0.164 | .0091 |

## Table 3.—Weight of Plate Iron, per Lineal Foot, in Pounds

Based on 480 lb. per cubic foot. For steel add 2%

| Width, in. | Thickness, in. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/8 | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 11/16 | 3/4 | 13/16 | 7/8 | 15/16 | 1 |
| 12 | 5.00 | 7.50 | 10.00 | 12.50 | 15.00 | 17.50 | 20.00 | 22.50 | 25.00 | 27.50 | 30.00 | 32.50 | 35.00 | 37.50 | 40.00 |
| 13 | 5.42 | 8.13 | 10.83 | 13.54 | 16.25 | 18.96 | 21.67 | 24.38 | 27.08 | 29.79 | 32.50 | 35.21 | 37.92 | 40.63 | 43.33 |
| 14 | 5.83 | 8.75 | 11.67 | 14.58 | 17.50 | 20.42 | 23.33 | 26.25 | 29.17 | 32.08 | 35.00 | 37.92 | 40.83 | 43.75 | 46.67 |
| 15 | 6.25 | 9.38 | 12.50 | 15.63 | 18.75 | 21.88 | 25.00 | 28.13 | 31.25 | 34.38 | 37.50 | 40.63 | 43.75 | 46.88 | 50.00 |
| 16 | 6.67 | 10.00 | 13.33 | 16.67 | 20.00 | 23.33 | 26.67 | 30.00 | 33.33 | 36.67 | 40.00 | 43.33 | 46.67 | 50.00 | 53.33 |
| 17 | 7.08 | 10.63 | 14.17 | 17.71 | 21.25 | 24.79 | 28.33 | 31.88 | 35.42 | 38.96 | 42.50 | 46.05 | 49.59 | 53.13 | 56.67 |
| 18 | 7.50 | 11.25 | 15.00 | 18.75 | 22.50 | 26.25 | 30.00 | 33.75 | 37.50 | 41.25 | 45.00 | 48.75 | 52.50 | 56.25 | 60.00 |
| 19 | 7.92 | 11.87 | 15.83 | 19.79 | 23.75 | 27.71 | 31.67 | 35.67 | 39.58 | 43.54 | 47.50 | 51.45 | 55.41 | 59.37 | 63.33 |
| 20 | 8.33 | 12.50 | 16.67 | 20.83 | 25.00 | 29.17 | 33.33 | 37.50 | 41.67 | 45.83 | 50.00 | 54.17 | 58.33 | 62.50 | 66.67 |
| 21 | 8.75 | 13.13 | 17.50 | 21.88 | 26.25 | 30.63 | 35.00 | 39.38 | 43.75 | 48.13 | 52.50 | 56.88 | 61.25 | 65.63 | 70.00 |
| 22 | 9.17 | 13.75 | 18.33 | 22.92 | 27.50 | 32.08 | 36.67 | 41.25 | 45.83 | 50.42 | 55.00 | 59.58 | 64.17 | 68.75 | 73.33 |
| 23 | 9.58 | 14.38 | 19.17 | 23.96 | 28.75 | 33.54 | 38.33 | 43.13 | 47.92 | 52.71 | 57.50 | 62.30 | 67.09 | 71.88 | 76.67 |
| 24 | 10.00 | 15.00 | 20.00 | 25.00 | 30.00 | 35.00 | 40.00 | 45.00 | 50.00 | 55.00 | 60.00 | 65.00 | 70.00 | 75.00 | 80.00 |
| 25 | 10.42 | 15.62 | 20.83 | 26.04 | 31.25 | 36.46 | 41.67 | 46.88 | 52.08 | 57.29 | 62.50 | 67.70 | 72.91 | 78.13 | 83.33 |
| 26 | 10.83 | 16.25 | 21.67 | 27.08 | 32.50 | 37.92 | 43.33 | 48.75 | 54.17 | 59.58 | 65.00 | 70.42 | 75.83 | 81.25 | 86.67 |
| 27 | 11.25 | 16.88 | 22.50 | 28.13 | 33.75 | 39.38 | 45.00 | 50.63 | 56.25 | 61.88 | 67.50 | 73.13 | 78.75 | 84.38 | 90.00 |
| 28 | 11.67 | 17.50 | 23.33 | 29.17 | 35.00 | 40.83 | 46.67 | 52.50 | 58.33 | 64.17 | 70.00 | 75.84 | 81.67 | 87.50 | 93.33 |
| 29 | 12.08 | 18.13 | 24.17 | 30.21 | 36.25 | 42.29 | 48.33 | 54.38 | 60.42 | 66.46 | 72.50 | 78.55 | 84.59 | 90.63 | 96.67 |
| 30 | 12.50 | 18.75 | 25.00 | 31.25 | 37.50 | 43.75 | 50.00 | 56.25 | 62.50 | 68.75 | 75.00 | 81.25 | 87.50 | 93.75 | 100.0 |
| 32 | 13.33 | 20.00 | 26.67 | 33.33 | 40.00 | 46.67 | 53.33 | 60.00 | 66.67 | 73.33 | 80.00 | 86.67 | 93.33 | 100.0 | 106.7 |
| 34 | 14.17 | 21.25 | 28.33 | 35.42 | 42.50 | 49.58 | 56.67 | 63.75 | 70.83 | 77.91 | 85.00 | 92.08 | 99.17 | 106.3 | 113.3 |
| 36 | 15.00 | 22.50 | 30.00 | 37.50 | 45.00 | 52.50 | 60.00 | 67.50 | 75.00 | 82.50 | 90.00 | 97.50 | 105.0 | 112.5 | 120.0 |
| 38 | 15.83 | 23.75 | 31.67 | 39.59 | 47.50 | 55.42 | 63.33 | 71.25 | 79.17 | 87.09 | 95.00 | 102.9 | 110.8 | 118.8 | 126.7 |
| 40 | 16.67 | 25.00 | 33.33 | 41.67 | 50.00 | 58.33 | 66.67 | 75.00 | 83.33 | 91.67 | 100.0 | 108.3 | 116.7 | 125.0 | 133.3 |
| 42 | 17.50 | 26.25 | 35.00 | 43.75 | 52.50 | 61.25 | 70.00 | 78.75 | 87.50 | 96.25 | 105.0 | 113.7 | 122.5 | 131.3 | 140.0 |
| 44 | 18.33 | 27.50 | 36.67 | 45.84 | 55.00 | 64.17 | 73.33 | 82.50 | 91.67 | 100.8 | 110.0 | 119.2 | 128.3 | 137.5 | 146.7 |
| 46 | 19.17 | 28.75 | 38.33 | 47.92 | 57.50 | 67.08 | 76.67 | 86.25 | 95.83 | 105.4 | 115.0 | 124.6 | 134.2 | 143.8 | 153.3 |
| 48 | 20.00 | 30.00 | 40.00 | 50.00 | 60.00 | 70.00 | 80.00 | 90.00 | 100.0 | 110.0 | 120.0 | 130.0 | 140.0 | 150.0 | 160.0 |
| 50 | 20.83 | 31.25 | 41.67 | 52.08 | 62.50 | 72.91 | 83.33 | 93.75 | 104.2 | 114.6 | 125.0 | 135.4 | 145.8 | 156.3 | 166.7 |
| 52 | 21.67 | 32.50 | 43.33 | 54.17 | 65.00 | 75.83 | 86.67 | 97.50 | 108.3 | 119.2 | 130.0 | 140.8 | 151.7 | 162.5 | 173.3 |
| 54 | 22.50 | 33.75 | 45.00 | 56.25 | 67.50 | 78.75 | 90.00 | 101.75 | 112.5 | 123.8 | 135.0 | 146.3 | 157.5 | 168.8 | 180.0 |
| 56 | 23.33 | 35.00 | 46.67 | 58.33 | 70.00 | 81.66 | 93.33 | 105.00 | 116.7 | 128.3 | 140.0 | 151.7 | 163.3 | 175.0 | 186.7 |
| 58 | 24.17 | 36.25 | 48.33 | 60.42 | 72.50 | 84.58 | 96.67 | 108.75 | 120.8 | 132.9 | 145.0 | 157.1 | 169.2 | 181.3 | 193.3 |
| 60 | 25.00 | 37.50 | 50.00 | 62.50 | 75.00 | 87.50 | 100.00 | 112.50 | 125.0 | 137.5 | 150.0 | 162.5 | 175.0 | 187.5 | 200.0 |
| 66 | 27.50 | 41.25 | 55.00 | 68.75 | 82.50 | 96.25 | 110.00 | 123.75 | 137.5 | 151.3 | 165.0 | 178.8 | 192.5 | 206.3 | 220.0 |
| 72 | 30.00 | 45.00 | 60.00 | 75.00 | 90.00 | 105.00 | 120.00 | 135.00 | 150.0 | 165.0 | 180.0 | 195.0 | 210.0 | 225.0 | 240.0 |
| 78 | 32.50 | 48.75 | 65.00 | 81.25 | 97.50 | 113.75 | 130.00 | 146.25 | 162.5 | 178.8 | 195.0 | 211.3 | 227.5 | 243.8 | 260.0 |
| 84 | 35.00 | 52.50 | 70.00 | 87.50 | 105.00 | 122.50 | 140.00 | 157.50 | 175.0 | 192.5 | 210.0 | 227.5 | 245.0 | 262.5 | 280.0 |
| 90 | 37.50 | 56.25 | 75.00 | 93.75 | 112.50 | 131.25 | 150.00 | 168.75 | 187.5 | 206.3 | 225.0 | 243.8 | 262.5 | 281.3 | 300.0 |
| 96 | 40.00 | 60.00 | 80.00 | 100.00 | 120.00 | 140.00 | 160.00 | 180.00 | 200.0 | 220.0 | 240.0 | 260.0 | 280.0 | 300.0 | 320.0 |
| 102 | 42.50 | 63.75 | 85.00 | 106.25 | 127.50 | 148.75 | 170.00 | 191.25 | 212.5 | 233.8 | 255.0 | 276.3 | 297.5 | 318.8 | 340.0 |
| 108 | 45.00 | 67.50 | 90.00 | 112.50 | 135.00 | 157.50 | 180.00 | 202.50 | 225.0 | 247.5 | 270.0 | 292.5 | 315.0 | 337.5 | 360.0 |
| 114 | 47.50 | 71.25 | 95.00 | 118.75 | 142.50 | 166.25 | 190.00 | 213.75 | 237.5 | 261.3 | 285.0 | 308.8 | 332.5 | 356.3 | 380.0 |
| 120 | 50.00 | 75.00 | 100.00 | 125.00 | 150.00 | 175.00 | 200.00 | 225.00 | 250.0 | 275.0 | 300.0 | 325.0 | 350.0 | 375.0 | 400.0 |

## Table 4.—Weights of Rectangular Sections—Steel

Pounds per lineal foot—For iron multiply by 0.98

| Width, in. | Thickness, in. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/16 | 1/8 | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 11/16 | 3/4 | 13/16 | 7/8 | 15/16 | 1 |
| 1/4 | 0.053 | 0.106 | 0.159 | 0.213 | 0.266 | 0.319 | 0.372 | 0.425 | 0.478 | 0.531 | 0.584 | 0.638 | 0.691 | 0.744 | 0.797 | 0.850 |
| 1/2 | .106 | .213 | .319 | .425 | .531 | .638 | .744 | .850 | .956 | 1.063 | 1.169 | 1.275 | 1.381 | 1.488 | 1.594 | 1.700 |
| 3/4 | .159 | .319 | .478 | .638 | .797 | .956 | 1.116 | 1.275 | 1.434 | 1.594 | 1.753 | 1.913 | 2.072 | 2.231 | 2.391 | 2.550 |
| 1 | .213 | .425 | .638 | .850 | 1.063 | 1.275 | 1.488 | 1.700 | 1.913 | 2.125 | 2.338 | 2.550 | 2.763 | 2.975 | 3.188 | 3.400 |
| 1 1/4 | .266 | .531 | .797 | 1.063 | 1.328 | 1.594 | 1.859 | 2.125 | 2.391 | 2.656 | 2.922 | 3.188 | 3.453 | 3.719 | 3.984 | 4.250 |
| 1 1/2 | .319 | .638 | .956 | 1.275 | 1.594 | 1.913 | 2.231 | 2.550 | 2.869 | 3.188 | 3.506 | 3.825 | 4.144 | 4.463 | 4.781 | 5.100 |
| 1 3/4 | .372 | .744 | 1.116 | 1.488 | 1.859 | 2.231 | 2.603 | 2.975 | 3.347 | 3.719 | 4.091 | 4.463 | 4.834 | 5.206 | 5.578 | 5.950 |
| 2 | .425 | .850 | 1.275 | 1.700 | 2.125 | 2.550 | 2.975 | 3.400 | 3.825 | 4.250 | 4.675 | 5.100 | 5.525 | 5.950 | 6.375 | 6.800 |
| 2 1/4 | .478 | .956 | 1.434 | 1.913 | 2.391 | 2.869 | 3.347 | 3.825 | 4.303 | 4.781 | 5.259 | 5.738 | 6.216 | 6.694 | 7.172 | 7.650 |
| 2 1/2 | .531 | 1.063 | 1.594 | 2.125 | 2.656 | 3.188 | 3.719 | 4.250 | 4.781 | 5.313 | 5.844 | 6.375 | 6.906 | 7.438 | 7.969 | 8.500 |
| 2 3/4 | .584 | 1.169 | 1.753 | 2.338 | 2.922 | 3.506 | 4.091 | 4.675 | 5.259 | 5.844 | 6.428 | 7.013 | 7.597 | 8.181 | 8.766 | 9.350 |
| 3 | .638 | 1.275 | 1.913 | 2.550 | 3.188 | 3.825 | 4.463 | 5.100 | 5.738 | 6.375 | 7.013 | 7.650 | 8.288 | 8.925 | 9.563 | 10.20 |
| 3 1/4 | .691 | 1.381 | 2.072 | 2.763 | 3.453 | 4.144 | 4.834 | 5.525 | 6.216 | 6.906 | 7.597 | 8.288 | 8.978 | 9.669 | 10.36 | 11.05 |
| 3 1/2 | .744 | 1.488 | 2.231 | 2.975 | 3.719 | 4.463 | 5.206 | 5.950 | 6.694 | 7.438 | 8.181 | 8.925 | 9.669 | 10.41 | 11.16 | 11.90 |
| 3 3/4 | .797 | 1.594 | 2.391 | 3.188 | 3.984 | 4.781 | 5.578 | 6.375 | 7.172 | 7.969 | 8.766 | 9.563 | 10.36 | 11.16 | 11.95 | 12.75 |
| 4 | .850 | 1.700 | 2.550 | 3.400 | 4.250 | 5.100 | 5.950 | 6.800 | 7.650 | 8.500 | 9.350 | 10.20 | 11.05 | 11.90 | 12.75 | 13.60 |
| 4 1/4 | .903 | 1.806 | 2.709 | 3.613 | 4.516 | 5.419 | 6.322 | 7.225 | 8.128 | 9.031 | 9.934 | 10.84 | 11.74 | 12.64 | 13.55 | 14.45 |
| 4 1/2 | .956 | 1.913 | 2.869 | 3.825 | 4.781 | 5.738 | 6.694 | 7.650 | 8.606 | 9.563 | 10.52 | 11.48 | 12.43 | 13.39 | 14.34 | 15.30 |
| 4 3/4 | 1.000 | 2.019 | 3.028 | 4.038 | 5.047 | 6.056 | 7.066 | 8.075 | 9.084 | 10.09 | 11.10 | 12.11 | 13.12 | 14.13 | 15.14 | 16.15 |
| 5 | 1.063 | 2.125 | 3.188 | 4.250 | 5.313 | 6.375 | 7.438 | 8.500 | 9.563 | 10.63 | 11.69 | 12.75 | 13.81 | 14.88 | 15.94 | 17.00 |
| 5 1/4 | 1.116 | 2.231 | 3.347 | 4.463 | 5.578 | 6.694 | 7.809 | 8.925 | 10.04 | 11.16 | 12.27 | 13.39 | 14.50 | 15.62 | 16.73 | 17.85 |
| 5 1/2 | 1.169 | 2.338 | 3.506 | 4.675 | 5.844 | 7.013 | 8.181 | 9.350 | 10.52 | 11.69 | 12.86 | 14.03 | 15.19 | 16.36 | 17.53 | 18.70 |
| 5 3/4 | 1.222 | 2.444 | 3.666 | 4.888 | 6.109 | 7.331 | 8.553 | 9.775 | 11.00 | 12.22 | 13.44 | 14.66 | 15.88 | 17.11 | 18.33 | 19.55 |
| 6 | 1.275 | 2.550 | 3.825 | 5.100 | 6.375 | 7.650 | 8.925 | 10.20 | 11.48 | 12.75 | 14.03 | 15.30 | 16.58 | 17.85 | 19.13 | 20.40 |
| 6 1/4 | 1.328 | 2.656 | 3.984 | 5.313 | 6.641 | 7.969 | 9.297 | 10.63 | 11.95 | 13.28 | 14.61 | 15.94 | 17.27 | 18.59 | 19.92 | 21.25 |
| 6 1/2 | 1.381 | 2.763 | 4.144 | 5.525 | 6.906 | 8.288 | 9.669 | 11.05 | 12.43 | 13.81 | 15.19 | 16.58 | 17.96 | 19.34 | 20.72 | 22.10 |
| 6 3/4 | 1.434 | 2.869 | 4.303 | 5.738 | 7.172 | 8.606 | 10.04 | 11.48 | 12.91 | 14.34 | 15.78 | 17.21 | 18.65 | 20.08 | 21.52 | 22.95 |
| 7 | 1.488 | 2.975 | 4.463 | 5.950 | 7.438 | 8.925 | 10.41 | 11.90 | 13.39 | 14.88 | 16.36 | 17.85 | 19.34 | 20.83 | 22.31 | 23.80 |
| 7 1/4 | 1.541 | 3.081 | 4.622 | 6.163 | 7.703 | 9.244 | 10.78 | 12.33 | 13.87 | 15.41 | 16.95 | 18.49 | 20.03 | 21.57 | 23.11 | 24.65 |
| 7 1/2 | 1.594 | 3.188 | 4.781 | 6.375 | 7.969 | 9.563 | 11.16 | 12.75 | 14.34 | 15.94 | 17.53 | 19.13 | 20.72 | 22.31 | 23.91 | 25.50 |
| 7 3/4 | 1.647 | 3.294 | 4.941 | 6.588 | 8.234 | 9.881 | 11.53 | 13.18 | 14.82 | 16.47 | 18.12 | 19.76 | 21.41 | 23.06 | 24.70 | 26.35 |
| 8 | 1.700 | 3.400 | 5.100 | 6.800 | 8.500 | 10.20 | 11.90 | 13.60 | 15.30 | 17.00 | 18.70 | 20.40 | 22.10 | 23.80 | 25.50 | 27.20 |
| 8 1/4 | 1.753 | 3.506 | 5.259 | 7.013 | 8.766 | 10.52 | 12.27 | 14.03 | 15.78 | 17.53 | 19.28 | 21.04 | 22.79 | 24.54 | 26.30 | 28.05 |
| 8 1/2 | 1.806 | 3.613 | 5.419 | 7.225 | 9.031 | 10.84 | 12.64 | 14.45 | 16.26 | 18.06 | 19.87 | 21.68 | 23.48 | 25.29 | 27.09 | 28.90 |
| 8 3/4 | 1.859 | 3.719 | 5.578 | 7.438 | 9.297 | 11.16 | 13.02 | 14.88 | 16.73 | 18.59 | 20.45 | 22.31 | 24.17 | 26.03 | 27.89 | 29.75 |
| 9 | 1.913 | 3.825 | 5.738 | 7.650 | 9.563 | 11.48 | 13.39 | 15.30 | 17.21 | 19.13 | 21.04 | 22.95 | 24.86 | 26.78 | 28.69 | 30.60 |
| 9 1/4 | 1.966 | 3.931 | 5.897 | 7.863 | 9.828 | 11.79 | 13.76 | 15.73 | 17.69 | 19.66 | 21.62 | 23.59 | 25.55 | 27.52 | 29.48 | 31.45 |
| 9 1/2 | 2.019 | 4.038 | 6.056 | 8.075 | 10.09 | 12.11 | 14.13 | 16.15 | 18.17 | 20.19 | 22.21 | 24.23 | 26.24 | 28.26 | 30.28 | 32.30 |
| 9 3/4 | 2.072 | 4.144 | 6.216 | 8.288 | 10.36 | 12.43 | 14.50 | 16.58 | 18.65 | 20.72 | 22.79 | 24.86 | 26.93 | 29.01 | 31.08 | 33.15 |
| 10 | 2.125 | 4.250 | 6.375 | 8.500 | 10.63 | 12.75 | 14.88 | 17.00 | 19.13 | 21.25 | 23.38 | 25.50 | 27.63 | 29.75 | 31.88 | 34.00 |
| 10 1/2 | 2.231 | 4.463 | 6.694 | 8.925 | 11.16 | 13.39 | 15.62 | 17.85 | 20.08 | 22.31 | 24.54 | 26.78 | 29.01 | 31.24 | 33.47 | 35.70 |
| 11 | 2.338 | 4.675 | 7.013 | 9.350 | 11.69 | 14.03 | 16.36 | 18.70 | 21.04 | 23.38 | 25.71 | 28.05 | 30.39 | 32.73 | 35.06 | 37.40 |
| 11 1/2 | 2.444 | 4.888 | 7.331 | 9.775 | 12.22 | 14.66 | 17.11 | 19.55 | 21.99 | 24.44 | 26.88 | 29.33 | 31.77 | 34.21 | 36.66 | 39.10 |
| 12 | 2.550 | 5.100 | 7.650 | 10.20 | 12.75 | 15.30 | 17.85 | 20.40 | 22.95 | 25.50 | 28.05 | 30.60 | 33.15 | 35.70 | 38.25 | 40.80 |
| 12 1/2 | 2.660 | 5.310 | 7.970 | 10.63 | 13.28 | 15.94 | 18.59 | 21.25 | 23.91 | 26.56 | 29.20 | 31.90 | 34.50 | 37.20 | 39.80 | 42.50 |
| 13 | 2.760 | 5.530 | 8.290 | 11.05 | 13.81 | 16.58 | 19.34 | 22.10 | 24.86 | 27.63 | 30.40 | 33.20 | 35.90 | 38.70 | 41.40 | 44.20 |
| 13 1/2 | 2.870 | 5.740 | 8.610 | 11.48 | 14.34 | 17.21 | 20.08 | 22.95 | 25.82 | 28.69 | 31.60 | 34.40 | 37.30 | 40.20 | 43.00 | 45.90 |
| 14 | 2.980 | 5.950 | 8.930 | 11.90 | 14.88 | 17.85 | 20.83 | 23.80 | 26.78 | 29.75 | 32.70 | 35.70 | 38.70 | 41.70 | 44.60 | 47.60 |
| 14 1/2 | 3.080 | 6.160 | 9.240 | 12.33 | 15.41 | 18.49 | 21.57 | 24.65 | 27.73 | 30.81 | 33.90 | 37.00 | 40.10 | 43.10 | 46.20 | 49.30 |
| 15 | 3.190 | 6.380 | 9.560 | 12.75 | 15.94 | 19.13 | 22.31 | 25.50 | 28.69 | 31.88 | 35.10 | 38.30 | 41.40 | 44.60 | 47.80 | 51.00 |
| 15 1/2 | 3.290 | 6.590 | 9.880 | 13.18 | 16.47 | 19.76 | 23.06 | 26.35 | 29.64 | 32.94 | 36.20 | 39.50 | 42.80 | 46.10 | 49.40 | 52.70 |
| 16 | 3.400 | 6.800 | 10.20 | 13.60 | 17.00 | 20.40 | 23.80 | 27.20 | 30.60 | 34.00 | 37.40 | 40.80 | 44.20 | 47.60 | 51.00 | 54.40 |
| 16 1/2 | 3.510 | 7.010 | 10.52 | 14.03 | 17.53 | 21.04 | 24.54 | 28.05 | 31.56 | 35.06 | 38.60 | 42.10 | 45.60 | 49.10 | 52.60 | 56.10 |
| 17 | 3.610 | 7.230 | 10.84 | 14.45 | 18.06 | 21.68 | 25.29 | 28.90 | 32.51 | 36.13 | 39.70 | 43.40 | 47.00 | 50.60 | 54.20 | 57.80 |
| 17 1/2 | 3.720 | 7.440 | 11.16 | 14.88 | 18.59 | 22.31 | 26.03 | 29.75 | 33.47 | 37.19 | 40.90 | 44.60 | 48.30 | 52.10 | 55.80 | 59.50 |
| 18 | 3.830 | 7.650 | 11.48 | 15.30 | 19.13 | 22.95 | 26.78 | 30.60 | 34.43 | 38.25 | 42.10 | 45.90 | 49.70 | 53.60 | 57.40 | 61.20 |
| 18 1/2 | 3.930 | 7.860 | 11.79 | 15.73 | 19.66 | 23.59 | 27.52 | 31.45 | 35.38 | 39.31 | 43.20 | 47.20 | 51.10 | 55.00 | 59.00 | 62.90 |
| 19 | 4.040 | 8.080 | 12.11 | 16.15 | 20.19 | 24.23 | 28.26 | 32.30 | 36.34 | 40.38 | 44.40 | 48.50 | 52.50 | 56.50 | 60.60 | 64.60 |
| 19 1/2 | 4.140 | 8.290 | 12.43 | 16.58 | 20.72 | 24.86 | 29.01 | 33.15 | 37.29 | 41.44 | 45.60 | 49.70 | 53.90 | 58.00 | 62.20 | 66.30 |
| 20 | 4.250 | 8.500 | 12.75 | 17.00 | 21.25 | 25.50 | 29.75 | 34.00 | 38.25 | 42.50 | 46.80 | 51.00 | 55.30 | 59.50 | 63.80 | 68.00 |

### Table 5.—Cross-sectional Areas of Rectangular Sections, Square Inches

| Width, in. | Thickness, in. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/16 | 1/8 | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 11/16 | 3/4 | 13/16 | 7/8 | 15/16 | 1 |
| 1/4 | 0.016 | 0.031 | 0.047 | 0.063 | 0.078 | 0.094 | 0.109 | 0.125 | 0.141 | 0.156 | 0.172 | 0.188 | 0.203 | 0.219 | 0.234 | 0.250 |
| 1/2 | .031 | .063 | .094 | .125 | .156 | .188 | .219 | .250 | .281 | .313 | .344 | .375 | .406 | .438 | .469 | .500 |
| 3/4 | .047 | .094 | .141 | .188 | .234 | .281 | .328 | .375 | .422 | .469 | .516 | .563 | .609 | .656 | .703 | .750 |
| 1 | .063 | .125 | .188 | .250 | .313 | .375 | .438 | .500 | .563 | .625 | .688 | .750 | .813 | .875 | .938 | 1.000 |
| 1 1/4 | .078 | .156 | .234 | .313 | .391 | .469 | .547 | .625 | .703 | .781 | .859 | .938 | 1.016 | 1.094 | 1.172 | 1.250 |
| 1 1/2 | .094 | .188 | .281 | .375 | .469 | .563 | .656 | .750 | .844 | .938 | 1.031 | 1.125 | 1.219 | 1.313 | 1.406 | 1.500 |
| 1 3/4 | .109 | .219 | .328 | .438 | .547 | .656 | .766 | .875 | .984 | 1.094 | 1.203 | 1.313 | 1.422 | 1.531 | 1.641 | 1.750 |
| 2 | .125 | .250 | .375 | .500 | .625 | .750 | .875 | 1.000 | 1.125 | 1.250 | 1.375 | 1.500 | 1.625 | 1.750 | 1.875 | 2.000 |
| 2 1/4 | .141 | .281 | .422 | .563 | .703 | .844 | .984 | 1.125 | 1.266 | 1.406 | 1.547 | 1.688 | 1.828 | 1.969 | 2.109 | 2.250 |
| 2 1/2 | .156 | .313 | .469 | .625 | .781 | .938 | 1.094 | 1.250 | 1.406 | 1.563 | 1.719 | 1.875 | 2.031 | 2.188 | 2.344 | 2.500 |
| 2 3/4 | .172 | .344 | .516 | .688 | .859 | 1.031 | 1.203 | 1.375 | 1.547 | 1.719 | 1.891 | 2.063 | 2.234 | 2.406 | 2.578 | 2.750 |
| 3 | .188 | .375 | .563 | .750 | .938 | 1.125 | 1.313 | 1.500 | 1.688 | 1.875 | 2.063 | 2.250 | 2.438 | 2.625 | 2.813 | 3.000 |
| 3 1/4 | .203 | .406 | .609 | .813 | 1.016 | 1.219 | 1.422 | 1.625 | 1.828 | 2.031 | 2.234 | 2.438 | 2.641 | 2.844 | 3.047 | 3.250 |
| 3 1/2 | .219 | .438 | .656 | .875 | 1.094 | 1.313 | 1.531 | 1.750 | 1.969 | 2.188 | 2.406 | 2.625 | 2.844 | 3.063 | 3.281 | 3.500 |
| 3 3/4 | .234 | .469 | .703 | .938 | 1.172 | 1.406 | 1.641 | 1.875 | 2.109 | 2.344 | 2.578 | 2.813 | 3.047 | 3.281 | 3.516 | 3.750 |
| 4 | .250 | .500 | .750 | 1.000 | 1.250 | 1.500 | 1.750 | 2.000 | 2.250 | 2.500 | 2.750 | 3.000 | 3.250 | 3.500 | 3.750 | 4.000 |
| 4 1/4 | .266 | .531 | .797 | 1.063 | 1.328 | 1.594 | 1.859 | 2.125 | 2.391 | 2.656 | 2.922 | 3.188 | 3.453 | 3.719 | 3.984 | 4.250 |
| 4 1/2 | .281 | .563 | .844 | 1.125 | 1.406 | 1.688 | 1.969 | 2.250 | 2.531 | 2.813 | 3.094 | 3.375 | 3.656 | 3.938 | 4.219 | 4.500 |
| 4 3/4 | .297 | .594 | .891 | 1.188 | 1.484 | 1.781 | 2.078 | 2.375 | 2.672 | 2.969 | 3.266 | 3.563 | 3.859 | 4.156 | 4.453 | 4.750 |
| 5 | .313 | .625 | .938 | 1.250 | 1.563 | 1.875 | 2.188 | 2.500 | 2.813 | 3.125 | 3.438 | 3.750 | 4.063 | 4.375 | 4.688 | 5.000 |
| 5 1/4 | .328 | .656 | .984 | 1.313 | 1.641 | 1.969 | 2.297 | 2.625 | 2.953 | 3.281 | 3.609 | 3.938 | 4.266 | 4.594 | 4.922 | 5.250 |
| 5 1/2 | .344 | .688 | 1.031 | 1.375 | 1.719 | 2.063 | 2.406 | 2.750 | 3.094 | 3.438 | 3.781 | 4.125 | 4.469 | 4.813 | 5.156 | 5.500 |
| 5 3/4 | .359 | .719 | 1.078 | 1.438 | 1.797 | 2.156 | 2.516 | 2.875 | 3.234 | 3.594 | 3.953 | 4.313 | 4.672 | 5.031 | 5.391 | 5.750 |
| 6 | .375 | .750 | 1.125 | 1.500 | 1.875 | 2.250 | 2.625 | 3.000 | 3.375 | 3.750 | 4.125 | 4.500 | 4.875 | 5.250 | 5.625 | 6.000 |
| 6 1/4 | .391 | .781 | 1.172 | 1.563 | 1.953 | 2.344 | 2.734 | 3.125 | 3.516 | 3.906 | 4.297 | 4.688 | 5.078 | 5.469 | 5.859 | 6.250 |
| 6 1/2 | .406 | .813 | 1.219 | 1.625 | 2.031 | 2.438 | 2.844 | 3.250 | 3.656 | 4.063 | 4.469 | 4.875 | 5.281 | 5.688 | 6.094 | 6.500 |
| 6 3/4 | .422 | .844 | 1.266 | 1.688 | 2.109 | 2.531 | 2.953 | 3.375 | 3.797 | 4.219 | 4.641 | 5.063 | 5.484 | 5.906 | 6.328 | 6.750 |
| 7 | .438 | .875 | 1.313 | 1.750 | 2.188 | 2.625 | 3.063 | 3.500 | 3.938 | 4.375 | 4.813 | 5.250 | 5.688 | 6.125 | 6.563 | 7.000 |
| 7 1/4 | .453 | .906 | 1 359 | 1.813 | 2.266 | 2.719 | 3.172 | 3.625 | 4.078 | 4.531 | 4.984 | 5.438 | 5.891 | 6.344 | 6.797 | 7.250 |
| 7 1/2 | .469 | .938 | 1.406 | 1.875 | 2.344 | 2.813 | 3.281 | 3.750 | 4.219 | 4.688 | 5.156 | 5.625 | 6.094 | 6.563 | 7.031 | 7.500 |
| 7 3/4 | .484 | .969 | 1.453 | 1.938 | 2.422 | 2.906 | 3.391 | 3.875 | 4.359 | 4.844 | 5.328 | 5.813 | 6.297 | 6.781 | 7.266 | 7.750 |
| 8 | .500 | 1.000 | 1.500 | 2.000 | 2.500 | 3.000 | 3.500 | 4.000 | 4.500 | 5.000 | 5.500 | 6.000 | 6.500 | 7.000 | 7.500 | 8.000 |
| 8 1/4 | .516 | 1.031 | 1.547 | 2.063 | 2.578 | 3.094 | 3.609 | 4.125 | 4.641 | 5.156 | 5.672 | 6.188 | 6.703 | 7.219 | 7.734 | 8.250 |
| 8 1/2 | .531 | 1.063 | 1.594 | 2.125 | 2.656 | 3.188 | 3.719 | 4.250 | 4.781 | 5.313 | 5.844 | 6.375 | 6.906 | 7.438 | 7.969 | 8.500 |
| 8 3/4 | .547 | 1.094 | 1.641 | 2.188 | 2.734 | 3.281 | 3.828 | 4.375 | 4.922 | 5.469 | 6.016 | 6.563 | 7.109 | 7.656 | 8.203 | 8.750 |
| 9 | .563 | 1.125 | 1.688 | 2.250 | 2.813 | 3.375 | 3.938 | 4.500 | 5.063 | 5.625 | 6.188 | 6.750 | 7.313 | 7.875 | 8.438 | 9.000 |
| 9 1/4 | .578 | 1.156 | 1.734 | 2.313 | 2.891 | 3.469 | 4.047 | 4.625 | 5.203 | 5.781 | 6.359 | 6.938 | 7.516 | 8.094 | 8.672 | 9.250 |
| 9 1/2 | .594 | 1.188 | 1.781 | 2.375 | 2.969 | 3.563 | 4.156 | 4.750 | 5.344 | 5.938 | 6.531 | 7.125 | 7.719 | 8.313 | 8.906 | 9.500 |
| 9 3/4 | .609 | 1.219 | 1.828 | 2.438 | 3.047 | 3.656 | 4.266 | 4.875 | 5.484 | 6.094 | 6.703 | 7.313 | 7.922 | 8.531 | 9.141 | 9.750 |
| 10 | .625 | 1.250 | 1.875 | 2.500 | 3.125 | 3.750 | 4.375 | 5.000 | 5.625 | 6.250 | 6.875 | 7.500 | 8.125 | 8.750 | 9.375 | 10.00 |
| 10 1/2 | .656 | 1.313 | 1.969 | 2.625 | 3.281 | 3.938 | 4.594 | 5.250 | 5.906 | 6.563 | 7.219 | 7.875 | 8.531 | 9.188 | 9.844 | 10.50 |
| 11 | .688 | 1.375 | 2.063 | 2.750 | 3.438 | 4.125 | 4.813 | 5.500 | 6.188 | 6.875 | 7.563 | 8.250 | 8.938 | 9.625 | 10.31 | 11.00 |
| 11 1/2 | .719 | 1.438 | 2.156 | 2.875 | 3.594 | 4.313 | 5.031 | 5.750 | 6.469 | 7.188 | 7.906 | 8.625 | 9.344 | 10.06 | 10.78 | 11.50 |
| 12 | .750 | 1.500 | 2.250 | 3.000 | 3.750 | 4.500 | 5.250 | 6.000 | 6.750 | 7.500 | 8.250 | 9.000 | 9.750 | 10.50 | 11.25 | 12.00 |
| 12 1/2 | .781 | 1.563 | 2.344 | 3.125 | 3.906 | 4.688 | 5.469 | 6.250 | 7.031 | 7.813 | 8.594 | 9.375 | 10.16 | 10.94 | 11.72 | 12.50 |
| 13 | .813 | 1.625 | 2.438 | 3.250 | 4.063 | 4.875 | 5.688 | 6.500 | 7.313 | 8.125 | 8.938 | 9.750 | 10.56 | 11.38 | 12.19 | 13.00 |
| 13 1/2 | .844 | 1.688 | 2.531 | 3.375 | 4.219 | 5.063 | 5.906 | 6.750 | 7.594 | 8.438 | 9.281 | 10.13 | 10.97 | 11.81 | 12.66 | 13.50 |
| 14 | .875 | 1.750 | 2.625 | 3.500 | 4.375 | 5.250 | 6.125 | 7.000 | 7.875 | 8.750 | 9.625 | 10.50 | 11.38 | 12.25 | 13.13 | 14.00 |
| 14 1/2 | .906 | 1.813 | 2.719 | 3.625 | 4.531 | 5.438 | 6.344 | 7.250 | 8.156 | 9.063 | 9.969 | 10.88 | 11.78 | 12.69 | 13.59 | 14.50 |
| 15 | .938 | 1.875 | 2.813 | 3.750 | 4.688 | 5.625 | 6.563 | 7.500 | 8.438 | 9.375 | 10.31 | 11.25 | 12.19 | 13.13 | 14.06 | 15.00 |
| 15 1/2 | .969 | 1.938 | 2.906 | 3.875 | 4.844 | 5.813 | 6.781 | 7.750 | 8.719 | 9.688 | 10.66 | 11.63 | 12.59 | 13.56 | 14.53 | 15.50 |
| 16 | 1.000 | 2.000 | 3.000 | 4.000 | 5.000 | 6.000 | 7.000 | 8.000 | 9.000 | 10.00 | 11.00 | 12.00 | 13.00 | 14.00 | 15.00 | 16.00 |
| 16 1/2 | 1.031 | 2.063 | 3.094 | 4.125 | 5.156 | 6.188 | 7.219 | 8.250 | 9.281 | 10.31 | 11.34 | 12.38 | 13.41 | 14.44 | 15.47 | 16.50 |
| 17 | 1.063 | 2.125 | 3.188 | 4.250 | 5.313 | 6.375 | 7.438 | 8.500 | 9.563 | 10.63 | 11.69 | 12.75 | 13.81 | 14.88 | 15.94 | 17.00 |
| 17 1/2 | 1.094 | 2.188 | 3.281 | 4.375 | 5.469 | 6.563 | 7.656 | 8.750 | 9.844 | 10.94 | 12.03 | 13.13 | 14.22 | 15.31 | 16.41 | 17.50 |
| 18 | 1.125 | 2.250 | 3.375 | 4.500 | 5.625 | 6.750 | 7.875 | 9.000 | 10.13 | 11.25 | 12.38 | 13.50 | 14.63 | 15.75 | 16.88 | 18.00 |
| 18 1/2 | 1.156 | 2.313 | 3.469 | 4.625 | 5.781 | 6.938 | 8.094 | 9.250 | 10.41 | 11.56 | 12.72 | 13.88 | 15.03 | 16.19 | 17.34 | 18.50 |
| 19 | 1.188 | 2.375 | 3.563 | 4.750 | 5.938 | 7.125 | 8.313 | 9.500 | 10.69 | 11.88 | 13.06 | 14.25 | 15.44 | 16.63 | 17.81 | 19.00 |
| 19 1/2 | 1.219 | 2.438 | 3.656 | 4.875 | 6.094 | 7.313 | 8.531 | 9.750 | 10.97 | 12.19 | 13.41 | 14.63 | 15.84 | 17.06 | 18.28 | 19.50 |
| 20 | 1.250 | 2.500 | 3.750 | 5.000 | 6.250 | 7.500 | 8.750 | 10.00 | 11.25 | 12.50 | 13.75 | 15.00 | 16.25 | 17.50 | 18.75 | 20.00 |

**Table 6.—Weight of Sheet Brass and Copper***

(Revere Copper and Brass Co., Rome, N. Y.)

| Thickness | | Width in Inches | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| B. & S. Gage | Inch | 1/16 | 1/8 | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 2 | 3 | 4 | 6 | 8 | 10 | 12 |
| | | Pounds per Lineal Foot of Brass Sheet | | | | | | | | | | | | | | | | | | |
| 6 | 0.1620 | 0.0372 | 0.0744 | 0.1115 | 0.1487 | 0.1859 | 0.2231 | 0.2603 | 0.2974 | 0.3718 | 0.4461 | 0.5205 | 0.5949 | 1.190 | 1.785 | 2.379 | 3.569 | 4.759 | 5.949 | 7.138 |
| 7 | .1443 | .0331 | .0662 | .0994 | .1325 | .1656 | .1987 | 2318 | .2649 | .3312 | .3974 | .4636 | .5299 | 1.060 | 1.590 | 2.119 | 3.179 | 4.239 | 5.299 | 6.358 |
| 8 | .1285 | .0295 | .0590 | .0885 | .1180 | .1475 | .1769 | .2064 | .2359 | .2949 | .3539 | .4129 | .4719 | 0.9437 | 1.416 | 1.887 | 2.831 | 3.775 | 4.719 | 5.662 |
| 9 | .1144 | .0263 | .0525 | .0788 | .1050 | .1313 | .1575 | .1838 | .2100 | .2625 | .3151 | .3676 | .4201 | .8402 | 1.260 | 1.680 | 2.520 | 3.361 | 4.201 | 5.041 |
| 10 | .1019 | .0234 | .0468 | .0702 | .0935 | .1169 | .1403 | .1637 | .1871 | .2339 | .2806 | .3274 | .3742 | .7484 | 1.123 | 1.497 | 2.245 | 2.993 | 3.742 | 4.490 |
| 11 | .0907 | .0208 | .0416 | .0625 | .0833 | .1041 | .1249 | .1457 | .1665 | .2082 | .2498 | .2914 | .3331 | .6661 | 0.9992 | 1.332 | 1.998 | 2.664 | 3.331 | 3.997 |
| 12 | .0808 | .0185 | .0371 | .0556 | .0742 | .0927 | .1113 | .1298 | .1483 | .1854 | .2225 | .2596 | .2967 | .5934 | .8901 | 1.187 | 1.780 | 2.374 | 2.967 | 3.560 |
| 13 | .0720 | .0165 | .0331 | .0496 | .0661 | .0826 | .0991 | .1157 | .1322 | .1652 | .1983 | .2313 | .2644 | .5288 | .7932 | 1.058 | 1.586 | 2.115 | 2.644 | 3.173 |
| 14 | .0641 | .0147 | .0294 | .0441 | .0588 | .0736 | .0883 | .1030 | .1177 | .1471 | .1765 | .2060 | .2354 | .4708 | .7061 | 0.9415 | 1.412 | 1.883 | 2.354 | 2.825 |
| 15 | .0571 | .0131 | .0262 | .0393 | .0524 | .0655 | .0786 | .0917 | .1048 | .1310 | .1573 | .1835 | .2097 | .4193 | .6290 | .8387 | 1.258 | 1.677 | 2.097 | 2.516 |
| 16 | .0508 | .0117 | .0233 | .0350 | .0466 | .0583 | .0700 | .0816 | .0933 | .1166 | .1399 | .1632 | .1865 | .3731 | .5596 | .7462 | 1.119 | 1.492 | 1.865 | 2.238 |
| 17 | .0453 | .0104 | .0208 | .0312 | .0416 | .0520 | .0624 | .0728 | .0832 | .1040 | .1248 | .1455 | .1663 | .3327 | .4990 | .6654 | 0.9980 | 1.331 | 1.663 | 1.996 |
| 18 | .0403 | .0092 | .0185 | .0278 | .0370 | .0462 | .0555 | .0647 | .0740 | .0925 | .1110 | .1295 | .1480 | .2960 | .4439 | .5919 | .8879 | 1.184 | 1.480 | 1.776 |
| 19 | .0359 | .0082 | .0165 | .0247 | .0330 | .0412 | .0494 | .0577 | .0659 | .0824 | .0989 | .1153 | .1318 | .2636 | .3958 | .5273 | .7909 | 1.055 | 1.318 | 1.582 |
| 20 | .0320 | .0073 | .0147 | .0220 | .0294 | .0367 | .0441 | .0514 | .0588 | .0734 | .0881 | .1028 | .1175 | .2350 | .3525 | .4700 | .7050 | 0.9400 | 1.175 | 1.410 |
| 21 | .0285 | .0065 | .0131 | .0196 | .0262 | .0327 | .0392 | .0458 | .0523 | .0654 | .0785 | .0916 | .1047 | .2093 | .3140 | .4186 | .6279 | .8372 | 1.047 | 1.256 |
| 22 | .0254 | .0058 | .0116 | .0174 | .0232 | .0290 | .0348 | .0406 | .0465 | .0581 | .0697 | .0813 | .0929 | .1858 | .2787 | .3716 | .5574 | .7432 | 0.9290 | 1.115 |
| 23 | .0226 | .0052 | .0104 | .0156 | .0208 | .0259 | .0311 | .0363 | .0415 | .0519 | .0622 | .0726 | .0830 | .1660 | .2490 | .3319 | .4979 | .6639 | .8299 | 0.9958 |
| 24 | .0201 | .0046 | .0092 | .0138 | .0185 | .0231 | .0277 | .0323 | .0369 | .0461 | .0554 | .0646 | .0738 | .1476 | .2214 | .2952 | .4428 | .5905 | .7381 | .8857 |
| 25 | .0179 | .0041 | .0082 | .0123 | .0164 | .0205 | .0247 | .0288 | .0329 | .0411 | .0493 | .0575 | .0657 | .1315 | .1972 | .2629 | .3944 | .5258 | .6573 | .7887 |
| 26 | .0159 | .0036 | .0073 | .0110 | .0146 | .0183 | .0219 | .0255 | .0292 | .0365 | .0438 | .0511 | .0584 | .1168 | .1752 | .2335 | .3503 | .4671 | .5838 | .7006 |
| 27 | .0142 | .0033 | .0065 | .0098 | .0130 | .0163 | .0196 | .0228 | .0261 | .0326 | .0391 | .0456 | .0521 | .1043 | .1564 | .2086 | .3129 | .4171 | .5214 | .6257 |
| 28 | .0126 | .0029 | .0058 | .0087 | .0116 | .0145 | .0174 | .0202 | .0231 | .0289 | .0347 | .0405 | .0463 | .0925 | .1388 | .1851 | .2776 | .3701 | .4627 | .5552 |
| 29 | .0113 | .0026 | .0052 | .0078 | .0104 | .0130 | .0156 | .0182 | .0208 | .0259 | .0311 | .0363 | .0415 | .0830 | .1245 | .1660 | .2490 | .3319 | .4149 | .4979 |
| 30 | .0100 | .0023 | .0046 | .0069 | .0092 | .0115 | .0138 | .0161 | .0184 | .0230 | .0275 | .0321 | .0367 | .0734 | .1102 | .1469 | .2203 | .2938 | .3672 | .4406 |
| 31 | .0089 | .0020 | .0041 | .0061 | .0082 | .0103 | .0124 | .0145 | .0165 | .0206 | .0248 | .0289 | .0330 | .0660 | .0991 | .1321 | .1982 | .2642 | .3303 | .3963 |
| 32 | .0080 | .0019 | .0038 | .0055 | .0074 | .0092 | .0111 | .0129 | .0148 | .0184 | .0221 | .0258 | .0295 | .0589 | .0884 | .1179 | .1769 | .2358 | .2948 | .3537 |
| 33 | .0071 | .0016 | .0033 | .0049 | .0066 | .0082 | .0099 | .0115 | .0131 | .0164 | .0197 | .0230 | .0262 | .0525 | .0788 | .1050 | .1575 | .2100 | .2625 | .3150 |
| 34 | .0063 | .0014 | .0029 | .0043 | .0058 | .0073 | .0088 | .0102 | .0117 | .0146 | .0175 | .0204 | .0234 | .0467 | .0701 | .0935 | .1403 | .1870 | .2337 | .2805 |
| 35 | .0056 | .0013 | .0026 | .0039 | .0052 | .0065 | .0078 | .0091 | .0104 | .0130 | .0156 | .0182 | .0208 | .0416 | .0625 | .0833 | .1249 | .1666 | .2082 | .2498 |
| 36 | .0050 | .0011 | .0023 | .0034 | .0046 | .0058 | .0070 | .0081 | .0093 | .0116 | .0139 | .0162 | .0185 | .0371 | .0556 | .0742 | .1113 | .1484 | .1854 | .2225 |
| 37 | .0045 | .0010 | .0021 | .0031 | .0041 | .0052 | .0062 | .0072 | .0083 | .0103 | .0124 | .0144 | .0165 | .0330 | .0495 | .0661 | .0991 | .1321 | .1651 | .1981 |
| 38 | .0040 | .0009 | .0018 | .0028 | .0037 | .0046 | .0055 | .0065 | .0074 | .0092 | .0111 | .0129 | .0147 | .0295 | .0442 | .0590 | .0884 | .1179 | .1474 | .1768 |
| 39 | .0035 | .0008 | .0016 | .0024 | .0033 | .0041 | .0049 | .0057 | .0066 | .0082 | .0098 | .0115 | .0131 | .0262 | .0394 | .0525 | .0788 | .1050 | .1312 | .1575 |
| 40 | .0031 | .0007 | .0014 | .0021 | .0029 | .0036 | .0044 | .0051 | .0058 | .0073 | .0088 | .0102 | .0117 | .0234 | .0351 | .0468 | .0701 | .0935 | .1169 | .1402 |
| 42 | .0025 | .0006 | .0011 | .0017 | .0023 | .0029 | .0035 | .0041 | .0046 | .0058 | .0070 | .0081 | .0093 | .0185 | .0278 | .0371 | .0556 | .0742 | .0927 | .1112 |
| 44 | .0020 | .0005 | .0009 | .0014 | .0018 | .0023 | .0028 | .0032 | .0037 | .0046 | .0056 | .0065 | .0074 | .0148 | .0223 | .0297 | .0445 | .0593 | .0742 | .0889 |
| 46 | .0016 | .0004 | .0007 | .0011 | .0014 | .0017 | .0021 | .0024 | .0028 | .0035 | .0042 | .0049 | .0056 | .0111 | .0167 | .0222 | .0334 | .0445 | .0556 | .0711 |
| 48 | .0012 | .0003 | .0006 | .0008 | .0011 | .0014 | .0017 | .0019 | .0022 | .0028 | .0033 | .0039 | .0044 | .0088 | .0132 | .0176 | .0264 | .0353 | .0441 | .0529 |

By combining various widths given, the weight of any size piece can be determined.

*Density of brass = 0.306 lb. per cu. in. To obtain the weight of sheet copper multiply the figures in the table by 1.0508.

*(Continued from p. 6–03.)*

length = $v = 9.4248\ (D^2 - d^2) = 37.699\ (Dt - t^2)$; weight of 1 ft. length = $wv = 9.4248\ w(D^2 - d^2) = 37.699\ w(Dt - t^2)$. In using Table 2 for tubes, multiply the coefficient for round rods by $(D^2 - d^2)$ or by $4(Dt - t^2)$.

**Spheres.**—Weight of sphere = $0.5236\ wD^3$; for hollow spheres multiply the coefficient for spheres in Table 2 by $(D^3 - d^3)$.

**Hexagons.**—To find weight of hexagonal bar multiply the weight of square bars by 0.866 (short diameter of hexagon = side of square).

**Octagons.**—To find weight of octagonal bar, multiply the weight of square bar by 0.8284 (short diameter of octagon = side of square).

**COMMERCIAL RODS AND BARS.**—Tables 15 and 16 give the weights and areas of commercial square, round and hexagonal steel bars. Tables 11, 17, 18 and 19 give the weights of non-ferrous rods and bars.

### Table 7.—Weight of Copper Sheets

(Revere Copper & Brass Co., Rome, N. Y.)

Thickness in Inches. Weight in Pounds per Square Foot.

| Thick. | Wt. | Thick. | Wt. | Thick. | Wt. | Thick. | Wt. | Thick. | Wt. | Thick. | Wt. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.001 | 0.046 | 0.005 | 0.232 | 0.009 | 0.417 | 0.040 | 1.854 | 0.080 | 3.709 | 0.300 | 13.91 |
| .002 | .093 | .006 | .278 | .010 | .463 | .050 | 2.318 | .090 | 4.173 | .400 | 18.547 |
| .003 | .139 | .007 | .324 | .020 | .927 | .060 | 2.78 | .100 | 4.637 | .500 | 23.184 |
| .004 | .185 | .008 | .371 | .030 | 1.391 | .070 | 3.245 | .200 | 9.273 | ....... | ....... |

### Table 8.—Weight of Everdur Plates and Sheets

(American Brass Co., Waterbury, Conn.)

| U. S. Std. Sheet Gage No. | Thickness, in. | Lb. per sq. ft. | Gage No. | Thickness, in. | Lb. per sq. ft. | Gage No. | Thickness, in. | Lb. per sq. ft. | Gage No. | Thickness, in. | Lb. per sq. ft. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 7–0 | 0.500 | 22.032 | 7 | 0.188 | 8.284 | 14 | 0.078 | 3.437 | 22 | 0.031 | 1.366 |
| 3–0 | .375 | 16.054 | 8 | .172 | 7.579 | 15 | .070 | 3.084 | 23 | .028 | 1.234 |
| 0 | .313 | 13.792 | 9 | .156 | 6.874 | 16 | .063 | 2.776 | 24 | .025 | 1.102 |
| 3 | .250 | 11.016 | 10 | .141 | 6.213 | 18 | .050 | 2.203 | 25 | .022 | 0.969 |
| 4 | .234 | 10.311 | 11 | .125 | 5.508 | 19 | .044 | 1.939 | 26 | .019 | .837 |
| 5 | .219 | 9.650 | 12 | .109 | 4.803 | 20 | .038 | 1.674 | ...... | ...... | ...... |
| 6 | .203 | 8.945 | 13 | .094 | 4.142 | 21 | .034 | 1.498 | ...... | ...... | ...... |

### Table 9.—Weight of Everdur Flats

(American Brass Co., Waterbury, Conn.)

| Width, in..... | 1 | 2 | 3 | 4 | 6 |
|---|---|---|---|---|---|
| Thickness, in. | Lb. per Lineal Foot | | | | |
| 1/4 | 0.918 | 1.836 | 2.754 | 3.672 | 5.508 |
| 3/8 | 1.377 | 2.754 | 4.131 | 5.508 | 8.262 |
| 1/2 | 1.836 | 3.672 | 5.508 | 7.344 | 11.016 |
| 3/4 | 2.754 | 5.508 | 8.262 | 11.016 | 16.524 |
| 1 | 3.672 | 7.344 | 11.016 | 14.688 | 22.032 |

### Table 10.—Weight of Aluminum Plates or Sheets.*

(Aluminum Company of America, Pittsburgh)

| B. & S. Gage No. | Thickness, in. | Lb. per sq. ft. | B. & S. Gage No. | Thickness, in. | Lb. per sq. ft. | B. & S. Gage No. | Thickness, in. | Lb. per sq. ft. | B. & S. Gage No. | Thickness, in. | Lb. per sq. ft. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0000 | 0.4600 | 6.48 | 8 | 0.1285 | 1.81 | 19 | 0.0359 | 0.506 | 30 | 0.001 | 0.141 |
| 000 | .4096 | 5.77 | 9 | .1144 | 1.61 | 20 | .0320 | .450 | 31 | .0089 | .126 |
| 00 | .3648 | 5.14 | 10 | .1019 | 1.44 | 21 | .0285 | .401 | 32 | .0080 | .113 |
| 0 | .3249 | 4.58 | 11 | .0907 | 1.28 | 22 | .0253 | .357 | 33 | .0071 | .100 |
| 1 | .2893 | 4.08 | 12 | .0808 | 1.14 | 23 | .0226 | .318 | 34 | .0063 | .0888 |
| 2 | .2576 | 3.63 | 13 | .0720 | 1.01 | 24 | .0201 | .283 | 35 | .0056 | .0790 |
| 3 | .2294 | 3.23 | 14 | .0641 | 0.903 | 25 | .0179 | .252 | 36 | .0050 | .0704 |
| 4 | .2043 | 2.88 | 15 | .0571 | .804 | 26 | .0159 | .225 | 37 | .0045 | .0627 |
| 5 | .1819 | 2.56 | 16 | .0508 | .716 | 27 | .0142 | .200 | 38 | .0040 | .0558 |
| 6 | .1620 | 2.28 | 17 | .0453 | .638 | 28 | .0126 | .178 | 39 | .0035 | .0497 |
| 7 | .1443 | 2.03 | 18 | .0403 | .568 | 29 | .0113 | .159 | 40 | .0031 | .0442 |

* For weights of aluminum sheets heavier than those shown in this table, see Table 11. For other data on aluminum see pages 4–25 to 4–40.

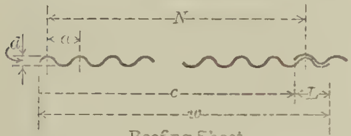
### **Table 11.—Weight of Bar or Sheet Aluminum** (Sp. Gr. 2.68)

(Aluminum Co. of America, Pittsburgh)

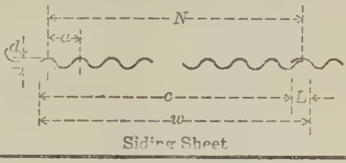
| Thickness, Side or Diam., in. | Sheets, lb. per sq. ft. | Square Bars, 1 ft. Long, lb. | Round Bars, 1 ft. Long, lb. | Thickness, Side or Diam., in. | Sheets, lb. per sq. ft. | Square Bars, 1 ft. Long, lb. | Round Bars, 1 ft. Long, lb. | Thickness, Side or Diam., in. | Sheets, lb. per sq. ft. | Square Bars, 1 ft. Long, lb. | Round Bars, 1 ft. Long, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/16 | 0.8803 | 0.005 | 0.004 | 3/4 | 10.564 | 0.659 | 0.518 | 1 7/16 | 20.248 | 2.423 | 1.903 |
| 1/8 | 1.7607 | .018 | .014 | 13/16 | 11.444 | .774 | .608 | 1 1/2 | 21.129 | 2.638 | 2.072 |
| 3/16 | 2.6411 | .041 | .032 | 7/8 | 12.325 | .898 | .705 | 1 9/16 | 22.009 | 2.862 | 2.248 |
| 1/4 | 3.5215 | .073 | .058 | 15/16 | 13.205 | 1.030 | .809 | 1 5/8 | 22.889 | 3.096 | 2.431 |
| 5/16 | 4.4019 | .115 | .090 | 1 | 14.086 | 1.172 | .921 | 1 11/16 | 23.770 | 3.339 | 2.622 |
| 3/8 | 5.2822 | .165 | .129 | 1 1/16 | 14.966 | 1.324 | 1.039 | 1 3/4 | 24.650 | 3.590 | 2.820 |
| 7/16 | 6.1626 | .224 | .176 | 1 1/8 | 15.846 | 1.484 | 1.165 | 1 13/16 | 25.530 | 3.852 | 3.025 |
| 1/2 | 7.0430 | .293 | .230 | 1 3/16 | 16.727 | 1.653 | 1.298 | 1 7/8 | 26.411 | 4.122 | 3.237 |
| 9/16 | 7.9234 | .371 | .291 | 1 1/4 | 17.607 | 1.832 | 1.439 | 1 15/16 | 27.291 | 4.401 | 3.457 |
| 5/8 | 8.8037 | .453 | .360 | 1 5/16 | 18.487 | 2.019 | 1.585 | 2 | 28.172 | 4.690 | 3.683 |
| 11/16 | 9.6841 | .554 | .435 | 1 3/8 | 19.368 | 2.217 | 1.741 | ..... | ........ | ........ | ....... |

### **Table 12.—Dimensions and Weights of Corrugated Aluminum Sheets**

Nominal pitch, 2 1/2 in. Depth of corrugation, $d$, 3/4 in.



Roofing Sheet



Siding Sheet

| No. of Corrugations | Overall Width, $w$, in | Covering Width, $c$, in. | Actual Pitch, $a$, in. | Side Lap, in. | End Lap, in. | Gage, B. & S. | Weight, lb. per lin. ft. | Weight, lb. per sq. ft. | Total Weight of One Sheet in Pounds For Various Standard Lengths | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | 5 ft. | 6 ft. | 7 ft. | 8 ft. | 9 ft. | 10 ft. | 11 ft. | 12 ft. |
| STANDARD ROOFING | | | | | | | | | | | | | | | | |
| 9 | 27 | 24 | 2.67 | 3 | 6 | 14 | 2.46 | 1.09 | 12.29 | 14.75 | 17.20 | 19.65 | 22.12 | 24.58 | 27.04 | 29.50 |
| 9 | 27 | 24 | 2.67 | 3 | 6 | 16 | 1.95 | 0.87 | 9.74 | 11.69 | 13.64 | 15.59 | 17.53 | 19.49 | 21.43 | 23.38 |
| 9 | 27 | 24 | 2.67 | 3 | 6 | 18 | 1.55 | 0.67 | 7.75 | 9.30 | 10.86 | 12.40 | 13.92 | 15.50 | 17.05 | 18.60 |
| 9 | 27 | 24 | 2.67 | 3 | 6 | 20 | 1.24 | 0.55 | 6.17 | 7.40 | 8.64 | 9.86 | 11.10 | 12.35 | 13.57 | 14.81 |
| 9 | 27 | 24 | 2.67 | 3 | 6 | 22 | 0.98 | 0.43 | 4.89 | 5.87 | 6.85 | 7.83 | 8.81 | 9.78 | 10.77 | 11.75 |
| STANDARD SIDING | | | | | | | | | | | | | | | | |
| 9 | 25 1/2 | 24 | 2.66 | 1 1/2 | 4 | 14 | 2.34 | 1.09 | 11.68 | 14.01 | 16.35 | 18.69 | 21.02 | 23.36 | 25.69 | 28.03 |
| 9 | 25 1/2 | 24 | 2.66 | 1 1/2 | 4 | 16 | 1.86 | 0.87 | 9.28 | 11.14 | 12.99 | 14.85 | 16.70 | 18.56 | 20.42 | 22.28 |
| 9 | 25 1/2 | 24 | 2.66 | 1 1/2 | 4 | 18 | 1.47 | 0.67 | 7.34 | 8.83 | 10.30 | 11.75 | 13.22 | 14.69 | 16.16 | 17.62 |
| 9 | 25 1/2 | 24 | 2.66 | 1 1/2 | 4 | 20 | 1.17 | 0.55 | 5.85 | 7.04 | 8.21 | 9.38 | 10.56 | 11.73 | 12.90 | 14.08 |
| 9 | 25 1/2 | 24 | 2.66 | 1 1/2 | 4 | 22 | 0.93 | 0.43 | 4.66 | 5.59 | 6.52 | 7.52 | 8.37 | 9.30 | 10.23 | 11.16 |
| EXTRA WIDE ROOFING | | | | | | | | | | | | | | | | |
| 14 | 40 1/4 | 37 1/2 | 2.67 | 3 | 6 | 14 | 3.69 | 1.09 | 18.45 | 22.13 | 25.85 | 29.54 | 33.23 | 36.92 | 40.62 | 44.31 |
| 14 | 40 1/4 | 37 1/2 | 2.67 | 3 | 6 | 16 | 2.93 | 0.87 | 14.65 | 17.54 | 20.40 | 23.42 | 26.35 | 29.27 | 32.20 | 35.13 |
| 14 | 40 1/4 | 37 1/2 | 2.67 | 3 | 6 | 18 | 2.33 | 0.67 | 11.63 | 13.95 | 16.28 | 18.60 | 20.93 | 23.26 | 25.58 | 27.91 |
| 14 | 40 1/4 | 37 1/2 | 2.67 | 3 | 6 | 20 | 1.85 | 0.55 | 9.23 | 11.08 | 12.92 | 14.76 | 16.62 | 18.45 | 20.31 | 22.15 |
| 14 | 40 1/4 | 37 1/2 | 2.67 | 3 | 6 | 22 | 1.47 | 0.43 | 7.34 | 8.81 | 10.28 | 11.75 | 13.22 | 14.69 | 16.16 | 17.63 |
| EXTRA WIDE SIDING | | | | | | | | | | | | | | | | |
| 14 | 38 3/4 | 37 1/2 | 2.66 | 1 1/2 | 4 | 14 | 3.58 | 1.09 | 17.80 | 22.01 | 25.06 | 28.64 | 32.22 | 35.80 | 39.38 | 42.96 |
| 14 | 38 3/4 | 37 1/2 | 2.66 | 1 1/2 | 4 | 16 | 2.85 | 0.87 | 14.23 | 17.08 | 19.92 | 22.77 | 25.61 | 28.46 | 31.31 | 34.15 |
| 14 | 38 3/4 | 37 1/2 | 2.66 | 1 1/2 | 4 | 18 | 2.25 | 0.67 | 11.27 | 13.53 | 15.78 | 18.03 | 20.29 | 22.54 | 24.80 | 27.05 |
| 14 | 38 3/4 | 37 1/2 | 2.66 | 1 1/2 | 4 | 20 | 1.80 | 0.55 | 8.98 | 10.77 | 12.57 | 14.36 | 16.16 | 17.95 | 19.75 | 21.54 |
| 14 | 38 3/4 | 37 1/2 | 2.66 | 1 1/2 | 4 | 22 | 1.43 | 0.43 | 7.14 | 8.57 | 10.00 | 11.42 | 12.85 | 14.28 | 15.71 | 17.14 |

### **Table 13.—Safe Uniformly Distributed Load on 2 1/2-inch Corrugated Sheet**

4S 3/4-H Aluminum Alloy

| Gage, B. & S. | Thickness, in. | Safe Loads for Various Span Lengths, lb. per sq. ft. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 24 in. | 30 in. | 36 in. | 42 in. | 48 in. | 54 in. | 60 in. | 66 in. | 72 in. |
| 14 | 0.064 | 346 | 222 | 154 | 113 | 86 | 68 | 55 | 46 | 38 |
| 16 | .051 | 276 | 177 | 123 | 90 | 69 | 55 | 44 | 36 | 31 |
| 18 | .040 | 217 | 139 | 96 | 71 | 54 | 43 | 35 | 28 | 24 |
| 20 | .032 | 173 | 111 | 77 | 56 | 43 | 34 | 28 | 23 | 19 |
| 22 | .025 | 135 | 87 | 60 | 44 | 34 | 27 | 22 | 18 | 15 |

### Table 14.—Approximate Weight of Sheet Zinc

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Thickness, in. | 0.002 | 0.004 | 0.008 | 0.010 | 0.012 | 0.014 | 0.016 | 0.018 | ....... |
| Wt. per sq. ft., lb. | 0.0748 | 0.150 | 0.299 | 0.374 | 0.449 | 0.524 | 0.599 | 0.673 | ....... |
| Thickness, in. | 0.020 | 0.024 | 0.028 | 0.032 | 0.036 | 0.040 | 0.045 | 0.050 | ....... |
| Wt. per sq. ft., lb. | 0.748 | 0.90 | 1.05 | 1.20 | 1.35 | 1.50 | 1.68 | 1.87 | ....... |
| Thickness, in. | 0.055 | 0.060 | 0.070 | 0.080 | 0.090 | 0.100 | 0.125 | 0.250 | 0.375 |
| Wt. per sq. ft., lb. | 2.06 | 2.24 | 2.62 | 2.99 | 3.37 | 3.74 | 4.68 | 9.35 | 14.03 |

### Table 15.—Weights and Areas of Square and Round Steel Bars

Steel = 489.6 lb. per cu. ft. Iron = 480 lb. per cu. ft. For weight of iron bars, multiply values in table by 0.98

| Size, in. | Weight, lb. per ft. | | Area, sq. in. | | Size, in. | Weight, lb. per ft. | | Area, sq. in. | | Size, in. | Weight, lb. per ft. | | Area, sq. in. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Square | Round | Square | Round | | Square | Round | Square | Round | | Square | Round | Square | Round |
| **0** | ...... | ...... | ...... | ...... | **3** | 30.600 | 24.033 | 9.000 | 7.0686 | **6** | 122.40 | 96.13 | 36.000 | 28.274 |
| 1/16 | 0.013 | 0.010 | 0.0039 | 0.0031 | 3 1/16 | 31.89 | 25.05 | 9.379 | 7.366 | 6 1/8 | 127.55 | 100.18 | 37.516 | 29.465 |
| 1/8 | .053 | .042 | .0156 | .0123 | 3 1/8 | 33.20 | 26.08 | 9.766 | 7.670 | 6 1/4 | 132.81 | 104.31 | 39.063 | 30.680 |
| 3/16 | .120 | .094 | .0352 | .0276 | 3 3/16 | 34.54 | 27.13 | 10.160 | 7.980 | 6 3/8 | 138.18 | 108.53 | 40.641 | 31.919 |
| 1/4 | .213 | .167 | .0625 | .0491 | 3 1/4 | 35.91 | 28.21 | 10.563 | 8.296 | 6 1/2 | 143.65 | 112.82 | 42.250 | 33.183 |
| 5/16 | .332 | .261 | .0977 | .0767 | 3 5/16 | 37.31 | 29.30 | 10.973 | 8.618 | 6 5/8 | 149.23 | 117.20 | 43.891 | 34.472 |
| 3/8 | .478 | .376 | .1406 | .1105 | 3 3/8 | 38.73 | 30.42 | 11.391 | 8.946 | 6 3/4 | 154.91 | 121.67 | 45.563 | 35.785 |
| 7/16 | .651 | .511 | .1914 | .1503 | 3 7/16 | 40.18 | 31.55 | 11.816 | 9.281 | 6 7/8 | 160.70 | 126.22 | 47.266 | 37.122 |
| 1/2 | .850 | .668 | .2500 | .1963 | 3 1/2 | 41.65 | 32.71 | 12.250 | 9.621 | **7** | 166.60 | 130.85 | 49.000 | 38.485 |
| 9/16 | 1.076 | .845 | .3164 | .2485 | 3 9/16 | 43.15 | 33.89 | 12.691 | 9.968 | 7 1/8 | 172.60 | 135.56 | 50.766 | 39.871 |
| 5/8 | 1.328 | 1.043 | .3906 | .3068 | 3 5/8 | 44.68 | 35.09 | 13.141 | 10.321 | 7 1/4 | 178.71 | 140.36 | 52.563 | 41.282 |
| 11/16 | 1.607 | 1.262 | .4727 | .3712 | 3 11/16 | 46.23 | 36.31 | 13.598 | 10.680 | 7 3/8 | 184.93 | 145.24 | 54.391 | 42.718 |
| 3/4 | 1.913 | 1.502 | .5625 | .4418 | 3 3/4 | 47.81 | 37.55 | 14.063 | 11.045 | 7 1/2 | 191.25 | 150.21 | 56.250 | 44.179 |
| 13/16 | 2.245 | 1.763 | .6602 | .5185 | 3 13/16 | 49.42 | 38.81 | 14.535 | 11.416 | 7 5/8 | 197.68 | 155.26 | 58.141 | 45.664 |
| 7/8 | 2.603 | 2.044 | .7656 | .6013 | 3 7/8 | 51.05 | 40.10 | 15.016 | 11.793 | 7 3/4 | 204.21 | 160.39 | 60.063 | 47.173 |
| 15/16 | 2.988 | 2.347 | .8789 | .6903 | 3 15/16 | 52.71 | 41.40 | 15.504 | 12.177 | 7 7/8 | 210.85 | 165.67 | 62.016 | 48.707 |
| **1** | 3.400 | 2.670 | 1.0000 | .7854 | **4** | 54.40 | 42.73 | 16.000 | 12.566 | **8** | 217.60 | 170.90 | 64.000 | 50.265 |
| 1 1/16 | 3.838 | 3.015 | 1.1289 | .8866 | 4 1/16 | 56.11 | 44.07 | 16.504 | 12.962 | 8 1/8 | 224.45 | 176.28 | 66.016 | 51.849 |
| 1 1/8 | 4.303 | 3.380 | 1.2656 | .9940 | 4 1/8 | 57.85 | 45.44 | 17.016 | 13.364 | 8 1/4 | 231.41 | 181.75 | 68.063 | 53.456 |
| 1 3/16 | 4.795 | 3.766 | 1.4102 | 1.1075 | 4 3/16 | 59.62 | 46.83 | 17.535 | 13.772 | 8 3/8 | 238.49 | 187.30 | 70.141 | 55.088 |
| 1 1/4 | 5.313 | 4.172 | 1.5625 | 1.2272 | 4 1/4 | 61.41 | 48.23 | 18.063 | 14.186 | 8 1/2 | 245.65 | 192.93 | 72.250 | 56.745 |
| 1 5/16 | 5.857 | 4.600 | 1.7227 | 1.3530 | 4 5/16 | 63.23 | 49.66 | 18.598 | 14.607 | 8 5/8 | 252.93 | 198.65 | 74.391 | 58.426 |
| 1 3/8 | 6.428 | 5.049 | 1.8906 | 1.4849 | 4 3/8 | 65.08 | 51.11 | 19.141 | 15.033 | 8 3/4 | 260.31 | 204.45 | 76.563 | 60.132 |
| 1 7/16 | 7.026 | 5.518 | 2.0664 | 1.6230 | 4 7/16 | 66.95 | 52.58 | 19.691 | 15.466 | 8 7/8 | 267.80 | 210.33 | 78.766 | 61.862 |
| 1 1/2 | 7.650 | 6.008 | 2.2500 | 1.7671 | 4 1/2 | 68.85 | 54.07 | 20.250 | 15.904 | **9** | 275.40 | 216.30 | 81.000 | 63.617 |
| 1 9/16 | 8.301 | 6.519 | 2.4414 | 1.9175 | 4 9/16 | 70.78 | 55.59 | 20.816 | 16.349 | 9 1/8 | 283.10 | 222.35 | 83.266 | 65.397 |
| 1 5/8 | 8.978 | 7.051 | 2.6406 | 2.0739 | 4 5/8 | 72.73 | 57.12 | 21.391 | 16.800 | 9 1/4 | 290.91 | 228.48 | 85.563 | 67.201 |
| 1 11/16 | 9.682 | 7.604 | 2.8477 | 2.2365 | 4 11/16 | 74.71 | 58.67 | 21.973 | 17.257 | 9 3/8 | 298.83 | 234.70 | 87.891 | 69.029 |
| 1 3/4 | 10.41 | 8.178 | 3.0625 | 2.4053 | 4 3/4 | 76.71 | 60.25 | 22.563 | 17.721 | 9 1/2 | 306.85 | 241.00 | 90.250 | 70.882 |
| 1 13/16 | 11.17 | 8.773 | 3.2852 | 2.5802 | 4 13/16 | 78.74 | 61.85 | 23.160 | 18.190 | 9 5/8 | 314.98 | 247.38 | 92.641 | 72.760 |
| 1 7/8 | 11.95 | 9.388 | 3.5156 | 2.7612 | 4 7/8 | 80.80 | 63.46 | 23.766 | 18.665 | 9 3/4 | 323.21 | 253.85 | 95.063 | 74.662 |
| 1 15/16 | 12.76 | 10.02 | 3.7539 | 2.9483 | 4 15/16 | 82.89 | 65.10 | 24.379 | 19.147 | 9 7/8 | 331.55 | 260.40 | 97.516 | 76.589 |
| **2** | 13.60 | 10.68 | 4.0000 | 3.1416 | **5** | 85.00 | 66.76 | 25.000 | 19.635 | **10** | 340.00 | 267.04 | 100.000 | 78.540 |
| 2 1/16 | 14.46 | 11.36 | 4.2539 | 3.3410 | 5 1/16 | 87.14 | 68.44 | 25.629 | 20.129 | 10 1/8 | 348.55 | 273.75 | 102.52 | 80.516 |
| 2 1/8 | 15.35 | 12.06 | 4.5156 | 3.5466 | 5 1/8 | 89.30 | 70.14 | 26.266 | 20.629 | 10 1/4 | 357.21 | 280.55 | 105.06 | 82.516 |
| 2 3/16 | 16.27 | 12.78 | 4.7852 | 3.7583 | 5 3/16 | 91.49 | 71.86 | 26.910 | 21.135 | 10 3/8 | 365.98 | 287.44 | 107.64 | 84.541 |
| 2 1/4 | 17.21 | 13.52 | 5.0625 | 3.9761 | 5 1/4 | 93.71 | 73.60 | 27.563 | 21.648 | 10 1/2 | 374.85 | 294.41 | 110.25 | 86.590 |
| 2 5/16 | 18.18 | 14.28 | 5.3477 | 4.2000 | 5 5/16 | 95.96 | 75.36 | 28.223 | 22.166 | 10 5/8 | 383.83 | 301.46 | 112.89 | 88.664 |
| 2 3/8 | 19.18 | 15.06 | 5.6406 | 4.4301 | 5 3/8 | 98.23 | 77.15 | 28.891 | 22.691 | 10 3/4 | 392.91 | 308.59 | 115.56 | 90.763 |
| 2 7/16 | 20.20 | 15.87 | 5.9414 | 4.6664 | 5 7/16 | 100.53 | 78.95 | 29.566 | 23.221 | 10 7/8 | 402.10 | 315.81 | 118.27 | 92.886 |
| 2 1/2 | 21.25 | 16.69 | 6.2500 | 4.9087 | 5 1/2 | 102.85 | 80.78 | 30.250 | 23.758 | **11** | 411.40 | 323.11 | 121.00 | 95.033 |
| 2 9/16 | 22.33 | 17.53 | 6.5664 | 5.1572 | 5 9/16 | 105.20 | 82.62 | 30.941 | 24.301 | 11 1/8 | 420.80 | 330.50 | 123.77 | 97.205 |
| 2 5/8 | 23.43 | 18.40 | 6.8906 | 5.4119 | 5 5/8 | 107.58 | 84.49 | 31.641 | 24.850 | 11 1/4 | 430.31 | 337.97 | 126.56 | 99.402 |
| 2 11/16 | 24.56 | 19.29 | 7.2227 | 5.6727 | 5 11/16 | 109.98 | 86.38 | 32.348 | 25.406 | 11 3/8 | 439.93 | 345.51 | 129.39 | 101.62 |
| 2 3/4 | 25.71 | 20.20 | 7.5625 | 5.9396 | 5 3/4 | 112.41 | 88.29 | 33.063 | 25.967 | 11 1/2 | 449.65 | 353.16 | 132.25 | 103.87 |
| 2 13/16 | 26.90 | 21.12 | 7.9102 | 6.2126 | 5 13/16 | 114.87 | 90.22 | 33.785 | 26.535 | 11 5/8 | 459.48 | 360.88 | 135.14 | 106.14 |
| 2 7/8 | 28.10 | 22.07 | 8.2656 | 6.4918 | 5 7/8 | 117.35 | 92.17 | 34.516 | 27.109 | 11 3/4 | 469.41 | 368.68 | 138.06 | 108.43 |
| 2 15/16 | 29.34 | 23.04 | 8.6289 | 6.7771 | 5 15/16 | 119.86 | 94.14 | 35.254 | 27.688 | 11 7/8 | 479.45 | 376.55 | 141.02 | 110.75 |
| | | | | | | | | | | **12** | 489.60 | 384.53 | 144.00 | 113.10 |

## Table 16.—Weights and Areas of Hexagon Steel

| Short Diam., in. | Weight per ft., lb. | Area, Sq. in. | Short Diam., in. | Weight per ft., lb. | Area, Sq. in. | Short Diam., in. | Weight per ft., lb. | Area, Sq. in. |
|---|---|---|---|---|---|---|---|---|
| 1/16 | 0.0115 | 0.0034 | 1 1/16 | 3.3316 | 0.9776 | 3 1/2 | 36.1528 | 10.6082 |
| 3/32 | .0259 | .0076 | 1 1/8 | 3.7351 | 1.0960 | 3 5/8 | 38.7828 | 11.3799 |
| 1/8 | .0461 | .0135 | 1 3/16 | 4.1616 | 1.2211 | 3 3/4 | 41.5034 | 12.1782 |
| 5/32 | .0721 | .0211 | 1 1/4 | 4.6114 | 1.3531 | 3 7/8 | 44.3141 | 13.0030 |
| 3/16 | .1037 | .0304 | 1 5/16 | 5.0841 | 1.4918 | 4 | 47.2188 | 13.8553 |
| 7/32 | .1412 | .0414 | 1 3/8 | 5.5797 | 1.6373 | 4 1/8 | 50.2174 | 14.7351 |
| 1/4 | .1845 | .0541 | 1 7/16 | 6.0987 | 1.7895 | 4 1/4 | 53.3062 | 15.6415 |
| 9/32 | .2334 | .0685 | 1 1/2 | 6.6401 | 1.9484 | 4 3/8 | 56.4889 | 16.5734 |
| 5/16 | .2882 | .0846 | 1 9/16 | 7.2053 | 2.1142 | 4 1/2 | 59.7618 | 17.5358 |
| 11/32 | .3487 | .1023 | 1 5/8 | 7.7930 | 2.2867 | 4 5/8 | 63.1287 | 18.5237 |
| 3/8 | .4150 | .1218 | 1 11/16 | 8.4040 | 2.4660 | 4 3/4 | 66.5895 | 19.5392 |
| 13/32 | .4871 | .1429 | 1 3/4 | 9.0383 | 2.6521 | 4 7/8 | 70.1367 | 20.5800 |
| 7/16 | .5649 | .1658 | 1 13/16 | 9.6955 | 2.8449 | 5 | 73.7817 | 21.6496 |
| 15/32 | .6485 | .1903 | 1 7/8 | 10.3757 | 3.0445 | 5 1/8 | 77.5168 | 22.7455 |
| 1/2 | .7378 | .2165 | 1 15/16 | 11.0787 | 3.2508 | 5 1/4 | 81.3458 | 23.8691 |
| 17/32 | .8329 | .2444 | 2 | 11.8051 | 3.4639 | 5 3/8 | 85.2651 | 25.0191 |
| 9/16 | .9317 | .2740 | 2 1/16 | 12.5544 | 3.6838 | 5 1/2 | 89.2745 | 26.1956 |
| 19/32 | 1.0404 | .3053 | 2 1/8 | 13.3269 | 3.9105 | 5 5/8 | 93.3779 | 27.3996 |
| 5/8 | 1.1529 | .3383 | 2 3/16 | 14.1224 | 4.1439 | 5 3/4 | 97.5752 | 28.6312 |
| 21/32 | 1.2710 | .3729 | 2 1/4 | 14.9408 | 4.3841 | 5 7/8 | 101.866 | 29.8904 |
| 11/16 | 1.3949 | .4093 | 2 5/16 | 15.7822 | 4.6309 | 6 | 106.244 | 31.1749 |
| 23/32 | 1.5246 | .4474 | 2 3/8 | 16.6468 | 4.8846 | 6 1/4 | 115.285 | 33.8278 |
| 3/4 | 1.6601 | .4871 | 2 7/16 | 17.5347 | 5.1452 | 6 1/2 | 124.690 | 36.5876 |
| 25/32 | 1.8013 | .5286 | 2 1/2 | 18.4452 | 5.4123 | 6 3/4 | 134.468 | 39.4565 |
| 13/16 | 1.9483 | .5717 | 2 5/8 | 20.3361 | 5.9672 | 7 | 144.614 | 42.4336 |
| 27/32 | 2.1011 | .6165 | 2 3/4 | 22.3190 | 6.5490 | 7 1/2 | 166.010 | 48.7118 |
| 7/8 | 2.2596 | .6630 | 2 7/8 | 24.3940 | 7.1579 | 8 | 188.879 | 55.4222 |
| 29/32 | 2.4238 | .7112 | 3 | 26.5614 | 7.7938 | 9 | 239.051 | 70.1441 |
| 15/16 | 2.5939 | .7611 | 3 1/8 | 28.8209 | 8.4568 | 10 | 295.127 | 86.5982 |
| 31/32 | 2.7697 | .8127 | 3 1/4 | 31.1728 | 9.1470 | 11 | 357.102 | 104.7834 |
| 1 | 2.9513 | .8660 | 3 3/8 | 33.6168 | 9.8641 | 12 | 424.991 | 124.7041 |

## Table 17.—Weight per Foot of Brass Rods, Pounds.*

(Revere Copper & Brass Co., Rome, N. Y.)

| In. | Round | Square | In. | Round | Square | In. | Round | Square |
|---|---|---|---|---|---|---|---|---|
| 1/16 | 0.0113 | 0.0144 | 5/8 | 1.132 | 1.441 | 1 9/16 | 7.073 | 9.006 |
| 1/8 | .0453 | .0576 | 21/32 | 1.247 | 1.587 | 1 5/8 | 7.651 | 9.741 |
| 5/32 | .0707 | .0898 | 11/16 | 1.369 | 1.744 | 1 11/16 | 8.250 | 10.50 |
| 3/16 | .1019 | .1297 | 23/32 | 1.496 | 1.907 | 1 3/4 | 8.873 | 11.30 |
| 7/32 | .1386 | .1769 | 3/4 | 1.630 | 2.075 | 1 13/16 | 9.518 | 12.12 |
| 1/4 | .1811 | .2306 | 13/16 | 1.913 | 2.435 | 1 7/8 | 10.19 | 12.97 |
| 9/32 | .2291 | .2912 | 7/8 | 2.218 | 2.824 | 1 15/16 | 10.88 | 13.85 |
| 5/16 | .2829 | .3602 | 15/16 | 2.546 | 3.242 | 2 | 11.59 | 14.76 |
| 11/32 | .3423 | .4364 | 1 | 2.897 | 3.689 | 2 1/8 | 13.08 | 16.66 |
| 3/8 | .4074 | .5188 | 1 1/16 | 3.271 | 4.164 | 2 1/4 | 14.67 | 18.68 |
| 13/32 | .4780 | .6079 | 1 1/8 | 3.667 | 4.669 | 2 3/8 | 16.34 | 20.81 |
| 7/16 | .5546 | .7061 | 1 3/16 | 4.086 | 5.202 | 2 1/2 | 18.11 | 23.06 |
| 15/32 | .6364 | .8112 | 1 1/4 | 4.527 | 5.764 | 2 5/8 | 19.96 | 25.42 |
| 1/2 | .7243 | .9222 | 1 5/16 | 4.991 | 6.355 | 2 3/4 | 21.91 | 27.90 |
| 17/32 | .8175 | 1.040 | 1 3/8 | 5.478 | 6.974 | 2 7/8 | 23.95 | 30.49 |
| 9/16 | .9167 | 1.167 | 1 7/16 | 5.987 | 7.623 | 3 | 26.08 | 33.20 |
| 19/32 | 1.021 | 1.301 | 1 1/2 | 6.519 | 8.390 | ...... | .......... | .......... |

* Weight per cubic inch = 0.3074 lb. Specific gravity = 8.509.

## Table 18.—Weight of Everdur Round and Hexagon Rods

(American Brass Co., Waterbury, Conn.)

| Diam.,* in. | Lb. per ft. |  | Diam.,* in. | Lb. per ft. |  | Diam.,* in. | Lb. per ft. |  |
|---|---|---|---|---|---|---|---|---|
|  | Round | Hexagon |  | Round | Hexagon |  | Round | Hexagon |
| 1/16 | 0.01138 | .......... | 1/2 | 0.7212 | 0.7950 | 1 1/4 | 4.506 | 4.969 |
| 1/8 | .04517 | .......... | 9/16 | .9125 | .......... | 1 1/2 | 6.489 | 7.155 |
| 3/16 | .1013 | .......... | 5/8 | 1.127 | 1.242 | 1 3/4 | 8.832 | 9.739 |
| 1/4 | .1803 | 0.1986 | 11/16 | 1.363 | .......... | 2 | 11.536 | 12.720 |
| 5/16 | .2816 | .3107 | 3/4 | 1.622 | 1.789 | 2 1/4 | 14.600 | .......... |
| 3/8 | .4054 | .4471 | 7/8 | 2.208 | .......... | 2 1/2 | 18.025 | .......... |
| 7/16 | .5519 | .......... | 1 | 2.884 | 3.180 | 2 3/4 | 21.810 | .......... |

* Across flats of Hexagon.

### Table 19.—Weight per Foot of Copper Rods, Pounds

(Revere Copper & Brass Co., Rome, N. Y.)

| Diam., in. | Round | Square | Diam., in. | Round | Square | Diam., in. | Round | Square |
|---|---|---|---|---|---|---|---|---|
| 1/8 | 0.0474 | 0.0603 | 1 5/8 | 8.002 | 10.19 | 3 1/4 | 32.01 | 40.75 |
| 1/4 | .1894 | .2412 | 1 3/4 | 9.281 | 11.82 | 3 1/2 | 37.12 | 47.27 |
| 3/8 | .4261 | .5426 | 1 7/8 | 10.65 | 13.56 | 3 3/4 | 42.61 | 54.26 |
| 1/2 | .7576 | .9646 | 2 | 12.12 | 15.43 | 4 | 48.49 | 61.73 |
| 5/8 | 1.184 | 1.507 | 2 1/8 | 13.68 | 17.42 | 4 1/4 | 54.74 | 69.69 |
| 3/4 | 1.705 | 2.170 | 2 1/4 | 15.34 | 19.53 | 4 1/2 | 61.37 | 78.13 |
| 7/8 | 2.320 | 2.954 | 2 3/8 | 17.09 | 21.76 | 4 3/4 | 68.37 | 87.06 |
| 1 | 3.030 | 3.858 | 2 1/2 | 18.94 | 24.12 | 5 | 75.76 | 96.46 |
| 1 1/8 | 3.835 | 4.883 | 2 5/8 | 20.88 | 26.59 | 5 1/4 | 83.53 | 106.3 |
| 1 1/4 | 4.735 | 6.029 | 2 3/4 | 22.92 | 29.18 | 5 1/2 | 91.67 | 116.7 |
| 1 3/8 | 5.729 | 7.295 | 2 7/8 | 25.05 | 31.89 | 5 3/4 | 100.2 | 127.6 |
| 1 1/2 | 6.818 | 8.681 | 3 | 27.27 | 34.73 | 6 | 109.1 | 138.9 |

For weight of octagon rod, multiply weight of round rod by 1.084.
For weight of hexagon rod, multiply weight of round rod by 1.12.

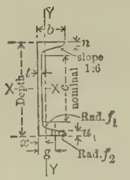


**Fig. 1. Channel**

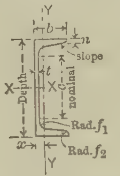


**Fig. 2. Car Channel**

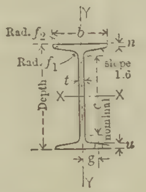


**Fig. 3. I-Beam**

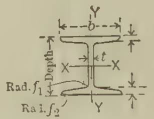


**Fig. 4. H-Beam**

STRUCTURAL SECTIONS.—Tables 20 to 26 give dimensions and properties of aluminum structural shapes. For similar data on steel structural shapes see pp. 17–47 to 17–80.

### Table 20.—Dimensions and Properties of Aluminum Channels

Maximum, intermediate and minimum sizes and weights

(Aluminum Co. of America)

| Depth, in. | Flange Width, $b$, in. | Weight, lb. per ft. | Thickness, $t$, in. | Area, sq. in. | Dimensions, in. See Fig. 1 $n$ | $f_1$ | $f_2$ | $c$ | Axis XX * $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | Axis YY * $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | $x$, in. | Rivet Data Diam., in. | $g$, in. | $u$, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3 | 1.410 | 1.46 | 0.170 | 1.21 | 0.170 | 0.27 | 0.10 | 1 3/4 | 1.66 | 1.10 | 1.17 | 0.20 | 0.20 | 0.40 | 0.44 | 1/2 | 7/8 | 1/4 |
| " | 1.498 | 1.78 | .258 | 1.47 | .170 | .27 | .10 | 1 3/4 | 1.85 | 1.24 | 1.12 | .25 | .23 | .41 | .44 | 1/2 | 7/8 | 1/4 |
| " | 1.596 | 2.13 | .356 | 1.76 | .170 | .27 | .10 | 1 3/4 | 2.07 | 1.38 | 1.08 | .31 | .27 | .42 | .46 | 1/2 | 7/8 | 1/4 |
| 4 | 1.580 | 1.90 | .180 | 1.57 | .180 | .28 | .11 | 2 3/4 | 3.83 | 1.92 | 1.56 | .32 | .28 | .45 | .46 | 1/2 | 1 | 5/16 |
| " | 1.467 | 2.22 | .247 | 1.84 | .180 | .28 | .11 | 2 3/4 | 4.19 | 2.10 | 1.51 | .37 | .31 | .45 | .45 | 1/2 | 1 | 5/16 |
| " | 1.720 | 2.58 | .320 | 2.13 | .180 | .28 | .11 | 2 3/4 | 4.58 | 2.29 | 1.47 | .43 | .34 | .45 | .46 | 1/2 | 1 | 5/16 |
| 5 | 1.750 | 2.38 | .190 | 1.97 | .190 | .29 | .11 | 3 3/4 | 7.49 | 3.00 | 1.95 | .48 | .38 | .49 | .43 | 1/2 | 1 1/8 | 5/16 |
| " | 1.885 | 3.20 | .325 | 2.64 | .190 | .29 | .11 | 3 3/4 | 8.90 | 3.56 | 1.83 | .63 | .45 | .49 | .48 | 1/2 | 1 1/8 | 5/16 |
| " | 2.032 | 4.09 | .472 | 3.38 | .190 | .29 | .11 | 3 3/4 | 10.43 | 4.17 | 1.76 | .81 | .53 | .49 | .51 | 1/2 | 1 1/8 | 5/16 |
| 6 | 1.920 | 2.91 | .200 | 2.40 | .200 | .30 | .12 | 4 1/2 | 13.12 | 4.37 | 2.34 | .69 | .49 | .54 | .51 | 5/8 | 1 1/8 | 5/16 |
| " | 2.034 | 3.73 | .314 | 3.09 | .200 | .30 | .12 | 4 1/2 | 15.18 | 5.06 | 2.22 | .87 | .56 | .53 | .50 | 5/8 | 1 1/8 | 3/8 |
| " | 2.090 | 3.47 | .210 | 2.87 | .210 | .31 | .13 | 5 1/2 | 21.27 | 6.08 | 2.72 | .97 | .63 | .58 | .54 | 5/8 | 1 1/4 | 3/8 |
| 7 | 2.110 | 3.64 | .230 | 3.01 | .210 | .31 | .13 | 5 1/2 | 21.84 | 6.24 | 2.69 | 1.01 | .64 | .58 | .54 | 5/8 | 1 1/4 | 7/16 |
| " | 2.299 | 5.24 | .419 | 4.33 | .210 | .31 | .13 | 5 1/2 | 27.24 | 7.78 | 2.51 | 1.38 | .78 | .56 | .53 | 5/8 | 1 1/4 | 7/16 |
| " | 2.509 | 7.02 | .629 | 5.80 | .210 | .31 | .13 | 5 1/2 | 33.25 | 9.50 | 2.39 | 1.83 | .95 | .56 | .58 | 5/8 | 1 1/2 | 7/16 |
| 8 | 2.260 | 4.08 | .220 | 3.38 | .220 | .32 | .13 | 6 1/4 | 32.57 | 8.14 | 3.11 | 1.32 | .78 | .63 | .57 | 3/4 | 1 3/8 | 3/8 |
| " | 2.343 | 4.89 | .303 | 4.04 | .220 | .32 | .13 | 6 1/4 | 36.11 | 9.03 | 2.99 | 1.53 | .85 | .61 | .55 | 3/4 | 1 3/8 | 3/8 |
| " | 2.619 | 7.56 | .579 | 6.25 | .220 | .32 | .13 | 6 1/4 | 47.89 | 11.97 | 2.77 | 2.22 | 1.09 | .60 | .59 | 3/4 | 1 1/2 | 7/16 |
| 9 | 2.430 | 4.74 | .230 | 3.91 | .230 | .33 | .14 | 7 1/4 | 47.68 | 10.60 | 3.49 | 1.75 | 0.96 | .67 | .60 | 3/4 | 1 3/8 | 7/16 |
| " | 2.485 | 5.34 | .285 | 4.41 | .230 | .33 | .14 | 7 1/4 | 51.02 | 11.34 | 3.40 | 1.93 | 1.01 | .66 | .59 | 3/4 | 1 3/8 | 7/16 |
| " | 2.812 | 8.90 | .612 | 7.35 | .230 | .33 | .14 | 7 1/4 | 70.89 | 15.75 | 3.11 | 2.94 | 1.34 | .63 | .61 | 3/4 | 1 1/2 | 1/2 |
| 10 | 2.600 | 5.43 | .240 | 4.49 | .240 | .34 | .14 | 8 1/4 | 67.37 | 13.47 | 3.87 | 2.28 | 1.16 | .71 | .63 | 3/4 | 1 1/2 | 7/16 |
| " | 2.739 | 7.11 | .379 | 5.88 | .240 | .34 | .14 | 8 1/4 | 78.95 | 15.79 | 3.66 | 2.81 | 1.32 | .69 | .61 | 3/4 | 1 1/2 | 7/16 |
| " | 3.033 | 10.67 | .673 | 8.82 | .240 | .34 | .14 | 8 1/4 | 103.45 | 20.69 | 3.43 | 3.95 | 1.66 | .67 | .65 | 3/4 | 1 3/4 | 1/2 |
| 12 | 2.940 | 7.33 | .280 | 6.06 | .280 | .38 | .17 | 10 | 128.96 | 21.49 | 4.61 | 3.87 | 1.73 | .80 | .70 | 7/8 | 1 3/4 | 1/2 |
| " | 3.047 | 8.89 | .387 | 7.35 | .280 | .38 | .17 | 10 | 144.37 | 24.06 | 4.43 | 4.47 | 1.89 | .78 | .67 | 7/8 | 1 3/4 | 1/2 |
| " | 3.292 | 12.45 | .632 | 10.29 | .280 | .38 | .17 | 10 | 179.65 | 29.94 | 4.18 | 5.82 | 2.24 | .75 | .69 | 7/8 | 2 | 5/8 |

* $I$ = moment of inertia; $S$ = section modulus; $r$ = radius of gyration; $x$ = distance to neutral axis.

## Table 21.—Dimensions and Properties of Aluminum Car Channels

(Aluminum Co. of America)

| Depth, in. | Flange Width, $b$, in. | Weight, lb. per ft. | Thickness, $t$, in. | Area, sq. in. | Dimensions, in. See Fig. 2 | | | | | Axis XX* | | | Axis YY * | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Slope | $n$ | $f_1$ | $f_2$ | $c$ | $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | $x$, in. |
| 3 | 2.00 | 2.61 | 0.250 | 2.15 | 1 : 12 | 0.250 | 0.25 | ..... | 1.71 | 3.15 | 2.10 | 1.21 | 0.69 | 0.53 | 0.57 | 0.70 |
| 4 | 2.50 | 3.41 | .318 | 2.82 | 1 : 35 | .313 | .38 | 0.13 | 2.50 | 6.84 | 3.42 | 1.56 | 1.62 | 0.95 | 0.76 | 0.81 |
| 6 | 3.50 | 6.34 | .375 | 5.24 | 1 : 50 | .413 | .48 | .42 | 4.10 | 29.94 | 9.98 | 2.39 | 5.60 | 2.34 | 1.03 | 1.10 |
| 7 | 3.00 | 6.41 | .360 | 5.29 | 1 : 18 | .360 | .55 | .20 | 4.89 | 40.35 | 11.53 | 2.76 | 3.79 | 1.78 | 0.85 | 0.87 |
| 8 | 3.00 | 7.13 | .380 | 5.89 | 1 : 18 | .380 | .55 | .20 | 5.85 | 57.02 | 14.26 | 3.11 | 4.12 | 1.90 | .84 | .83 |
| 10 | 3.500 | 8.84 | .375 | 7.30 | 1 : 9 | .375 | .625 | .188 | 7 1/2 | 109.62 | 21.92 | 3.88 | 7.19 | 2.80 | .99 | .93 |
| “ | 3.563 | 9.59 | .438 | 7.93 | 1 : 9 | .375 | .625 | .188 | 7 1/2 | 114.87 | 22.97 | 3.81 | 7.73 | 2.93 | .99 | .92 |
| “ | 3.625 | 10.34 | .500 | 8.55 | 1 : 9 | .375 | .625 | .188 | 7 1/2 | 120.03 | 24.01 | 3.75 | 8.25 | 3.04 | .98 | .91 |

* $I$ = moment of inertia; $S$ = section modulus; $r$ = radius of gyration; $x$ = distance to neutral axis.

## Table 22.—Dimensions and Properties of Aluminum I-beams

(Aluminum Co. of America)

| Depth of Beam, in. | Flange Width, $b$, in. | Weight, lb. per ft. | Thickness, $t$, in. | Area, sq. in. | Dimensions, in. See Fig. 3 | | | | Axis XX * | | | Axis YY * | | | Riveting Data, in. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | $n$ | $f_1$ | $f_2$ | $c$ | $I$, in.$^4$ | $S$, in.$^3$ | $r$ | $I$, in.$^4$ | $S$, in.$^3$ | $r$ | Diam. | $g$ | $u$ |
| 3 | 2.330 | 2.02 | 0.170 | 1.67 | 0.170 | 0.27 | 0.10 | 1 3/4 | 2.52 | 1.68 | 1.23 | 0.46 | 0.39 | 0.52 | 3/8 | 3/4 | 5/16 |
| “ | 2.411 | 2.31 | .251 | 1.91 | .170 | .27 | .10 | 1 3/4 | 2.71 | 1.80 | 1.19 | .51 | .42 | .52 | 3/8 | 3/4 | 5/16 |
| “ | 2.509 | 2.67 | .349 | 2.21 | .170 | .27 | .10 | 1 3/4 | 2.93 | 1.95 | 1.15 | .59 | .47 | .52 | 3/8 | 3/4 | 5/16 |
| 4 | 2.660 | 2.72 | .190 | 2.25 | .190 | .29 | .11 | 2 3/4 | 6.06 | 3.03 | 1.64 | .76 | .57 | .58 | 1/2 | 3/4 | 5/16 |
| “ | 2.723 | 3.03 | .253 | 2.50 | .190 | .29 | .11 | 2 3/4 | 6.40 | 3.20 | 1.60 | .82 | .61 | .57 | 1/2 | 3/4 | 5/16 |
| “ | 2.796 | 3.38 | .326 | 2.79 | .190 | .29 | .11 | 2 3/4 | 6.79 | 3.39 | 1.56 | .90 | .65 | .57 | 1/2 | 3/4 | 5/16 |
| “ | 2.870 | 3.74 | .400 | 3.09 | .190 | .29 | .11 | 2 3/4 | 7.18 | 3.59 | 1.52 | .99 | .69 | .57 | 1/2 | 3/4 | 5/16 |
| 5 | 3.000 | 3.53 | .210 | 2.92 | .210 | .31 | .13 | 3 1/2 | 12.26 | 4.90 | 2.05 | 1.21 | .81 | .64 | 1/2 | 7/8 | 3/8 |
| “ | 3.137 | 4.36 | .347 | 3.60 | .210 | .31 | .13 | 3 1/2 | 13.69 | 5.48 | 1.95 | 1.41 | .90 | .63 | 1/2 | 7/8 | 3/8 |
| “ | 3.284 | 5.25 | .494 | 4.34 | .210 | .31 | .13 | 3 1/2 | 15.22 | 6.09 | 1.87 | 1.66 | 1.01 | .62 | 1/2 | 7/8 | 3/8 |
| 6 | 3.330 | 4.43 | .230 | 3.66 | .230 | .33 | .14 | 4 1/2 | 22.08 | 7.36 | 2.46 | 1.82 | 1.09 | .71 | 5/8 | 1 | 3/8 |
| “ | 3.443 | 5.25 | .343 | 4.34 | .230 | .33 | .14 | 4 1/2 | 24.11 | 8.04 | 2.36 | 2.04 | 1.19 | .69 | 5/8 | 1 | 3/8 |
| “ | 3.565 | 6.13 | .465 | 5.07 | .230 | .33 | .14 | 4 1/2 | 26.31 | 8.77 | 2.28 | 2.31 | 1.30 | .68 | 5/8 | 1 | 3/8 |
| 7 | 3.660 | 5.42 | .250 | 4.48 | .250 | .35 | .15 | 5 1/4 | 36.69 | 10.48 | 2.86 | 2.63 | 1.44 | .77 | 5/8 | 1 1/8 | 3/8 |
| “ | 3.755 | 6.23 | .345 | 5.15 | .250 | .35 | .15 | 5 1/4 | 39.40 | 11.26 | 2.77 | 2.88 | 1.53 | .75 | 5/8 | 1 1/8 | 3/8 |
| “ | 3.860 | 7.12 | .450 | 5.88 | .250 | .35 | .15 | 5 1/4 | 42.40 | 12.12 | 2.69 | 3.17 | 1.64 | .73 | 5/8 | 1 1/8 | 3/8 |
| 8 | 4.000 | 6.53 | .270 | 5.40 | .270 | .37 | .16 | 6 1/4 | 57.55 | 14.39 | 3.27 | 3.73 | 1.86 | .83 | 3/4 | 1 1/8 | 7/16 |
| “ | 4.079 | 7.30 | .349 | 6.03 | .270 | .37 | .16 | 6 1/4 | 60.92 | 15.23 | 3.18 | 3.99 | 1.95 | .81 | 3/4 | 1 1/8 | 7/16 |
| “ | 4.171 | 8.19 | .441 | 6.77 | .270 | .37 | .16 | 6 1/4 | 64.85 | 16.21 | 3.10 | 4.31 | 2.07 | .80 | 3/4 | 1 1/8 | 7/16 |
| “ | 4.262 | 9.07 | .532 | 7.49 | .270 | .37 | .16 | 6 1/4 | 68.73 | 17.18 | 3.03 | 4.66 | 2.19 | .79 | 3/4 | 1 1/8 | 1/2 |
| 9 | 4.330 | 7.72 | .290 | 6.38 | .290 | .39 | .17 | 7 | 85.90 | 19.09 | 3.67 | 5.09 | 2.35 | .89 | 3/4 | 1 1/4 | 1/2 |
| “ | 4.437 | 8.89 | .397 | 7.35 | .290 | .39 | .17 | 7 | 92.40 | 20.53 | 3.55 | 5.54 | 2.50 | .87 | 3/4 | 1 1/4 | 1/2 |
| “ | 4.601 | 10.68 | .561 | 8.82 | .290 | .39 | .17 | 7 | 102.36 | 22.75 | 3.41 | 6.30 | 2.74 | .85 | 3/4 | 1 1/4 | 1/2 |
| 10 | 4.660 | 9.01 | .310 | 7.45 | .310 | .41 | .19 | 8 | 123.39 | 24.68 | 4.07 | 6.78 | 2.91 | .95 | 3/4 | 1 3/8 | 1/2 |
| “ | 4.797 | 10.67 | .447 | 8.82 | .310 | .41 | .19 | 8 | 134.81 | 26.96 | 3.91 | 7.50 | 3.13 | .92 | 3/4 | 1 3/8 | 1/2 |
| “ | 4.944 | 12.45 | .594 | 10.29 | .310 | .41 | .19 | 8 | 147.06 | 29.41 | 3.78 | 8.36 | 3.38 | .90 | 3/4 | 1 3/8 | 1/2 |
| 12 | 5.000 | 11.31 | .350 | 9.35 | .350 | .45 | .21 | 9 3/4 | 218.13 | 36.35 | 4.83 | 9.35 | 3.74 | 1.00 | 3/4 | 1 1/2 | 9/16 |
| “ | 5.078 | 12.44 | .428 | 10.28 | .350 | .45 | .21 | 9 3/4 | 229.36 | 38.23 | 4.72 | 9.87 | 3.89 | 0.98 | 3/4 | 1 1/2 | 9/16 |
| “ | 5.250 | 14.49 | .460 | 11.97 | .460 | .56 | .28 | 9 1/4 | 272.15 | 45.36 | 4.77 | 13.54 | 5.16 | 1.06 | 3/4 | 1 1/2 | 3/4 |
| “ | 5.355 | 16.01 | .565 | 13.23 | .460 | .56 | .28 | 9 1/4 | 287.27 | 47.88 | 4.66 | 14.50 | 5.42 | 1.05 | 3/4 | 1 1/2 | 3/4 |
| “ | 5.477 | 17.78 | .687 | 14.70 | .460 | .56 | .28 | 9 1/4 | 304.84 | 50.81 | 4.56 | 15.71 | 5.74 | 1.03 | 3/4 | 1 3/4 | 3/4 |

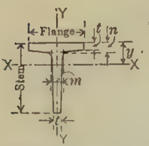
* $I$ = moment of inertia; $S$ = section modulus; $r$ = radius of gyration, in.

## Table 23.—Dimensions and Properties of Aluminum H-beams

(Aluminum Co. of America)

| Depth in. | Flange Width, $b$, in. | Weight, lb. per ft. | Thickness, $t$, in. | Area, sq. in. | Dimensions, in. See Fig. 4 | | | | Axis XX* | | | Axis YY* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | $n$ | $m$ | $f_1$ | $f_2$ | $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. |
| 4 | 4.000 | 4.85 | 0.313 | 4.00 | 0.290 | 0.453 | 0.313 | 0.145 | 10.72 | 5.36 | 1.64 | 3.56 | 1.78 | 0.94 |
| 5 | 5.000 | 6.63 | .313 | 5.48 | .330 | .503 | .313 | .165 | 23.82 | 9.53 | 2.08 | 7.82 | 3.13 | 1.19 |
| 6 | 5.938 | 8.04 | .250 | 6.64 | .360 | .542 | .313 | .180 | 44.06 | 14.69 | 2.58 | 14.18 | 4.77 | 1.46 |
| 6 | 6.000 | 8.49 | .313 | 7.02 | .360 | .542 | .313 | .180 | 45.19 | 15.06 | 2.54 | 14.65 | 4.88 | 1.44 |
| 6 | 6.125 | 9.40 | .438 | 7.77 | .360 | .542 | .313 | .180 | 47.44 | 15.81 | 2.47 | 15.65 | 5.11 | 1.42 |
| 8 | 7.938 | 11.51 | .313 | 9.52 | .358 | .560 | .313 | .179 | 112.94 | 28.23 | 3.45 | 34.15 | 8.60 | 1.89 |
| 8 | 8.000 | 12.11 | .375 | 10.01 | .358 | .560 | .313 | .179 | 115.58 | 28.90 | 3.40 | 35.01 | 8.75 | 1.87 |
| 8 | 8.125 | 13.32 | .500 | 11.01 | .358 | .560 | .313 | .179 | 120.92 | 30.23 | 3.31 | 36.79 | 9.06 | 1.83 |

* $I$ = moment of inertia; $S$ = section modulus; $r$ = radius of gyration.

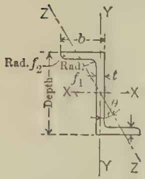


## Table 24.—Dimensions and Properties of Aluminum Tees

(Aluminum Co. of America)

| Flange × Stem, in. | Weight, lb. per ft. | Area, sq. in. | Dimensions, in. | | | | Axis XX * | | | | Axis YY * | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | *t* | *m* | *n* | *f* | *I*, in.$^4$ | *S*, in.$^3$ | *r*, in. | *y*, in. | *I*, in.$^4$ | *S* in.$^3$ | *r*, in. |
| 1 × 1 | 0.32 | 0.27 | 1/8 | 5/32 | 5/32 | 1/8 | 0.02 | 0.03 | 0.29 | 0.29 | 0.01 | 0.02 | 0.21 |
| 1 1/2 × 1 1/4 | 0.45 | .37 | 1/8 | 5/32 | 5/32 | 1/8 | .05 | .05 | .36 | .33 | .04 | .05 | .32 |
| 1 1/2 × 1 1/2 | 0.89 | .74 | 1/4 | 9/32 | 9/32 | 3/16 | .14 | .14 | .44 | .46 | .08 | .10 | .32 |
| 1 1/2 × 2 | 0.88 | .73 | 3/16 | 1/4 | 1/4 | 3/16 | .27 | .20 | .61 | .62 | .06 | .08 | .29 |
| 2 × 2 | 1.29 | 1.07 | 1/4 | 5/16 | 5/16 | 1/4 | .37 | .26 | .59 | .58 | .18 | .18 | .41 |
| 2 × 2 | 1.55 | 1.28 | 5/16 | 3/8 | 3/8 | 1/4 | .43 | .31 | .58 | .61 | .23 | .23 | .42 |
| 2 1/4 × 2 1/4 | 1.47 | 1.21 | 1/4 | 5/16 | 5/16 | 1/4 | .53 | .33 | .66 | .64 | .26 | .23 | .46 |
| 2 1/2 × 1 1/4 | 1.03 | 0.85 | 3/16 | 5/16 | 9/32 | 3/16 | .08 | .09 | .31 | .30 | .28 | .22 | .57 |
| 2 1/2 × 2 1/2 | 2.29 | 1.89 | 3/8 | 7/16 | 7/16 | 1/4 | 1.02 | .58 | .73 | .76 | .53 | .42 | .53 |
| 2 1/2 × 3 | 2.17 | 1.80 | 5/16 | 3/8 | 3/8 | 1/4 | 1.49 | .72 | .91 | .92 | .44 | .35 | .50 |
| 3 × 2 1/2 | 2.19 | 1.81 | 5/16 | 3/8 | 3/8 | 5/16 | 0.94 | .51 | .72 | .68 | .75 | .50 | .65 |
| 3 × 3 | 2.79 | 2.31 | 3/8 | 7/16 | 7/16 | 5/16 | 1.83 | .86 | .89 | .88 | .90 | .60 | .63 |
| 4 × 2 | 2.78 | 2.30 | 3/8 | 7/16 | 7/16 | 1/4 | 0.60 | .40 | .51 | .48 | 2.10 | 1.05 | .96 |
| 4 × 2 1/2 | 3.06 | 2.53 | 3/8 | 7/16 | 7/16 | 3/8 | 1.17 | .62 | .68 | .62 | 2.11 | 1.05 | .91 |
| 4 × 3 | 3.31 | 2.73 | 3/8 | 7/16 | 7/16 | 3/8 | 1.99 | .89 | .85 | .78 | 2.11 | 1.06 | .88 |
| 4 × 4 | 4.92 | 4.07 | 1/2 | 9/16 | 9/16 | 1/2 | 5.74 | 2.02 | 1.19 | 1.17 | 2.82 | 1.41 | .83 |
| 4 × 4 1/2 | 5.24 | 4.33 | 1/2 | 9/16 | 9/16 | 1/2 | 8.04 | 2.55 | 1.36 | 1.35 | 2.83 | 1.41 | .81 |
| 4 × 5 | 5.56 | 4.60 | 1/2 | 9/16 | 9/16 | 1/2 | 10.84 | 3.14 | 1.54 | 1.54 | 2.83 | 1.42 | .79 |
| 5 × 3 | 4.14 | 3.42 | 3/8 | 5/8 | 7/16 | 3/8 | 2.37 | 1.06 | 0.83 | 0.76 | 4.13 | 1.65 | 1.10 |
| 6 1/2 × 6 1/2 | 7.09 | 5.86 | 0.4 | 0.45 | 0.55 | 0.40 | 23.57 | 4.96 | 2.01 | 1.75 | 9.51 | 2.92 | 1.27 |

* *I* = moment of inertia; *S* = section modulus; *r* = radius of gyration; *y* = distance to neutral axis.

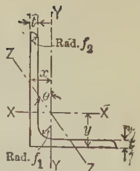


## Table 25.—Dimensions and Properties of Aluminum Zees

(Aluminum Co. of America)

| Depth, in. | | Flange, Width, in. | Weight, lb. per ft. | Area, sq. in. | Dimensions, in. | | | Axis XX * | | | Axis YY * | | | Axis ZZ * | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal | Actual | | | | *t* | $f_1$ | $f_2$ | *I*, in.$^4$ | *S*, in.$^3$ | *r*, in. | *I*, in.$^4$ | *S*, in.$^3$ | *r*, in. | *I*, in.$^4$ | *r*, in. | θ |
| 3 | 3 | 2 11/16 | 2.40 | 1.98 | 1/4 | 5/16 | 1/4 | 2.89 | 1.92 | 1.21 | 2.64 | 1.03 | 1.15 | 0.59 | 0.54 | 43° 24′ |
| 3 | 3 1/16 | 2 3/4 | 3.02 | 2.50 | 5/16 | 5/16 | 1/4 | 3.65 | 2.39 | 1.21 | 3.47 | 1.34 | 1.18 | .76 | .55 | 44° 05′ |
| 3 | 3 | 2 11/16 | 3.48 | 2.87 | 3/8 | 5/16 | 1/4 | 3.86 | 2.57 | 1.16 | 3.76 | 1.50 | 1.14 | .82 | .53 | 44° 31′ |
| 3 | 3 1/16 | 2 3/4 | 4.09 | 3.38 | 7/16 | 5/16 | 1/4 | 4.57 | 2.98 | 1.16 | 4.59 | 1.81 | 1.17 | .99 | .54 | 45° 04′ |
| 3 | 3 | 2 11/16 | 4.48 | 3.70 | 1/2 | 5/16 | 1/4 | 4.60 | 3.06 | 1.11 | 4.71 | 1.93 | 1.13 | 1.03 | .53 | 45° 27′ |
| 3 | 3 1/16 | 2 3/4 | 5.08 | 4.20 | 9/16 | 5/16 | 1/4 | 5.26 | 3.44 | 1.12 | 5.53 | 2.24 | 1.15 | 1.22 | .54 | 45° 55′ |
| 4 | 4 | 3 1/16 | 2.93 | 2.42 | 1/4 | 5/16 | 1/4 | 6.32 | 3.16 | 1.62 | 4.01 | 1.36 | 1.29 | 1.08 | .67 | 36° 47′ |
| 4 | 4 1/16 | 3 1/8 | 3.68 | 3.04 | 5/16 | 5/16 | 1/4 | 7.97 | 3.92 | 1.62 | 5.24 | 1.76 | 1.31 | 1.39 | .68 | 37° 24′ |
| 4 | 4 1/8 | 3 3/16 | 4.44 | 3.67 | 3/8 | 5/16 | 1/4 | 9.66 | 4.68 | 1.62 | 6.54 | 2.18 | 1.33 | 1.72 | .68 | 37° 55′ |
| 4 | 4 | 3 1/16 | 4.92 | 4.06 | 7/16 | 5/16 | 1/4 | 9.68 | 4.84 | 1.54 | 6.53 | 2.30 | 1.27 | 1.74 | .66 | 37° 50′ |
| 4 | 4 1/16 | 3 1/8 | 5.65 | 4.67 | 1/2 | 5/16 | 1/4 | 11.20 | 5.51 | 1.55 | 7.75 | 2.70 | 1.29 | 2.06 | .66 | 38° 16′ |
| 4 | 4 1/8 | 3 3/16 | 6.40 | 5.29 | 9/16 | 5/16 | 1/4 | 12.76 | 6.19 | 1.55 | 9.05 | 3.11 | 1.31 | 2.41 | .68 | 38° 41′ |
| 5 | 5 | 3 1/4 | 4.13 | 3.41 | 5/16 | 5/16 | 1/4 | 13.41 | 5.36 | 1.98 | 5.94 | 1.92 | 1.32 | 1.89 | .74 | 30° 40′ |
| 5 | 5 1/16 | 3 5/16 | 4.98 | 4.12 | 3/8 | 5/16 | 1/4 | 16.23 | 6.41 | 1.99 | 7.40 | 2.37 | 1.34 | 2.33 | .75 | 31° 08′ |
| 5 | 5 1/8 | 3 3/8 | 5.84 | 4.83 | 7/16 | 5/16 | 1/4 | 19.12 | 7.46 | 1.99 | 8.95 | 2.84 | 1.36 | 2.81 | .76 | 31° 32′ |
| 5 | 5 | 3 1/4 | 6.37 | 5.27 | 1/2 | 5/16 | 1/4 | 19.23 | 7.69 | 1.91 | 8.82 | 2.94 | 1.29 | 2.82 | .73 | 31° 09′ |
| 5 | 5 1/16 | 3 5/16 | 7.21 | 5.96 | 9/16 | 5/16 | 1/4 | 21.87 | 8.64 | 1.92 | 10.28 | 3.39 | 1.31 | 3.29 | .74 | 31° 32′ |
| 5 | 5 1/8 | 3 3/8 | 8.05 | 6.66 | 5/8 | 5/16 | 1/4 | 24.56 | 9.59 | 1.92 | 11.82 | 3.86 | 1.33 | 3.79 | .75 | 31° 53′ |
| 6 | 6 | 3 1/2 | 5.58 | 4.61 | 3/8 | 5/16 | 1/4 | 25.40 | 8.47 | 2.35 | 8.83 | 2.67 | 1.38 | 3.08 | .82 | 26° 55′ |
| 6 | 6 1/16 | 3 9/16 | 6.54 | 5.40 | 7/16 | 5/16 | 1/4 | 29.88 | 9.86 | 2.35 | 10.66 | 3.19 | 1.40 | 3.70 | .83 | 27° 17′ |
| 6 | 6 1/8 | 3 5/8 | 7.51 | 6.20 | 1/2 | 5/16 | 1/4 | 34.44 | 11.24 | 2.36 | 12.58 | 3.73 | 1.42 | 4.36 | .84 | 27° 37′ |
| 6 | 6 | 3 1/2 | 8.10 | 6.69 | 9/16 | 5/16 | 1/4 | 34.71 | 11.57 | 2.28 | 12.32 | 3.83 | 1.36 | 4.36 | .81 | 27° 08′ |
| 6 | 6 1/16 | 3 9/16 | 9.05 | 7.48 | 5/8 | 5/16 | 1/4 | 38.93 | 12.84 | 2.28 | 14.15 | 4.35 | 1.38 | 5.01 | .82 | 27° 26′ |
| 6 | 6 1/8 | 3 5/8 | 10.00 | 8.27 | 11/16 | 5/16 | 1/4 | 43.24 | 14.12 | 2.29 | 16.07 | 4.90 | 1.39 | 5.70 | .83 | 27° 44′ |

* *I* = moment of inertia; *S* = section modulus; *r* = radius of gyration.



## Table 26.—Dimensions and Properties of Aluminum Angles

Maximum, Intermediate and Minimum Sizes and Weights
(Aluminum Co. of America)

| Size, in. | Wt., lb. per ft. | Thickness, $t$, in. | Area, sq. in. | Radius, in. | | Axis XX * | | | | Axis YY * | | | | Axis ZZ * | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Corner, $f_1$ | End, $f_2$ | $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | $y$, in. | $I$, in.$^4$ | $S$, in.$^3$ | $r$, in. | $x$, in. | $I$, in.$^4$ | $r$, in. | Angle $\theta$ |
| 3/4 × 3/4 | 0.16 | 3/32 | 0.13 | 1/8 | 3/32 | 0.006 | 0.012 | 0.22 | 0.21 | 0.006 | 0.012 | 0.22 | 0.21 | 0.003 | 0.14 | 45° 0′ |
| “ | .21 | 1/8 | .17 | 1/8 | 3/32 | .008 | .016 | .22 | .23 | .008 | .016 | .22 | .23 | .003 | .14 | 45° 0′ |
| “ | .30 | 3/16 | .25 | 1/8 | 3/32 | .011 | .022 | .21 | .25 | .011 | .022 | .21 | .25 | .005 | .14 | 45° 0′ |
| 1 × 1 | .22 | 3/32 | .18 | 1/8 | 3/32 | .016 | .022 | .30 | .28 | .016 | .022 | .30 | .28 | .007 | .19 | 45° 0′ |
| “ | .41 | 3/16 | .34 | 1/8 | 3/32 | .029 | .042 | .29 | .31 | .029 | .042 | .29 | .31 | .012 | .19 | 45° 0′ |
| “ | .53 | 1/4 | .44 | 1/8 | 3/32 | .036 | .054 | .29 | .34 | .036 | .054 | .29 | .34 | .016 | .19 | 45° 0′ |
| 1 1/4 × 1 1/4 | .28 | 3/32 | .23 | 3/16 | 1/8 | .033 | .036 | .38 | .34 | .033 | .036 | .38 | .34 | .014 | .24 | 45° 0′ |
| “ | .53 | 3/16 | .43 | 3/16 | 1/8 | .059 | .068 | .37 | .37 | .059 | .068 | .37 | .37 | .025 | .24 | 45° 0′ |
| “ | .83 | 5/16 | .68 | 3/16 | 1/8 | .088 | .106 | .36 | .42 | .088 | .106 | .36 | .42 | .040 | .24 | 45° 0′ |
| 1 1/2 × 1 1/4 | .40 | 1/8 | .33 | 3/16 | 1/8 | .070 | .066 | .46 | .44 | .044 | .047 | .37 | .32 | .022 | .26 | 33° 59′ |
| “ | .76 | 1/4 | .63 | 3/16 | 1/8 | .127 | .126 | .45 | .49 | .079 | .090 | .36 | .37 | .041 | .26 | 33° 36′ |
| “ | .92 | 5/16 | .76 | 3/16 | 1/8 | .151 | .153 | .44 | .52 | .093 | .109 | .35 | .39 | .050 | .26 | 33° 15′ |
| 1 1/2 × 1 1/2 | .33 | 3/32 | .28 | 3/16 | 1/8 | .058 | .053 | .46 | .40 | .058 | .053 | .46 | .40 | .024 | .30 | 45° 0′ |
| “ | .83 | 1/4 | .69 | 3/16 | 1/8 | .135 | .130 | .44 | .46 | .135 | .130 | .44 | .46 | .057 | .29 | 45° 0′ |
| “ | 1.19 | 3/8 | .99 | 3/16 | 1/8 | .184 | .185 | .43 | .51 | .184 | .185 | .43 | .51 | .083 | .29 | 45° 0′ |
| 1 3/4 × 1 3/4 | 0.51 | 1/8 | .42 | 3/16 | 1/8 | .121 | .094 | .53 | .47 | .121 | .094 | .53 | .47 | .050 | .34 | 45° 0′ |
| “ | 0.98 | 1/4 | .81 | 3/16 | 1/8 | .223 | .181 | .52 | .52 | .223 | .181 | .52 | .52 | .093 | .34 | 45° 0′ |
| “ | 1.62 | 7/16 | 1.34 | 3/16 | 1/8 | .342 | .295 | .50 | .59 | .342 | .295 | .50 | .59 | .155 | .34 | 45° 0′ |
| 2 × 1 1/2 | 0.51 | 1/8 | 0.42 | 3/16 | 1/8 | .17 | .12 | .63 | .60 | .08 | .07 | .44 | .36 | .04 | .32 | 28° 44′ |
| “ | 0.98 | 1/4 | 0.81 | 3/16 | 1/8 | .31 | .23 | .62 | .66 | .15 | .14 | .43 | .41 | .08 | .32 | 28° 20′ |
| “ | 1.42 | 3/8 | 1.17 | 3/16 | 1/8 | .43 | .33 | .60 | .70 | .20 | .19 | .41 | .45 | .12 | .32 | 27° 37′ |
| 2 × 2 | 0.59 | 1/8 | 0.49 | 1/4 | 1/8 | .18 | .13 | .61 | .53 | .18 | .13 | .61 | .53 | .08 | .40 | 45° 0′ |
| “ | 1.14 | 1/4 | 0.94 | 1/4 | 1/8 | .34 | .24 | .60 | .58 | .34 | .24 | .60 | .58 | .14 | .39 | 45° 0′ |
| “ | 1.89 | 7/16 | 1.57 | 1/4 | 1/8 | .53 | .39 | .58 | .65 | .53 | .39 | .58 | .65 | .23 | .39 | 45° 0′ |
| 2 1/2 × 1 1/2 | 0.87 | 3/16 | 0.72 | 1/4 | 1/8 | .46 | .27 | .79 | .84 | .12 | .11 | .41 | .35 | .08 | .32 | 19° 38′ |
| “ | 1.14 | 1/4 | 0.94 | 1/4 | 1/8 | .59 | .36 | .79 | .86 | .16 | .14 | .41 | .37 | .10 | .32 | 19° 25′ |
| “ | 1.40 | 5/16 | 1.16 | 1/4 | 1/8 | .71 | .44 | .78 | .89 | .19 | .17 | .40 | .39 | .12 | .32 | 19° 06′ |
| 2 1/2 × 2 | 0.67 | 1/8 | 0.55 | 1/4 | 1/8 | .34 | .19 | .79 | .72 | .20 | .13 | .60 | .48 | .10 | .43 | 31° 57′ |
| “ | 1.59 | 5/16 | 1.32 | 1/4 | 1/8 | .78 | .46 | .77 | .80 | .44 | .30 | .58 | .55 | .23 | .42 | 31° 41′ |
| “ | 2.43 | 1/2 | 2.01 | 1/4 | 1/8 | 1.13 | .69 | .75 | .87 | .63 | .46 | .56 | .62 | .35 | .42 | 30° 56′ |
| 2 1/2 × 2 1/2 | 0.75 | 1/8 | 0.62 | 1/4 | 1/8 | 0.37 | .20 | .77 | .65 | .37 | .20 | .77 | .65 | .15 | .50 | 45° 0′ |
| “ | 1.78 | 5/16 | 1.47 | 1/4 | 1/8 | 0.84 | .48 | .76 | .73 | .84 | .48 | .76 | .73 | .35 | .49 | 45° 0′ |
| “ | 2.73 | 1/2 | 2.26 | 1/4 | 1/8 | 1.22 | .72 | .73 | .80 | 1.22 | .72 | .73 | .80 | .53 | .48 | 45° 0′ |
| 3 × 2 | 1.44 | 1/4 | 1.19 | 5/16 | 3/16 | 1.06 | .52 | .94 | .97 | 0.38 | .25 | .56 | .48 | .22 | .43 | 23° 22′ |
| “ | 2.11 | 3/8 | 1.74 | 5/16 | 3/16 | 1.51 | .76 | .93 | 1.03 | .53 | .36 | .55 | .53 | .31 | .42 | 23° 0′ |
| “ | 2.73 | 1/2 | 2.26 | 5/16 | 3/16 | 1.90 | .99 | .92 | 1.07 | .66 | .46 | .54 | .58 | .40 | .42 | 22° 25′ |
| 3 × 2 1/2 | 1.58 | 1/4 | 1.31 | 5/16 | 1/4 | 1.12 | .53 | .92 | 0.89 | .70 | .38 | .73 | .64 | .35 | .52 | 34° 03′ |
| “ | 2.32 | 3/8 | 1.92 | 5/16 | 1/4 | 1.60 | .78 | .91 | .94 | 1.00 | .55 | .72 | .69 | .51 | .51 | 33° 54′ |
| “ | 3.02 | 1/2 | 2.49 | 5/16 | 1/4 | 2.03 | 1.01 | .90 | .99 | 1.26 | .72 | .71 | .74 | .65 | .51 | 33° 37′ |
| 3 × 3 | 1.33 | 3/16 | 1.10 | 5/16 | 1/4 | 0.93 | 0.42 | .92 | .80 | 0.93 | .42 | .92 | .80 | .38 | .59 | 45° 0′ |
| “ | 2.55 | 3/8 | 2.10 | 5/16 | 1/4 | 1.70 | 0.80 | .90 | .87 | 1.70 | .80 | .90 | .87 | .70 | .58 | 45° 0′ |
| “ | 4.06 | 5/8 | 3.35 | 5/16 | 1/4 | 2.57 | 1.26 | .88 | .97 | 2.57 | 1.26 | .88 | .97 | 1.12 | .58 | 45° 0′ |
| 3 1/2 × 3 | 1.89 | 1/4 | 1.57 | 3/8 | 1/4 | 1.84 | 0.74 | 1.08 | 1.01 | 1.28 | 0.57 | .90 | .76 | 0.63 | .63 | 36° 13′ |
| “ | 2.78 | 3/8 | 2.30 | 3/8 | 1/4 | 2.65 | 1.09 | 1.07 | 1.06 | 1.79 | 0.82 | .88 | .82 | 0.87 | .62 | 35° 37′ |
| “ | 3.63 | 1/2 | 3.00 | 3/8 | 1/4 | 3.38 | 1.42 | 1.06 | 1.11 | 2.27 | 1.06 | .87 | .86 | 1.13 | .61 | 35° 26′ |
| 3 1/2 × 3 1/2 | 2.05 | 1/4 | 1.69 | 3/8 | 1/4 | 1.93 | 0.76 | 1.07 | 0.94 | 1.93 | 0.76 | 1.07 | .94 | 0.80 | .69 | 45° 0′ |
| “ | 3.48 | 7/16 | 2.87 | 3/8 | 1/4 | 3.18 | 1.28 | 1.05 | 1.02 | 3.18 | 1.28 | 1.05 | 1.02 | 1.32 | .68 | 45° 0′ |
| “ | 4.83 | 5/8 | 3.99 | 3/8 | 1/4 | 4.26 | 1.77 | 1.03 | 1.09 | 4.26 | 1.77 | 1.03 | 1.09 | 1.81 | .67 | 45° 0′ |
| 4 × 3 | 2.05 | 1/4 | 1.69 | 3/8 | 1/4 | 2.68 | 0.96 | 1.26 | 1.21 | 1.29 | 0.56 | 0.87 | 0.72 | 0.70 | .64 | 28° 42′ |
| “ | 3.48 | 7/16 | 2.87 | 3/8 | 1/4 | 4.43 | 1.63 | 1.24 | 1.29 | 2.12 | 0.96 | 0.86 | .79 | 1.15 | .63 | 28° 28′ |
| “ | 4.83 | 5/8 | 3.99 | 3/8 | 1/4 | 5.95 | 2.25 | 1.22 | 1.36 | 2.82 | 1.32 | 0.84 | .86 | 1.58 | .63 | 28° 0′ |
| 4 × 4 | 2.35 | 1/4 | 1.94 | 3/8 | 1/4 | 2.94 | 1.00 | 1.23 | 1.07 | 2.94 | 1.00 | 1.23 | 1.07 | 1.21 | .79 | 45° 0′ |
| “ | 4.54 | 1/2 | 3.75 | 3/8 | 1/4 | 5.46 | 1.93 | 1.21 | 1.17 | 5.46 | 1.93 | 1.21 | 1.17 | 2.26 | .78 | 45° 0′ |
| “ | 6.58 | 3/4 | 5.44 | 3/8 | 1/4 | 7.57 | 2.77 | 1.18 | 1.26 | 7.57 | 2.77 | 1.18 | 1.26 | 3.25 | .77 | 45° 0′ |
| 5 × 3 | 2.89 | 5/16 | 2.39 | 3/8 | 5/16 | 6.04 | 1.80 | 1.59 | 1.65 | 1.64 | 0.70 | 0.83 | 0.66 | 0.99 | .64 | 19° 44′ |
| “ | 3.99 | 7/16 | 3.30 | 3/8 | 5/16 | 8.22 | 2.49 | 1.58 | 1.70 | 2.21 | 0.97 | 0.82 | 0.71 | 1.34 | .64 | 19° 34′ |
| “ | 5.56 | 5/8 | 4.60 | 3/8 | 5/16 | 11.17 | 3.47 | 1.56 | 1.78 | 2.97 | 1.34 | 0.80 | 0.79 | 1.85 | .63 | 19° 06′ |
| 5 × 4 | 3.91 | 3/8 | 3.23 | 7/16 | 5/16 | 7.91 | 2.26 | 1.56 | 1.50 | 4.49 | 1.50 | 1.18 | 1.01 | 2.30 | .84 | 31° 58′ |
| “ | 5.14 | 1/2 | 4.25 | 7/16 | 5/16 | 10.23 | 2.97 | 1.55 | 1.55 | 5.78 | 1.97 | 1.17 | 1.06 | 2.99 | .84 | 31° 52′ |
| “ | 6.33 | 5/8 | 5.23 | 7/16 | 5/16 | 12.38 | 3.64 | 1.54 | 1.60 | 6.97 | 2.41 | 1.15 | 1.11 | 3.64 | .83 | 31° 44′ |
| 5 × 5 | 4.36 | 3/8 | 3.60 | 1/2 | 3/8 | 8.37 | 2.30 | 1.52 | 1.36 | 8.37 | 2.30 | 1.52 | 1.36 | 3.44 | .98 | 45° 0′ |
| “ | 6.42 | 9/16 | 5.30 | 1/2 | 3/8 | 12.08 | 3.39 | 1.51 | 1.43 | 12.08 | 3.39 | 1.51 | 1.43 | 4.97 | .97 | 45° 0′ |
| “ | 8.39 | 3/4 | 6.93 | 1/2 | 3/8 | 15.39 | 4.41 | 1.49 | 1.51 | 15.39 | 4.41 | 1.49 | 1.51 | 6.44 | .96 | 45° 0′ |
| 6 × 4 | 4.36 | 3/8 | 3.60 | 1/2 | 3/8 | 13.02 | 3.17 | 1.90 | 1.90 | 4.63 | 1.50 | 1.13 | 0.91 | 2.67 | .86 | 23° 33′ |
| “ | 6.42 | 9/16 | 5.30 | 1/2 | 3/8 | 18.82 | 4.69 | 1.88 | 1.98 | 6.65 | 2.21 | 1.12 | 0.99 | 3.86 | .85 | 23° 22′ |
| “ | 8.39 | 3/4 | 6.93 | 1/2 | 3/8 | 24.08 | 6.11 | 1.86 | 2.06 | 8.43 | 2.87 | 1.10 | 1.07 | 4.98 | .85 | 23° 02′ |
| 6 × 6 | 5.27 | 3/8 | 4.35 | 1/2 | 3/8 | 14.85 | 3.38 | 1.85 | 1.60 | 14.85 | 3.38 | 1.85 | 1.60 | 6.07 | 1.18 | 45° 0′ |
| “ | 7.78 | 9/16 | 6.43 | 1/2 | 3/8 | 21.54 | 4.99 | 1.83 | 1.68 | 21.54 | 4.99 | 1.83 | 1.68 | 8.82 | 1.17 | 45° 0′ |
| “ | 10.20 | 3/4 | 8.43 | 1/2 | 3/8 | 27.64 | 6.52 | 1.81 | 1.76 | 27.64 | 6.52 | 1.81 | 1.76 | 11.43 | 1.16 | 45° 0′ |

* $I$ = moment of inertia; $S$ = section modulus; $r$ = radius of gyration; $x$, $y$ = distance from extreme fiber to neutral axis.

# STEEL, COPPER AND OTHER WIRES

Table 1 was calculated on the basis of 483.84 lb. per cu. ft. for steel wire. Iron wire is a trifle lighter.

The breaking strengths are calculated for 100,000 lb. per sq. in. throughout simply for convenience, so that the breaking strength of wires of any strength per square inch may be quickly determined by multiplying the values given in the tables by the ratio between strength per square inch and 100,000; thus a No. 15 wire with a strength per square inch of 150,000 lb. has a breaking strength of:

$$407 \times 150{,}000/100{,}000 = 610.5 \text{ lb. per sq. in.}$$

## 1. PLOW STEEL WIRE

The term "Plow" was given in England to steel wire of high quality which was used in connection with the construction of wire ropes used for plowing purposes. The term has become inseparably linked with a high-grade wire rope, lists of which will be found under Wire Rope, p. 6-34. It should not be confused with the quality of plate steel used for mold-boards of plows, for which a very ordinary grade is good enough.

The practice of various manufacturers differs somewhat, but ordinarily plow steel is a high-grade open-hearth steel, ranging in carbon from 0.50 to 0.95, depending on the diameter of finished wire. The lower carbon is used in the smaller, and the higher in the larger wire. Phosphorus and sulphur are 0.050 or under; manganese and silicon in varying quantities.

It has been the custom in the industry to classify the different grades of wire according to tensile strength in pounds per square inch throughout the general commercial range of wire sizes. Plow steel falls within the limits of 224,000 to 246,000 lb. per sq. in., with the lower strengths applying to the coarser sizes and the higher strengths to the finer sizes. This wire is used largely for wire ropes, springs, and similar purposes.

## 2. CAST STEEL WIRE

The term cast steel applied to a steel used in rope manufacture indicates, in general, a composition as follows: Carbon, 0.40 to 0.70; phosphorus and sulphur, 0.040 and under; manganese and silicon, in varying quantities. Cast steel wire invariably is made in the open-hearth furnace and is of lower tensile strength than plow steel. It has been the custom in the industry to classify the different grades of wire according to the tensile strength in pounds per square inch throughout the general commercial range of wire sizes. Cast steel falls within the limits of 179,200 to 201,600 lb. per sq. in., with the lower strengths applying to the coarser sizes and the higher strengths to the finer sizes. This wire is used largely for wire rope, springs, and similar purposes.

## 3. WIRE FOR TELEGRAPH AND TELEPHONE LINES

SIZES OF GALVANIZED IRON WIRE USED IN TELEGRAPH AND TELEPHONE LINES.—No. 4. Has not been much used until recently; is now used on important lines where the multiplex systems are applied.

No. 6. Used for important circuits between cities.

No. 8. Medium size for circuits of 400 miles or less.

No. 9. For similar locations to No. 8, but on somewhat shorter circuits; until recently was the size most largely used in this country.

Nos. 10, 11. For shorter circuits, railway telegraphs, private lines, police and fire-alarm lines, etc.

No. 12. For telephone lines, police and fire-alarm lines, etc.

No. 14. For telephone lines and short private lines; steel wire is used most generally in these sizes.

The coating of telegraph wire with zinc as a protection against oxidation is now generally admitted to be the most efficacious method.

The grades of line wire are generally known to the trade as "Extra Best Best" (E. B. B.), "Best Best" (B. B.), and "Steel."

**JOINTS IN TELEGRAPH WIRES.**—The fewer the joints in a line the better. All joints should be carefully made and well soldered over, for a bad joint may cause as much resistance to the electric current as several miles of wire.

**Table 1.—Properties of Steel Wire**

(John A. Roebling's Sons. Co., Trenton, N. J.)

| Steel Wire, Gage No. | Diam., in. | Area, sq. in. | Breaking Strain at 100,000 lb. per sq. in. | Weight in Pounds per 1000 ft. | Weight in Pounds per mile | Feet in 2000 lb. |
|---|---|---|---|---|---|---|
| 000 000 | 0.460 | 0.166190 | 16,619 | 558.4 | 2948 | 3,582 |
| 00 000 | .430 | .145220 | 14,522 | 487.9 | 2576 | 4,099 |
| 0 000 | .393 | .121304 | 12,130 | 407.6 | 2152 | 4,907 |
| 000 | .362 | .102922 | 10,292 | 345.8 | 1826 | 5,783 |
| 00 | .331 | .086049 | 8,605 | 289.1 | 1527 | 6,917 |
| 0 | .307 | .074023 | 7,402 | 248.7 | 1313 | 8,041 |
| 1 | .283 | .062902 | 6,290 | 211.4 | 1116 | 9,463 |
| 2 | .263 | .054325 | 5,433 | 182.5 | 964 | 10,957 |
| 3 | .244 | .046760 | 4,676 | 157.1 | 830 | 12,730 |
| 4 | .225 | .039761 | 3,976 | 133.6 | 705 | 14,970 |
| 5 | .207 | .033654 | 3,365 | 113.1 | 597 | 17,687 |
| 6 | .192 | .028953 | 2,895 | 97.3 | 514 | 20,559 |
| 7 | .177 | .024606 | 2,461 | 82.7 | 437 | 24,191 |
| 8 | .162 | .020612 | 2,061 | 69.3 | 366 | 28,878 |
| 9 | .148 | .017203 | 1,720 | 57.8 | 305 | 34,600 |
| 10 | .135 | .014314 | 1,431 | 48.1 | 254 | 41,584 |
| 11 | .120 | .011310 | 1,131 | 38.0 | 201 | 52,631 |
| 12 | .105 | .008659 | 866 | 29.1 | 154 | 68,752 |
| 13 | .092 | .006648 | 665 | 22.3 | 118 | 89,525 |
| 14 | .080 | .005027 | 503 | 16.9 | 89.2 | 118,413 |
| 15 | .072 | .004072 | 407 | 13.7 | 72.2 | 146,198 |
| 16 | .063 | .003117 | 312 | 10.5 | 55.3 | 191,022 |
| 17 | .054 | .002290 | 229 | 7.70 | 40.6 | 259,909 |
| 18 | .047 | .001735 | 174 | 5.83 | 30.8 | 343,112 |
| 19 | .041 | .001320 | 132 | 4.44 | 23.4 | 450,856 |
| 20 | .035 | .000962 | 96 | 3.23 | 17.1 | 618,620 |
| 21 | .032 | .000804 | 80 | 2.70 | 14.3 | 740,193 |
| 22 | .028 | .000616 | 62 | 2.07 | 10.9 | 966,651 |
| 23 | .025 | .000491 | 49 | 1.65 | 8.71 | ........ |
| 24 | .023 | .000415 | 42 | 1.40 | 7.37 | ........ |
| 25 | .020 | .000314 | 31 | 1.06 | 5.58 | ........ |
| 26 | .018 | .000254 | 25 | 0.855 | 4.51 | ........ |
| 27 | .017 | .000227 | 23 | .763 | 4.03 | ........ |
| 28 | .016 | .0002011 | 20 | .676 | 3.57 | ........ |
| 29 | .015 | .0001767 | 18 | .594 | 3.14 | ........ |
| 30 | .014 | .0001539 | 15 | .517 | 2.73 | ........ |
| 31 | .0135 | .0001431 | 14 | .481 | 2.54 | ........ |
| 32 | .013 | .0001327 | 13 | .446 | 2.36 | ........ |
| 33 | .011 | .0000950 | 9.5 | .319 | 1.69 | ........ |
| 34 | .010 | .0000785 | 7.9 | .264 | 1.39 | ........ |
| 35 | .0095 | .0000709 | 7.1 | .238 | 1.26 | ........ |
| 36 | .0090 | .0000636 | 6.4 | .214 | 1.13 | ........ |

**Table 2.—Strength of Piano Wire**

(American Steel & Wire Co., Cleveland)

| Size, Music Wire Gage | Equiv. Diam., in. | Ultimate Tensile Strength, Pounds | Ultimate Tensile Strength, lb. per sq. in. | Size, Music Wire Gage | Equiv. Diam., in. | Ultimate Tensile Strength, Pounds | Ultimate Tensile Strength, lb. per sq. in. |
|---|---|---|---|---|---|---|---|
| 12 | 0.029 | 237 | 360,000 | 20 | 0.045 | 509 | 320,000 |
| 13 | .031 | 268 | 355,000 | 21 | .047 | 547 | 315,000 |
| 14 | .033 | 300 | 350,000 | 22 | .049 | 585 | 310,000 |
| 15 | .035 | 322 | 345,000 | 23 | .051 | 623 | 305,000 |
| 16 | .037 | 366 | 340,000 | 24 | .055 | 713 | 300,000 |
| 17 | .039 | 400 | 335,000 | 25 | .059 | 806 | 295,000 |
| 18 | .041 | 436 | 330,000 | 26 | .063 | 904 | 290,000 |
| 19 | .043 | 472 | 325,000 | | | | |

Analysis of steel should show phosphorus under 0.045 and sulphur under 0.045.

Table 3.—Sizes and Weights of Telegraph and Telephone Wires

| Size, B.W.G. | Diam., in. | Weight, lb. per 1000 ft. | Weight, lb. per mile | Approximate Breaking Strength, lb. | | | Average Resistance per Mile at 68° F., International Ohms | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | E.B.B. | B.B. | Steel | E.B.B. | B.B. | Steel |
| 4 | 0.238 | 153 | 811 | 2028 | 2271 | 2433 | 5.98 | 7.15 | 8.32 |
| 6 | .203 | 112 | 590 | 1475 | 1652 | 1770 | 8.22 | 9.83 | 11.44 |
| 8 | .165 | 74 | 390 | 975 | 1092 | 1170 | 12.43 | 14.87 | 17.31 |
| 9 | .148 | 60 | 314 | 785 | 879 | 942 | 15.44 | 18.47 | 21.50 |
| 10 | .134 | 49 | 258 | 645 | 722 | 774 | 18.79 | 22.48 | 26.16 |
| 11 | .120 | 39 | 206 | 515 | 577 | 618 | 23.54 | 28.16 | 32.77 |
| 12 | .109 | 32 | 170 | 425 | 476 | 510 | 28.52 | 34.12 | 39.71 |
| 14 | .083 | 19 | 99 | 247 | 277 | 297 | 48.98 | 58.59 | 68.18 |

**SPECIFICATIONS.**—(Abstracted from the standard specifications of the American Steel and Wire Co., and John A. Roebling's Sons Co.)—The scope of the specifications cover the usual requirements of the grades E. B. B., B. B. and Steel.

*Finish.*—The rods, before drawing must not contain any joint, weld or splice. The galvanizing shall consist of a continuous coating of pure zinc, of uniform thickness, so applied that it adheres firmly and presents a smooth surface.

Electrical and Physical Requirements

| Grade | Permissible Variation in Mile-ohm-pound Constant See Note 1 | Average Tensile Strength, lb. per sq. in. | Constant for Minimum Actual Breaking Weight See Note 2 |
|---|---|---|---|
| E.B.B. ........ | 4700–5000 | 45,500 | 2.5 |
| B.B. .......... | 5600–6000 | 51,000 | 2.8 |
| Steel .......... | 6500–7000 | 54,500 | 3.0 |

**Note 1.**—Mile-ohm-pound constant = actual resistance per mile at 68° F. × actual weight, lb. per mile.

**Note 2.**—Actual breaking weight = weight per mile × constant in table.

**Variation in Diameter.**—Permissible variations in diameter are as follows: Sizes, 4 to 11, inclusive, B. W. G., variation 0.004 in. either way; sizes 12 to 14 inclusive, B. W. G., variation 0.003 in. either way.

**Galvanizing Test.**—Samples, after cleaning by gasoline and cotton waste, rinsing in clean water and wiped dry with clean cotton waste, shall be immersed in the solution to a depth of 4 in. Samples shall be so separated that the solution may act uniformly on all immersed portions. The solution shall be a solution of commercial copper sulphate, specific gravity 1.186 at 65° F., neutralized by the addition of an excess of chemically pure cupric oxide (CuO) which shall collect in the bottom of the containing vessel. Solution is to be filtered before use. Not less than 4 oz. of fresh solution is to be used for each test of 7 wires, and it must be maintained at a temperature between 62° F. and 68° F. during the test. Four immersions of 1 minute each shall be given the samples, which are to be washed in clear water of between 62° F. and 68° F. between each immersion and then wiped dry with cotton waste. On sizes Nos. 13 and 14 B. W. G., the fourth immersion shall be of $1/2$ minute duration. A bright copper deposit after the fourth immersion indicates that the wire represented by the samples is faulty. Copper deposited on zinc, or within 1 inch of the end of a sample, shall not be considered cause for rejection.

**Coils and Splices.**—All wire shall be furnished in coils, whose usual inside diameter is 21 $1/2$ in. The approximate length of wire in each coil shall be:

Nos. 4 and 5, $1/4$ mile; Nos. 6 and 7, $1/3$ mile; No. 8 and smaller, $1/2$ mile. One-half the coils of No. 10 wire, and smaller, and each coil of No. 10 and larger may contain one splice. The splice may be made by electrical welding or by the ordinary twist joint thoroughly galvanized. In handling this wire the greatest care should be observed to avoid kinks, bends, scratches or cuts. Joints should be made only with McIntire connectors.

**SELECTION OF WIRE.**—On account of its conductivity being about five times that of E. B. B. steel wire, and its breaking strength over three times its weight per mile, copper may be used of which the section is smaller and the weight less than an equivalent steel wire, allowing a greater number of wires to be strung on the poles. Besides this advantage, the reduction of section materially decreases the electrostatic capacity, while its non-magnetic character lessens the self-induction of the line, both of which features tend to increase the possible speed of signaling in telegraphing, and to give greater clearness of enunciation over telephone lines, especially those of great length.

Steel is not economical as a transmission line material, except where a copper wire considerably larger than necessary is required to support, theoretically, an ice load. The product of the mile-ohm and the price per pound of wires gives a measure of their relative cost, taking into consideration their efficiency as conductors.

Telephone wires may be either hard-drawn copper or galvanized steel. The copper runs from 165 to 64 mils, and the steel from 283 to 83 mils. The ground wire carried on top of the poles is usually a 3/8-in. galvanized Siemens-Martin strand.

## 4. NON-FERROUS WIRE

**SILICON-BRONZE WIRE** has a conductivity of from 40 to 98% that of copper wire, and about 4 times that of iron wire. Its tensile strength is nearly that of steel or 28 to 55 tons per sq. in. of section. The conductivity decreases as the tensile strength increases. Wire whose conductivity equals 95% of that of pure copper gives a tensile strength of 28 tons per sq. in., but when its conductivity is 34% that of pure copper, its strength is 50 tons per sq. in. It is largely used for telegraph wires and has great resistance to oxidation.

**Table 4.—Properties of Bronze Wire, 68° F.**

(John A. Roebling's Sons Co., Trenton, N. J.)

| Size, A. W. G. | Diameter, Inches | Pounds per | | Ohms per | | Breaking Strength, lb. |
|---|---|---|---|---|---|---|
| | | 1000 Feet | Mile | 1000 Feet | Mile | |
| 0000 | 0.460 | 637 | 3365 | 0.123 | 0.647 | 11,467 |
| 000 | .410 | 505 | 2669 | .155 | .816 | 9,120 |
| 00 | .365 | 407 | 2116 | .195 | 1.028 | 7,381 |
| 0 | .325 | 318 | 1678 | .246 | 1.298 | 6,285 |
| 1 | .289 | 252 | 1331 | .310 | 1.635 | 5,127 |
| 2 | .258 | 200 | 1055 | .391 | 2.063 | 4,170 |
| 3 | .229 | 159 | 837 | .493 | 2.600 | 3,348 |
| 4 | .204 | 126 | 664 | .621 | 3.280 | 2,662 |
| 5 | .182 | 100 | 526 | .783 | 4.135 | 2,114 |
| 6 | .162 | 79 | 417 | .988 | 5.215 | 1,682 |
| 7 | .144 | 63 | 331 | 1.245 | 6.575 | 1,346 |
| 8 | .128 | 50 | 263 | 1.570 | 8.293 | 1,072 |
| 9 | .114 | 39 | 208 | 1.980 | 10.455 | 853 |
| 10 | .102 | 31 | 165 | 2.497 | 13.185 | 683 |
| 11 | .091 | 25 | 131 | 3.149 | 16.625 | 543 |
| 12 | .081 | 20 | 104 | 3.971 | 20.965 | 436 |
| 13 | .072 | 15.6 | 82 | 5.007 | 26.425 | 350 |
| 14 | .065 | 12.4 | 64 | 6.313 | 33.325 | 281 |
| 15 | .057 | 9.8 | 52 | 7.961 | 42.025 | 225 |
| 16 | .051 | 7.8 | 41 | 10.040 | 53.000 | 181 |
| 17 | .045 | 6.2 | 33 | 12.660 | 66.850 | 145 |
| 18 | .040 | 4.9 | 26 | 15.963 | 84.275 | 116 |

**Table 5.—Weight of Bare and Insulated Copper Wire, Pounds**

(John A. Roebling's Sons Co., Trenton, N. J.)

| Size, B. & S. Gage | Weight, Pounds, per 1000 Feet, Solid | | | | | Weight, Pounds, per Mile, Solid | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Bare | Weatherproof | | Slow Burning Weather-proof | Slow Burn-ing | Bare | Weatherproof | | Slow Burning Weather-proof | Slow Burn-ing |
| | | Double Braid | Triple Braid | | | | Double Braid | Triple Braid | | |
| 0000 | 641 | 723 | 767 | 862 | 925 | 3384 | 3817 | 4050 | 4550 | 4890 |
| 000 | 508 | 587 | 629 | 710 | 760 | 2687 | 3098 | 3320 | 3750 | 4020 |
| 00 | 403 | 467 | 502 | 562 | 600 | 2127 | 2467 | 2650 | 2970 | 3170 |
| 0 | 320 | 377 | 407 | 462 | 495 | 1689 | 1989 | 2150 | 2440 | 2610 |
| 1 | 253 | 294 | 316 | 340 | 365 | 1335 | 1553 | 1670 | 1800 | 1930 |
| 2 | 201 | 239 | 260 | 280 | 320 | 1066 | 1264 | 1370 | 1480 | 1690 |
| 3 | 159 | 185 | 199 | 230 | 270 | 840 | 977 | 1050 | 1220 | 1425 |
| 4 | 126 | 151 | 164 | 190 | 220 | 665 | 795 | 865 | 1000 | 1160 |
| 5 | 100 | 122 | 135 | 155 | 190 | 528 | 646 | 710 | 820 | 1000 |
| 6 | 80 | 100 | 112 | 127 | 160 | 417 | 529 | 590 | 670 | 845 |
| 8 | 50 | 66 | 75 | 85 | 100 | 264 | 349 | 395 | 450 | 530 |
| 9 | 40 | 54 | 62 | .......... | ...... | 206 | 283 | 325 | .......... | ...... |
| 10 | 31 | 46 | 53 | 60 | 80 | 169 | 241 | 280 | 315 | 420 |
| 12 | 20 | 30 | 35 | 42 | 55 | 106 | 158 | 185 | 220 | 290 |
| 14 | 12.4 | 20 | 25 | 30 | 40 | 66 | 107 | 130 | 160 | 210 |
| 16 | 7.8 | 16 | 20 | 24 | 30 | 42 | 83 | 90 | 130 | 160 |
| 18 | 4.9 | 12 | 16 | 19 | 24 | 25 | 64 | 69 | 100 | 130 |
| 20 | 3.1 | 9 | 12 | .......... | ...... | 16 | 48 | 63 | .......... | ...... |

### Table 6.—Properties of Calsun Bronze Wire and Everdur Wire

| Amer. Wire Gage (B. & S.) | Diam., in. | Area, sq. in. | Weight | | Calsun Bronze Wire * | | Everdur Wire | |
|---|---|---|---|---|---|---|---|---|
| | | | Lb. per 1000 ft. | Feet per lb. | Minimum Tensile Strength, lb. per sq. in. | Nominal Breaking Strength, lb.† | Minimum Tensile Strength, lb. per sq. in. | Nominal Breaking Strength, lb.† |
| 1 | 0.2893 | 0.06573 | 243.0 | 4.116 | 109,500 | 7198 | 120,000 | 7888 |
| 2 | .2576 | .05212 | 192.6 | 5.191 | 114,500 | 5967 | 126,600 | 6598 |
| 3 | .2294 | .04133 | 152.8 | 6.546 | 118,500 | 4898 | 130,800 | 5406 |
| 4 | .2043 | .03278 | 121.2 | 8.254 | 121,300 | 3976 | 133,500 | 4376 |
| 5 | .1819 | .02599 | 96.05 | 10.41 | 123,300 | 3204 | 135,500 | 3521 |
| 6 | .1620 | .02061 | 76.18 | 13.13 | 125,000 | 2577 | 137,200 | 2828 |
| 7 | .1443 | .01635 | 60.44 | 16.54 | 126,500 | 2069 | 138,700 | 2268 |
| 8 | .1285 | .01297 | 47.93 | 20.86 | 127,900 | 1659 | 140,000 | 1816 |
| 9 | .1144 | .01028 | 37.99 | 26.32 | 129,200 | 1328 | 141,000 | 1449 |
| 10 | .1019 | .008155 | 30.14 | 33.18 | 130,300 | 1063 | 141,800 | 1156 |
| 11 | .0907 | .006461 | 23.88 | 41.88 | 131,200 | 848 | 142,400 | 920 |
| 12 | .0808 | .005128 | 18.95 | 52.77 | 132,000 | 677 | 143,000 | 733 |
| 13 | .0720 | .004072 | 15.05 | 66.45 | 132,600 | 540 | 143,400 | 584 |
| 14 | .0641 | .003227 | 11.93 | 83.84 | 133,200 | 430 | 143,800 | 464 |
| 15 | .0571 | .002561 | 9.464 | 105.7 | 133,600 | 342 | 144,000 | 369 |
| 16 | .0508 | .002027 | 7.491 | 133.5 | 134,000 | 272 | 144,300 | 292 |
| 17 | .0453 | .001612 | 5.957 | 167.9 | 134,250 | 216 | 144,600 | 233 |
| 18 | .0403 | .001276 | 4.714 | 212.1 | 134,500 | 172 | 144,800 | 185 |
| 19 | .0359 | .001012 | 3.741 | 267.3 | 134,750 | 136 | 144,900 | 147 |
| 20 | .0320 | .0008042 | 2.973 | 336.4 | 135,000 | 109 | 145,000 | 117 |

* Cu, 95.5, Al, 2.5, Sn, 2.00.
† Calculated directly from minimum tensile strength and nominal diameter.

### Table 7.—Properties of Hard Drawn Copper Wire at 68° F.

(American Brass Co., Waterbury, Conn.)

| Gage, A. W. G., or B. & S. | Diam., in. | Area, | | Weight, lb. per 1000 ft. | Resistance, Ohms per 1000 ft. | Elastic Limit | | Breaking Strength, Min., per Std. Size, Single Wire, lb. | Tensile Strength, Min., 1000 lb. per sq. in. | Elongation, Min., percent |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Circular mils. | Square Inches | | | Single Wire, lb. | 1000 lb. per sq. in. | | | |
| 0000 | 0.4600 | 211,600 | 0.1662 | 640.5 | 0.05027 | 4479. | 26.95 | 8143. | 49.00 | 3.75 * |
| 000 | .4096 | 167,800 | .1318 | 507.9 | .06339 | 3697. | 28.05 | 6722. | 51.00 | 3.25 * |
| 00 | .3648 | 133,100 | .1045 | 402.8 | .07993 | 3035. | 29.04 | 5519. | 52.80 | 2.80 * |
| 0 | .3249 | 105,500 | .08289 | 319.5 | .1008 | 2485. | 29.98 | 4517. | 54.50 | 2.40 * |
| 1 | .2893 | 83,690 | .06573 | 253.3 | .1271 | 2213. | 33.66 | 3688. | 56.10 | 2.17 * |
| 2 | .2576 | 66,370 | .05213 | 200.9 | .1603 | 1802. | 34.56 | 3003. | 57.60 | 1.98 * |
| 3 | .2294 | 52,640 | .04134 | 159.3 | .2021 | 1463. | 35.40 | 2439. | 59.00 | 1.79 * |
| 4 | .2043 | 41,740 | .03278 | 126.4 | .2548 | 1182. | 36.06 | 1970. | 60.10 | 1.24 † |
| 5 | .1819 | 33,100 | .02600 | 100.2 | .3213 | 954.7 | 36.72 | 1591. | 61.20 | 1.18 † |
| T | .1650 | 27,230 | .02138 | 82.41 | .3907 | 795.4 | 37.20 | 1326. | 62.00 | 1.14 † |
| 6 | .1620 | 26,250 | .02062 | 79.46 | .4052 | 768.2 | 37.26 | 1280. | 62.10 | 1.14 † |
| 7 | .1443 | 20,820 | .01635 | 63.02 | .5110 | 618.1 | 37.80 | 1030. | 63.00 | 1.09 † |
| T | .1340 | 17,960 | .01410 | 54.35 | .5924 | 536.5 | 38.04 | 894.1 | 63.40 | 1.07 † |
| 8 | .1285 | 16,510 | .01297 | 49.98 | .6443 | 495.6 | 38.22 | 826.0 | 63.70 | 1.06 † |
| 9 | .1144 | 13,090 | .01028 | 39.63 | .8124 | 396.7 | 38.58 | 661.2 | 64.30 | 1.02 † |
| T | .1040 | 10,820 | .008495 | 32.74 | .9835 | 330.3 | 38.88 | 550.5 | 64.80 | 1.00 † |
| 10 | .1019 | 10,380 | .008155 | 31.43 | 1.024 | 317.5 | 38.94 | 529.2 | 64.90 | 1.00 † |
| T | .0920 | 8,464 | .006648 | 25.62 | 1.257 | 260.9 | 39.24 | 434.8 | 65.40 | 0.97 † |
| 11 | .0907 | 8,234 | .006467 | 24.92 | 1.292 | 253.8 | 39.24 | 422.9 | 65.40 | .97 † |
| 12 | .0808 | 6,530 | .005129 | 19.77 | 1.629 | 202.2 | 39.42 | 336.9 | 65.70 | .95 † |
| T | .0800 | 6,400 | .005027 | 19.37 | 1.662 | 198.1 | 39.42 | 330.2 | 65.70 | .94 † |
| 13 | .0720 | 5,178 | .004067 | 15.68 | 2.054 | 160.8 | 39.54 | 268.0 | 65.90 | .92 † |
| T | .0650 | 4,225 | .003318 | 12.79 | 2.518 | 131.8 | 39.72 | 219.7 | 66.20 | .91 † |
| 14 | .0641 | 4,107 | .003225 | 12.43 | 2.590 | 128.1 | 39.72 | 213.5 | 66.20 | .90 † |
| 15 | .0571 | 3,257 | .002558 | 9.858 | 3.266 | 101.5 | 39.84 | 169.8 | 66.40 | .89 † |
| 16 | .0508 | 2,583 | .002028 | 7.818 | 4.119 | 81.06 | 39.96 | 135.1 | 66.60 | .87 † |
| 17 | .0453 | 2,048 | .001609 | 6.200 | 5.193 | 64.47 | 40.08 | 107.5 | 66.80 | .86 † |
| 18 | .0403 | 1,624 | .001276 | 4.917 | 6.549 | 51.28 | 40.20 | 85.47 | 67.00 | .85 † |
| 19 | .0359 | 1,288 | .001012 | 3.899 | 8.258 | 40.79 | 40.32 | 67.99 | 67.20 | .84 † |
| 20 | .0320 | 1,022 | .0008023 | 3.092 | 10.41 | 32.45 | 40.44 | 54.08 | 67.40 | .83 † |

* In 10 in.; † In 60 in.
T = telephone wire; not in B. & S. gage.

## Table 8.—Properties of Stranded Copper Wire

(John A. Roebling's Sons Co., Trenton, N. J.)

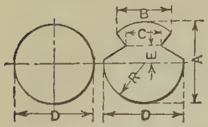
| Size, Cir. Mils or A.W.G. | Area, sq. in. | Concentric Strands | | | Ropes or Flexibles | | |
|---|---|---|---|---|---|---|---|
| | | Diam., in. | Lb. per 1000 ft. | Resistance at 68° F., Ohms per 1000 ft. | Diam., in. | Lb. per 1000 ft. | Resistance at 68° F., Ohms per 1000 ft. |
| 2,000,000 | 1.571 | 1.632 | 6175 | 0.00529 | 1.848 | 6296 | 0.00539 |
| 1,750,000 | 1.374 | 1.526 | 5403 | .00604 | 1.729 | 5509 | .00616 |
| 1,500,000 | 1.178 | 1.412 | 4631 | .00705 | 1.601 | 4722 | .00719 |
| 1,250,000 | 0.982 | 1.289 | 3859 | .00846 | 1.462 | 3935 | .00863 |
| 1,000,000 | .785 | 1.152 | 3088 | .01058 | 1.307 | 3148 | .01079 |
| 950,000 | .746 | 1.123 | 2933 | .01114 | 1.274 | 2991 | .01135 |
| 900,000 | .707 | 1.094 | 2779 | .01175 | 1.239 | 2833 | .01198 |
| 850,000 | .668 | 1.062 | 2624 | .01245 | 1.204 | 2676 | .01269 |
| 800,000 | .628 | 1.031 | 2470 | .01322 | 1.169 | 2518 | .01348 |
| 750,000 | .589 | 0.998 | 2316 | .01410 | 1.131 | 2361 | .01438 |
| 700,000 | .550 | .964 | 2161 | .01511 | 1.094 | 2204 | .01541 |
| 650,000 | .511 | .929 | 2007 | .01627 | 1.053 | 2046 | .01659 |
| 600,000 | .471 | .893 | 1853 | .01763 | 1.010 | 1889 | .01798 |
| 550,000 | .432 | .855 | 1698 | .01923 | 0.968 | 1731 | .01961 |
| 500,000 | .393 | .813 | 1544 | .02116 | .922 | 1574 | .02157 |
| 450,000 | .353 | .772 | 1389 | .02351 | .876 | 1417 | .02397 |
| 400,000 | .314 | .728 | 1235 | .02645 | .825 | 1259 | .02697 |
| 350,000 | .275 | .681 | 1081 | .03022 | .773 | 1102 | .03082 |
| 300,000 | .236 | .630 | 926 | .03526 | .713 | 944 | .03595 |
| 250,000 | .196 | .575 | 771.9 | .04231 | .651 | 787 | .04314 |
| 0000 | .166 | .528 | 653.3 | .04999 | .599 | 666.1 | .05097 |
| 000 | .132 | .470 | 518.1 | .06304 | .533 | 528.3 | .06428 |
| 00 | .105 | .419 | 410.9 | .07949 | .474 | 418.9 | .08105 |
| 0 | .0829 | .373 | 325.8 | .10024 | .423 | 332.2 | .10220 |
| 1 | .0657 | .328 | 258.4 | .12640 | .376 | 263.5 | .12888 |
| 2 | .0521 | .292 | 204.9 | .15939 | .335 | 208.9 | .16251 |
| 3 | .0413 | .260 | 162.5 | .20098 | .294 | 165.7 | .20492 |
| 4 | .0328 | .232 | 128.9 | .25343 | .262 | 131.4 | .25840 |
| 5 | .0260 | .206 | 102.2 | .31957 | .234 | 104.2 | .32584 |
| 6 | .0206 | .184 | 81.05 | .40297 | .208 | 82.64 | .41088 |
| 8 | .0130 | .146 | 50.97 | .64075 | .165 | 51.97 | .65332 |

## Table 9.—Weight of Stranded Copper Feed Wire

| Size Cir. Mils or A. W. G. | Weight per 1000 Feet, lb. | | | | | Weight per Mile, lb. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Bare | Weatherproof | | Slow Burning Weather-proof | Slow-Burn-ing | Bare | Weatherproof | | Slow Burning Weather-proof | Slow Burn-ing |
| | | Double Braid | Triple Braid | | | | Double Braid | Triple Braid | | |
| 2,000,000 | 6175 | 6690 | 7008 | ........ | 7540 | 32,604 | 35,323 | 37,000 | ........ | 39,800 |
| 1,750,000 | 5403 | 5894 | 6193 | ........ | 6700 | 28,529 | 31,119 | 32,700 | ........ | 35,400 |
| 1,500,000 | 4631 | 5098 | 5380 | ........ | 5830 | 24,453 | 26,915 | 28,400 | ........ | 30,800 |
| 1,250,000 | 3859 | 4264 | 4508 | ........ | 4940 | 20,378 | 22,516 | 23,800 | ........ | 26,100 |
| 1,000,000 | 3088 | 3456 | 3674 | 3860 | 3980 | 16,302 | 18,246 | 19,400 | 20,400 | 21,000 |
| 900,000 | 2779 | 3127 | 3332 | 3520 | 3640 | 14,672 | 16,513 | 17,600 | 18,600 | 19,200 |
| 800,000 | 2470 | 2799 | 2992 | 3180 | 3280 | 13,042 | 14,779 | 15,800 | 16,800 | 17,300 |
| 750,000 | 2316 | 2635 | 2822 | 3000 | 3100 | 12,227 | 13,913 | 14,900 | 15,850 | 16,350 |
| 700,000 | 2161 | 2471 | 2650 | 2820 | 2920 | 11,412 | 13,045 | 14,000 | 14,900 | 15,400 |
| 600,000 | 1853 | 2093 | 2235 | 2350 | 2460 | 9,871 | 11,052 | 11,800 | 12,400 | 13,000 |
| 500,000 | 1544 | 1765 | 1894 | 1990 | 2080 | 8,151 | 9,318 | 10,000 | 10,500 | 11,000 |
| 450,000 | 1389 | 1601 | 1724 | 1820 | 1900 | 7,336 | 8,452 | 9,100 | 9,600 | 10,000 |
| 400,000 | 1235 | 1436 | 1553 | 1650 | 1700 | 6,521 | 7,584 | 8,200 | 8,700 | 9,000 |
| 350,000 | 1081 | 1248 | 1345 | 1440 | 1500 | 5,706 | 6,589 | 7,100 | 7,600 | 7,900 |
| 300,000 | 926 | 1083 | 1174 | 1270 | 1310 | 4,891 | 5,721 | 6,200 | 6,700 | 6,900 |
| 250,000 | 772 | 907 | 985 | 1060 | 1120 | 4,076 | 4,788 | 5,200 | 5,600 | 5,900 |
| 0000 | 653 | 745 | 800 | 900 | 960 | 3,450 | 3,935 | 4,220 | 4,750 | 5,070 |
| 000 | 518 | 604 | 653 | 735 | 785 | 2,736 | 3,190 | 3,450 | 3,880 | 4,150 |
| 00 | 411 | 482 | 522 | 583 | 625 | 2,169 | 2,544 | 2,760 | 3,080 | 3,300 |
| 0 | 326 | 388 | 424 | 480 | 510 | 1,720 | 2,051 | 1,240 | 2,530 | 2,700 |
| 1 | 258 | 303 | 328 | 355 | 380 | 1,364 | 1,599 | 1,735 | 1,870 | 2,000 |
| 2 | 205 | 246 | 270 | 290 | 335 | 1,082 | 1,301 | 1,425 | 1,540 | 1,770 |
| 3 | 163 | 190 | 206 | 240 | 280 | 858 | 1,004 | 1,090 | 1,270 | 1,480 |
| 4 | 129 | 155 | 170 | 195 | 230 | 681 | 820 | 900 | 1,030 | 1,220 |
| 5 | 102 | 126 | 140 | 160 | 195 | 540 | 668 | 740 | 845 | 1,030 |
| 6 | 81 | 103 | 115 | 132 | 165 | 428 | 544 | 610 | 695 | 870 |
| 8 | 51 | 68 | 78 | 87 | 105 | 269 | 359 | 410 | 460 | 555 |

## Table 10.—Properties of Hard-drawn Copper and Bronze Trolley Wires

(American Brass Co., Waterbury, Conn.)



Coefficient of linear expansion, 0.000 009 4 per deg. F. Final modulus of elasticity, 16,000,000. Density, 0.32117 lb. per cu. in. at 68 deg. F. Based on Specifications of A.S.T.M. and A.R.E.A. for hard-drawn copper and bronze trolley wires, and Anaconda Specifications for Hitenso-trolley wire.

| | Grooved | Round | Grooved | Round | Grooved | Round | Grooved | Round | Grooved | Round |
|---|---|---|---|---|---|---|---|---|---|---|
| Size, cir. mils | 350,000 | 300,000 | | 0000 A.W.G. | | 000 A.W.G. | | 00 A.W.G. | | 0 A.W.G. |
| Area, cir. mils | 351,200 | 300,300 | 299,800 | 211,600 | 212,000 | 168,100 | 167,300 | 133,200 | 137,900 | 105,600 |
| " sq. in. | 0.2758 | 0.2359 | 0.2355 | 0.1662 | 0.1665 | 0.1320 | 0.1314 | 0.1046 | 0.1083 | 0.08296 |
| Lb. per 1000 ft. | 1063 | 909.0 | 907.6 | 640.5 | 641.7 | 508.8 | 506.4 | 403.2 | 417.4 | 319.7 |
| " " mile | 5612 | 4800 | 4792 | 3382 | 3388 | 2687 | 2674 | 2129 | 2204 | 1688 |
| Dimension, A, in. | 0.620 | ....... | 0.574 | ....... | 0.482 | ....... | 0.430 | ....... | 0.392 | ....... |
| " B, " | .376 | ....... | .376 | ....... | .376 | ....... | .340 | ....... | .318 | ....... |
| " C, " | .250 | ....... | .250 | ....... | .250 | ....... | .22 | ....... | .20 | ....... |
| " D, " | .620 | 0.548 | .574 | 0.460 | .482 | 0.410 | .429 | 0.365 | .388 | 0.325 |
| " E, " | 5/32 | ....... | 1/8 | ....... | 1/16 | ....... | 3/64 | ....... | 1/32 | ....... |
| " R, " | 0.310 | ....... | 0.287 | ....... | 0.241 | ....... | 0.215 | ....... | 0.196 | ....... |
| | Hard Drawn Copper—Conductivity, 97.16%, I.A.C.S. | | | | | | | | | |
| Resist., Ohms per mile at 68° F. | 0.1605 | 0.1877 | 0.1880 | 0.2664 | 0.2659 | 0.3353 | 0.3369 | 0.4230 | 0.4087 | 0.5336 |
| Tensile Strength, lb. per sq. in. | 42,800 | 46,400 | 44,200 | 49,000 | 46,600 | 51,000 | 48,500 | 52,800 | 50,200 | 54,500 |
| Breaking Strength, lb. | 11,800 | 10,940 | 10,410 | 8,143 | 7,759 | 6,734 | 6,373 | 5,524 | 5,437 | 4,521 |
| Elastic Limit, lb. | 6,492 | 6,019 | 5,725 | 4,479 | 4,267 | 3,703 | 3,505 | 3,038 | 2,990 | 2,487 |
| Elong. in 10 in., % | 4.50 | 4.50 | 4.50 | 3.75 | 3.75 | 3.25 | 3.25 | 2.80 | 2.80 | 2.40 |
| | Medium Strength Bronze—Conductivity, 65%, I.A.C.S. | | | | | | | | | |
| Resist., Ohms per mile at 68° F. | 0.2399 | 0.2805 | 0.2810 | 0.3981 | 0.3974 | 0.5012 | 0.5036 | 0.6324 | 0.6110 | 0.7976 |
| Tens. Strength, lb. per sq. in. | 56,200 | 57,800 | 57,800 | 61,000 | 61,000 | 63,000 | 63,000 | 65,000 | 65,000 | 68,000 |
| Breaking Strength, lb. | 15,500 | 13,630 | 13,610 | 10,140 | 10,160 | 8,318 | 8,278 | 6,801 | 7,040 | 5,641 |
| Elastic Limit, lb. | 9,300 | 8,180 | 8,167 | 6,083 | 6,094 | 4,991 | 4,967 | 4,081 | 4,224 | 3,385 |
| Elong. in 10 in., % | 4.00 | 4.50 | 4.00 | 3.75 | 3.25 | 3.25 | 2.75 | 2.75 | 2.25 | 2.40 |
| | Hitenso "A"—Conductivity, 85%, I.A.C.S. | | | | | | | | | |
| Resist., Ohms per mile at 68° F. | 0.1835 | 0.2145 | 0.2149 | 0.3045 | 0.3039 | 0.3832 | 0.3851 | 0.4836 | 0.4672 | 0.6099 |
| Tens. Strength, lb. per sq. in. | 57,000 | 58,300 | 58,300 | 61,500 | 61,500 | 64,000 | 64,000 | 66,000 | 66,000 | 68,000 |
| Breaking Strength, lb. | 15,720 | 13,750 | 13,730 | 10,220 | 10,240 | 8,450 | 8,410 | 6,906 | 7,148 | 5,641 |
| Elastic Limit, lb. | 9,432 | 8,250 | 8,238 | 6,132 | 6,144 | 5,070 | 5,046 | 4,143 | 4,289 | 3,385 |
| Elong. in 10 in., % | 4.00 | 4.50 | 4.00 | 3.75 | 3.25 | 3.25 | 2.75 | 2.75 | 2.25 | 2.40 |
| | Hitenso "B"—Conductivity, 80%, I.A.C.S. | | | | | | | | | |
| Resist., Ohms per mile at 68° F. | 0.1949 | 0.2279 | 0.2283 | 0.3235 | 0.3229 | 0.4072 | 0.4091 | 0.5138 | 0.4964 | 0.6480 |
| Tens. Strength, lb. per sq. in. | 59,500 | 61,500 | 61,500 | 65,000 | 65,000 | 67,000 | 67,000 | 69,000 | 69,000 | 72,000 |
| Breaking Strength, lb. | 16,410 | 14,510 | 14,480 | 10,800 | 10,820 | 8,846 | 8,804 | 7,219 | 7,473 | 5,973 |
| Elastic Limit, lb. | 9,846 | 8,703 | 8,690 | 6,481 | 6,494 | 5,308 | 5,282 | 4,332 | 4,484 | 3,584 |
| Elong. in 10 in., % | 4.00 | 4.50 | 4.00 | 3.75 | 3.25 | 3.25 | 2.75 | 2.75 | 2.25 | 2.40 |
| | High Strength Bronze—Conductivity, 40%, I.A.C.S. | | | | | | | | | |
| Resist., Ohms per mile at 68° F. | 0.3899 | 0.4559 | 0.4566 | 0.6470 | 0.6458 | 0.8144 | 0.8183 | 1.028 | 0.9928 | 1.296 |
| Tens. Strength, lb. per sq. in. | 62,500 | 64,800 | 64,800 | 69,000 | 69,000 | 71,000 | 71,000 | 73,000 | 73,000 | 76,000 |
| Breaking Strength, lb. | 17,240 | 15,280 | 15,260 | 11,470 | 11,490 | 9,374 | 9,329 | 7,638 | 7,906 | 6,305 |
| Elastic Limit, lb. | 10,340 | 9,170 | 9,156 | 6,880 | 6,893 | 5,624 | 5,598 | 4,583 | 4,744 | 3,783 |
| Elong. in 10 in., % | 4.00 | 4.50 | 4.00 | 3.75 | 3.25 | 3.25 | 2.75 | 2.75 | 2.25 | 2.40 |
| | Hitenso "C"—Conductivity, 55%, I.A.C.S. | | | | | | | | | |
| Resist., Ohms per mile at 68° F. | 0.2835 | 0.3315 | 0.3320 | 0.4705 | 0.4697 | 0.5923 | 0.5951 | 0.7473 | 0.7220 | 0.9426 |
| Tens. Strength, lb. per sq. in. | 65,000 | 68,000 | 68,000 | 72,000 | 72,000 | 74,000 | 74,000 | 76,000 | 76,000 | 78,000 |
| Breaking Strength, lb. | 17,930 | 16,040 | 16,010 | 11,970 | 11,990 | 9,770 | 9,724 | 7,952 | 8,231 | 6,471 |
| Elastic Limit, lb. | 10,760 | 9,623 | 9,608 | 7,179 | 7,193 | 5,862 | 5,834 | 4,771 | 4,938 | 3,882 |
| Elong. in 10 in., % | 4.00 | 4.50 | 4.00 | 3.75 | 3.25 | 3.25 | 2.75 | 2.75 | 2.25 | 2.40 |

**Table 11.—Properties of Hard-drawn Aluminum Wire**

| B. & S. Gage No. | Diameter, Mils | Cross-section | | Resistance, Ohms per 1000 ft. at 68° F. | Pounds per 1000 ft. | Pounds per Ohm | Feet per Ohm |
|---|---|---|---|---|---|---|---|
| | | Circular Mils | Square Inches | | | | |
| 0000 | 460. | 212 000. | 0.166 | 0.0804 | 195. | 2420. | 12400. |
| 000 | 410. | 168 000. | .132 | .101 | 154. | 1520. | 9860. |
| 00 | 365. | 133 000. | .105 | .128 | 122. | 957. | 7820. |
| 0 | 325. | 106 000. | .0829 | .161 | 97.0 | 602. | 6200. |
| 1 | 289. | 83 700. | .0657 | .203 | 76.9 | 379. | 4920. |
| 2 | 258. | 66 400. | .0521 | .256 | 61.0 | 238. | 3900. |
| 3 | 229. | 52 600. | .0413 | .323 | 48.4 | 150. | 3090. |
| 4 | 204. | 41 700. | .0328 | .408 | 38.4 | 94.2 | 2450. |
| 5 | 182. | 33 100. | .0260 | .514 | 30.4 | 59.2 | 1950. |
| 6 | 162. | 26 300. | .0206 | .648 | 24.1 | 37.2 | 1540. |
| 7 | 144. | 20 800. | .0164 | .817 | 19.1 | 23.4 | 1220. |
| 8 | 128. | 16 500. | .0130 | 1.03 | 15.2 | 14.7 | 970. |
| 9 | 114. | 13 100. | .0103 | 1.30 | 12.0 | 9.26 | 770. |
| 10 | 102. | 10 400. | .008 15 | 1.64 | 9.55 | 5.83 | 610. |
| 11 | 91. | 8230. | .006 47 | 2.07 | 7.57 | 3.66 | 484. |
| 12 | 81. | 6530. | .005 13 | 2.61 | 6.00 | 2.30 | 384 |
| 13 | 72. | 5180. | .004 07 | 3.29 | 4.76 | 1.45 | 304. |
| 14 | 64. | 4110. | .003 23 | 4.14 | 3.78 | 0.911 | 241. |
| 15 | 57. | 3260. | .002 56 | 5.22 | 2.99 | .573 | 191. |
| 16 | 51. | 2580. | .002 03 | 6.59 | 2.37 | .360 | 152. |
| 17 | 45. | 2050. | .001 61 | 8.31 | 1.88 | .227 | 120. |
| 18 | 40. | 1620. | .001 28 | 10.5 | 1.49 | .143 | 95.5 |
| 19 | 36. | 1290. | .001 01 | 13.2 | 1.18 | .0897 | 75.7 |
| 20 | 32. | 1020. | .000 802 | 16.7 | 0.939 | .0564 | 60.0 |
| 21 | 28.5 | 810. | .000 636 | 21.0 | .745 | .0355 | 47.6 |
| 22 | 25.3 | 642. | .000 505 | 26.5 | .591 | .0223 | 37.8 |
| 23 | 22.6 | 509. | .000 400 | 33.4 | .468 | .0140 | 29.9 |
| 24 | 20.1 | 404. | .000 317 | 42.1 | .371 | .008 82 | 23.7 |
| 25 | 17.9 | 320. | .000 252 | 53.1 | .295 | .005 55 | 18.8 |
| 26 | 15.9 | 254. | .000 200 | 67.0 | .234 | .003 49 | 14.9 |
| 27 | 14.2 | 202. | .000 158 | 84.4 | .185 | .002 19 | 11.8 |
| 28 | 12.6 | 160. | .000 126 | 106. | .147 | .001 38 | 9.39 |
| 29 | 11.3 | 127. | .000 099 5 | 134. | .117 | .000 868 | 7.45 |
| 30 | 10.0 | 101. | .000 078 9 | 169. | .0924 | .000 546 | 5.91 |
| 31 | 8.9 | 79.7 | .000 062 6 | 213. | .0733 | .000 343 | 4.68 |
| 32 | 8.0 | 63.2 | .000 049 6 | 269. | .0581 | .000 216 | 3.72 |
| 33 | 7.1 | 50.1 | .000 039 4 | 339. | .0461 | .000 136 | 2.95 |
| 34 | 6.3 | 39.8 | .000 031 2 | 428. | .0365 | .000 085 4 | 2.34 |
| 35 | 5.6 | 31.5 | .000 024 8 | 540. | .0290 | .000 053 7 | 1.85 |
| 36 | 5.0 | 25.0 | .000 019 6 | 681. | .0230 | .000 033 8 | 1.47 |
| 37 | 4.5 | 19.8 | .000 015 6 | 858. | .0182 | .000 021 2 | 1.17 |
| 38 | 4.0 | 15.7 | .000 012 3 | 1080. | .0145 | .000 013 4 | 0.924 |
| 39 | 3.5 | 12.5 | .000 009 79 | 1360. | .0115 | .000 008 40 | .733 |
| 40 | 3.1 | 9.9 | .000 007 77 | 1720. | .0091 | .000 005 28 | .581 |

## 5. WIRE GAGES AND SHEET METAL GAGES

The two wire gages in most extensive use in the United States are the American Wire Gage, also known as the Brown & Sharpe Gage, and the United States Steel Wire Gage more commonly called simply Steel Wire Gage.

Three other wire gages have remained in use but are now almost obsolete. These are the Birmingham Wire Gage, the Stubs Steel Wire Gage and the Old English Wire Gage. There are also certain special gages, as the drill and screw rod gages and the music wire gage. The one most frequently used is the Birmingham Gage sometimes called the Stubs Wire Gage. Its use in this country has been practically limited to large telegraph and telephone companies and certain departments of the Government. It is used by the Treasury Department for use in the collection of duties on imports of wire.

As there are many gages in use differing from each other, and even the thicknesses of a certain specified gage, as the Birmingham, are not assumed the same by all manufacturers, orders for sheets and wires should always state the weight per square foot, or the thickness in thousandths of an inch.

**THE AMERICAN WIRE GAGE** (Brown & Sharpe Gage) is practically the only gage used for copper and aluminum wire and, in general, for all wire used in electrical work. Sizes of stranded conductors larger than No. 0000 A.W.G. are specified by the cross-section in circular mils. The American Wire Gage also is used for sheets of copper, aluminum and other non-ferrous metals.

### Table 12.—Wire and Sheet Metal Gages

| Gage No. | American Wire Gage, B. & S. in. | Steel Wire Gage (Washburn & Moen), in. | Birmingham Wire Gage (Stubs Iron), in. | Old English Wire Gage (London), in. | Stubs Steel Wire Gage,* in. | British Standard Wire Gage, in. | Music Wire Gage | | U. S. Standard Sheet Metal Gage | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | American Steel Wire Co. | Roebling. Trenton Iron Co. | Thickness, in., Approx. | Weight, lb. per sq. ft. Iron | Steel |
| 0 000 000 | ........ | 0.4900 | ........ | ........ | ...... | 0 500 | ....... | ........ | 0.5000 | 20.0000 | 20.400 |
| 000 000 | ........ | .4615 | ........ | ........ | ...... | .464 | 0.004 | ........ | .4687 | 18.7500 | 19.125 |
| 00 000 | ........ | .4305 | ........ | ........ | ....... | .432 | .005 | ........ | .4375 | 17.5000 | 17.850 |
| 0 000 | 0.460 | .3938 | 0.454 | 0.454 | ....... | .400 | .006 | 0.007 | .4062 | 16.2500 | 16.575 |
| 000 | .410 | .3625 | .425 | .425 | ....... | .372 | .007 | .0075 | .3750 | 15.0000 | 15.300 |
| 00 | .365 | .3310 | .380 | .380 | ....... | .348 | .008 | .0085 | .3437 | 13.7500 | 14.025 |
| 0 | .325 | .3065 | .340 | .340 | ....... | .324 | .009 | .009 | .3125 | 12.5000 | 12.750 |
| 1 | .289 | .2830 | .300 | .300 | .227 | .300 | .010 | .010 | .2812 | 11.2500 | 11.475 |
| 2 | .258 | .2625 | .284 | .284 | .219 | .276 | .011 | .011 | .2656 | 10.6250 | 10.837 |
| 3 | .229 | .2437 | .259 | .259 | .212 | .252 | .012 | .012 | .2500 | 10.0000 | 10.200 |
| 4 | .204 | .2253 | .238 | .238 | .207 | .232 | .013 | .013 | .2344 | 9.3750 | 9.562 |
| 5 | .182 | .2070 | .220 | .220 | .204 | .212 | .014 | .014 | .2187 | 8.7500 | 8.925 |
| 6 | .162 | .1920 | .203 | .203 | .201 | .192 | .016 | .016 | .2035 | 8.1250 | 8.287 |
| 7 | .144 | .1770 | .180 | .180 | .199 | .176 | .018 | .018 | .1875 | 7.5000 | 7.650 |
| 8 | .128 | .1620 | .165 | .165 | .197 | .160 | .020 | .020 | .1719 | 6.8750 | 7.012 |
| 9 | .114 | .1483 | .148 | .148 | .194 | .144 | .022 | .022 | .1562 | 6.2500 | 6.375 |
| 10 | .102 | .1350 | .134 | .134 | .191 | .128 | .024 | .024 | .1406 | 5.6250 | 5.737 |
| 11 | .091 | .1205 | .120 | .120 | .188 | .116 | .026 | .026 | .1250 | 5.0000 | 5.100 |
| 12 | .081 | .1055 | .109 | .109 | .185 | .104 | .029 | .028 | .1094 | 4.3750 | 4.462 |
| 13 | .072 | .0915 | .095 | .095 | .182 | .092 | .031 | .030 | .0937 | 3.7500 | 3.825 |
| 14 | .064 | .0800 | .083 | .083 | .180 | .080 | .033 | .032 | .0781 | 3.1250 | 3.187 |
| 15 | .057 | .0720 | .072 | .072 | .178 | .072 | .035 | .034 | .0703 | 2.8125 | 2.869 |
| 16 | .051 | .0625 | .065 | .065 | .175 | .064 | .037 | .036 | .0625 | 2.5000 | 2.550 |
| 17 | .045 | .0540 | .058 | .058 | .172 | .056 | .039 | .038 | .0562 | 2.2500 | 2.295 |
| 18 | .040 | .0475 | .049 | .049 | .168 | .048 | .041 | .040 | .0500 | 2.0000 | 2.040 |
| 19 | .036 | .0410 | .042 | .040 | .164 | .040 | .043 | .042 | .0437 | 1.7500 | 1.785 |
| 20 | .032 | .0348 | .035 | .035 | .161 | .036 | .045 | .044 | .0375 | 1.5000 | 1.530 |
| 21 | .0285 | .0317 | .032 | .0315 | .157 | .032 | .047 | .046 | .0344 | 1.3750 | 1.402 |
| 22 | .0253 | .0286 | .028 | .0295 | .155 | .028 | .049 | .048 | .0312 | 1.2500 | 1.275 |
| 23 | .0226 | .0258 | .025 | .0270 | .153 | .024 | .051 | .051 | .0231 | 1.1250 | 1.147 |
| 24 | .0201 | .0230 | .022 | .0250 | .151 | .022 | .055 | .055 | .0250 | 1.0000 | 1 020 |
| 25 | .0179 | .0204 | .020 | .0230 | .148 | .020 | .059 | .059 | .0219 | 0.8750 | 0.892 |
| 26 | .0159 | .0181 | .018 | .0205 | .146 | .018 | .063 | .063 | .0187 | .7500 | .765 |
| 27 | .0142 | .0173 | .016 | .01875 | .143 | .0164 | .067 | .067 | .0172 | .6875 | .701 |
| 28 | .0126 | .0162 | .014 | .01650 | .139 | .0148 | .071 | .071 | .0156 | .6250 | .637 |
| 29 | .0113 | .0150 | .013 | .01550 | .134 | .0136 | .075 | .074 | .0141 | .5625 | .574 |
| 30 | .0100 | .0140 | .012 | .01375 | .127 | .0124 | .080 | .078 | .0125 | .5000 | .510 |
| 31 | .0089 | .0132 | .010 | .01225 | .120 | .0116 | 085 | .082 | .0109 | .4375 | .446 |
| 32 | .0080 | .0128 | .009 | .01125 | .115 | .0108 | .090 | .086 | .0101 | .40625 | .414 |
| 33 | .0071 | .0118 | .008 | .01025 | .112 | .0100 | .095 | .090 | .0094 | .3750 | .382 |
| 34 | .0063 | .0104 | .007 | .00950 | .110 | .0092 | .100 | .095 | .0086 | .34375 | .351 |
| 35 | .0056 | .0095 | .005 | .00900 | .108 | .0084 | .106 | .100 | .0078 | .3125 | .319 |
| 36 | .0050 | .0090 | .004 | .00750 | .106 | .0076 | .112 | .105 | .0070 | .28125 | .287 |
| 37 | .0045 | .0085 | ......... | .00650 | .103 | .0068 | .118 | .110 | .0066 | .26562 | .271 |
| 38 | .0040 | .0080 | ......... | .00575 | .101 | .0060 | .124 | .115 | .0062 | .2500 | .255 |
| 39 | .0035 | .0075 | ......... | .00500 | .099 | .0052 | .130 | .120 | ...... | ........ | ...... |
| 40 | .0031 | .0070 | ......... | .00450 | .097 | .0048 | .138 | .125 | ...... | ........ | ...... |
| 41 | ........ | .0066 | ......... | ......... | .095 | .0044 | .146 | .130 | ...... | ........ | ...... |
| 42 | ........ | .0062 | ......... | ......... | .092 | .0040 | .154 | ........ | ...... | ........ | ...... |
| 43 | ........ | .0060 | ......... | ......... | .088 | .0036 | .162 | ........ | ...... | ........ | ...... |
| 44 | ........ | .0058 | ......... | ......... | .085 | .0032 | .170 | ........ | ...... | ........ | ...... |
| 45 | ........ | .0055 | ......... | ......... | .081 | .0028 | .180 | ........ | ...... | ........ | ...... |
| 46 | ........ | .0052 | ......... | ......... | .079 | .0024 | ...... | ........ | ...... | ........ | ...... |
| 47 | ........ | .0050 | ......... | ......... | .077 | .0020 | ...... | ........ | ...... | ........ | ...... |
| 48 | ........ | .0048 | ......... | ......... | .075 | .0016 | ...... | ........ | ...... | ........ | ...... |
| 49 | ........ | .0046 | ......... | ......... | .072 | .0012 | ...... | ........ | ...... | ........ | ...... |
| 50 | ..... | .0044 | ......... | ......... | .069 | .0010 | ...... | ........ | ...... | ........ | ...... |

* See also Table 14.

**THE STEEL WIRE GAGE** is the same gage which formerly was known by the names of Washburn & Moen Gage and the American Steel & Wire Co. Gage. This gage, with certain of its sizes rounded off to even thousandths of an inch, is identical with the Roebling Gage. It is used for practically all steel wire in the United States.

While the two gages above are practically standard and have received official although not legal sanction, the trend of practice is toward the specifying of all sizes of wire and sheet material in decimals of an inch rather than by gage numbers.

**STUBS STEEL WIRE GAGE** has a somewhat limited use for tool steel wire and drill rods. It should not be confused with the Birmingham Wire Gage which is sometimes known as Stubs Iron Wire Gage.

**THE STANDARD WIRE GAGE** also known as the New British Standard, the English Legal Standard, or the Imperial Wire Gage is the legal standard of Great Britain for all wires. It was constructed by so modifying the Birmingham Wire Gage that the differences between successive diameters were the same for short lengths.

**MUSIC WIRE GAGE.**—Several gages are in use for music wire. Two of the most extensively used are given in Table 12. For data on others, see *Machinery's Handbook.*

**THE EDISON OR CIRCULAR MIL WIRE GAGE.**—The basis of the graduation is the sectional area, and the number of the wire corresponds. A wire of 100,000 circular mils area is No. 100; a wire of one-half the size will be No. 50; twice the size No. 200. With this gage, the number increases with the wire, and the number multiplied by 1000 will give the circular mils. The weight per mil-foot of copper wire, 0.000 003 027 05 pound, agrees with a specific gravity of 8.889. The ampere capacity which is given was deduced from experiments made in the Edison company's laboratory, and is based on a rise of temperature of 50° F. in the wire.

**The U. S. STANDARD GAGE FOR SHEET AND PLATE IRON AND STEEL.**—An Act of Congress in 1893 established the Standard Gage for sheet iron and steel. It is based on the fact that a cubic foot of iron weighs 480 pounds.

A sheet of iron 1 ft. square and 1 in. thick weighs 40 lb., or 640 oz., and 1 oz. in weight should be $^1/_{640}$ in. thick. The scale has been arranged so that each descriptive number

**Table 13.—Edison, or Circular Mil Gage for Electric Wires**

| Gage Number | Circular Mils | Diameter in Mils | Gage Number | Circular Mils | Diameter in Mils | Gage Number | Circular Mils | Diameter in Mils |
|---|---|---|---|---|---|---|---|---|
| 3 | 3,000 | 54.78 | 70 | 70,000 | 264.58 | 190 | 190,000 | 435.89 |
| 5 | 5,000 | 70.72 | 75 | 75,000 | 273.87 | 200 | 200,000 | 447.22 |
| 8 | 8,000 | 89.45 | 80 | 80,000 | 282.85 | 220 | 220,000 | 469.05 |
| 12 | 12,000 | 109.55 | 85 | 85,000 | 291.55 | 240 | 240,000 | 489.90 |
| 15 | 15,000 | 122.48 | 90 | 90,000 | 300.00 | 260 | 260,000 | 509.91 |
| 20 | 20,000 | 141.43 | 95 | 95,000 | 308.23 | 280 | 280,000 | 529.16 |
| 25 | 25,000 | 158.12 | 100 | 100,000 | 316.23 | 300 | 300,000 | 547.73 |
| 30 | 30,000 | 173.21 | 110 | 110,000 | 331.67 | 320 | 320,000 | 565.69 |
| 35 | 35,000 | 187.09 | 120 | 120,000 | 346.42 | 340 | 340,000 | 583.10 |
| 40 | 40,000 | 200.00 | 130 | 130,000 | 360.56 | 360 | 360,000 | 600.00 |
| 45 | 45,000 | 212.14 | 140 | 140,000 | 374.17 | ........ | .......... | .......... |
| 50 | 50,000 | 223.61 | 150 | 150,000 | 387.30 | ........ | .......... | .......... |
| 55 | 55,000 | 234.55 | 160 | 160,000 | 400.00 | ........ | .......... | .......... |
| 60 | 60,000 | 244.95 | 170 | 170,000 | 412.32 | ........ | .......... | .......... |
| 65 | 65,000 | 254.96 | 180 | 180,000 | 424.27 | ........ | .......... | .......... |

**Table 14.—Stubs Steel Wire Gage**

(For Nos. 1 to 50 see Table 12)

| No. | Size, in. | No. | Size, in. | No. | Size, in. | No. | Size, in. | No. | Size, in. | No. | Size, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Z | 0.413 | P | 0.323 | F | 0.257 | 51 | 0.066 | 61 | 0.038 | 71 | 0.026 |
| Y | .404 | O | .316 | E | .250 | 52 | .063 | 62 | .037 | 72 | .024 |
| X | .397 | N | .302 | D | .246 | 53 | .058 | 63 | .036 | 73 | .023 |
| W | .386 | M | .295 | C | .242 | 54 | .055 | 64 | .035 | 74 | .022 |
| V | .377 | L | .290 | B | .238 | 55 | .050 | 65 | .033 | 75 | .020 |
| U | .368 | K | .281 | A | .234 | 56 | .045 | 66 | .032 | 76 | .018 |
| T | .358 | J | .277 | 1 to 50 | See Table 12 | 57 | .042 | 67 | .031 | 77 | .016 |
| S | .348 | I | .272 | | | 58 | .041 | 68 | .030 | 78 | .015 |
| R | .339 | H | .266 | | | 59 | .040 | 69 | .029 | 79 | .014 |
| Q | .332 | G | .261 | | | 60 | .039 | 70 | .027 | 80 | .013 |

represents a certain number of ounces in weight and an equal number of 640ths of an inch in thickness.

The law enacts that on and after July 1, 1893, the new gage shall be used in determining duties and taxes levied on sheet and plate iron and steel; and that in its application a variation of 2 1/2% either way may be allowed.

The U. S. Standard Gage for Sheet and Plate Iron is a *weight* gage, not a thickness gage, and is based on the following assumptions:

| | Iron | Steel |
|---|---|---|
| Specific gravity | 7.7 | 7.854 |
| Weight per cubic foot | 480. | 489.6 |
| Weight per cubic inch | 0.2778 | 0.2833 |

It applies to iron only, and for steel the thickness would vary as 480/489.6.

**Table 15.—Twist Drill and Steel Wire Gage**

(Manufacturers' Standard)

| No. | Size, in. | No. | Size, in. | No. | Size, in. | No. | Size, in. | No. | Size, in. | No. | Size, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.2280 | 14 | 0.1820 | 27 | 0.1440 | 40 | 0.0980 | 53 | 0.0595 | 67 | 0.0320 |
| 2 | .2210 | 15 | .1800 | 28 | .1405 | 41 | .0960 | 54 | .0550 | 68 | .0310 |
| 3 | .2130 | 16 | .1770 | 29 | .1360 | 42 | .0935 | 55 | .0520 | 69 | .0292 |
| 4 | .2090 | 17 | .1730 | 30 | .1285 | 43 | .0890 | 56 | .0465 | 70 | .0280 |
| 5 | .2055 | 18 | .1695 | 31 | .1200 | 44 | .0860 | 57 | .0430 | 71 | .0260 |
| 6 | .2040 | 19 | .1660 | 32 | .1160 | 45 | .0820 | 58 | .0420 | 72 | .0250 |
| 7 | .2010 | 20 | .1610 | 33 | .1130 | 46 | .0810 | 59 | .0410 | 73 | .0240 |
| 8 | .1990 | 21 | .1590 | 34 | .1110 | 47 | .0785 | 60 | .0400 | 74 | .0225 |
| 9 | .1960 | 22 | .1570 | 35 | .1100 | 48 | .0760 | 61 | .0390 | 75 | .0210 |
| 10 | .1935 | 23 | .1540 | 36 | .1065 | 49 | .0730 | 62 | .0380 | 76 | .0200 |
| 11 | .1910 | 24 | .1520 | 37 | .1040 | 50 | .0700 | 63 | .0370 | 77 | .0180 |
| 12 | .1890 | 25 | .1495 | 38 | .1015 | 51 | .0670 | 64 | .0360 | 78 | .0160 |
| 13 | .1850 | 26 | .1470 | 39 | .0995 | 52 | .0635 | 65 | .0350 | 79 | .0145 |
| | | | | | | | | 66 | .0330 | 80 | .0135 |

# ROPES, CHAINS AND WIRE ROPE

## 1. ROPE

**Table 1.—Weight and Strength of Three-strand Manila Rope***

(Columbian Rope Co., Auburn, N. Y.,)

| Size, in. | | Approx. Weight, lb. per foot | Length in 1 lb. | Strength, lb. | | Size, in. | | Approx. Weight, lb. per foot | Length in 1 lb. | Strength, lb. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Diam. | Circum. | | | Breaking | Working | Diam. | Circum. | | | Breaking | Working |
| 3/16 | 5/8 | 0.015 | 66′-7″ | 450 | 90 | 1 5/16 | 4 | 0.480 | 2′-1″ | 15,000 | 3,000 |
| 1/4 | 3/4 | .020 | 50′-0″ | 600 | 120 | 1 1/2 | 4 1/2 | .600 | 1′-8″ | 18,500 | 3,700 |
| 5/16 | 1 | .029 | 34′-6″ | 1,000 | 200 | 1 5/8 | 5 | .744 | 1′-4″ | 22,500 | 4,500 |
| 3/8 | 1 1/8 | .041 | 24′-5″ | 1,350 | 230 | 1 3/4 | 5 1/2 | .895 | 1′-1″ | 26,500 | 5,300 |
| 7/16 | 1 1/4 | .0525 | 19′-0″ | 1,750 | 330 | 2 | 6 | 1.08 | 0′-11″ | 31,000 | 6,200 |
| 15/32 | 1 3/8 | .0625 | 16′-0″ | 2,250 | 450 | 2 1/8 | 6 1/2 | 1.25 | 9 1/2″ | 36,000 | 7,200 |
| 1/2 | 1 1/2 | .0750 | 13′-4″ | 2,650 | 530 | 2 1/4 | 7 | 1.46 | 8″ | 41,000 | 8,200 |
| 9/16 | 1 3/4 | .104 | 9′-7″ | 3,450 | 690 | 2 1/2 | 7 1/2 | 1.67 | 7″ | 46,500 | 9,300 |
| 5/8 | 2 | .133 | 7′-6″ | 4,400 | 880 | 2 5/8 | 8 | 1.91 | 6 1/4″ | 52,000 | 10,400 |
| 3/4 | 2 1/4 | .167 | 6′-0″ | 5,400 | 1030 | 2 7/8 | 8 1/2 | 2.15 | 5 5/8″ | 58,000 | 11,600 |
| 13/16 | 2 1/2 | .195 | 5′-2″ | 6,500 | 1300 | 3 | 9 | 2.42 | 5″ | 64,000 | 12,800 |
| 7/8 | 2 3/4 | .225 | 4′-5″ | 7,700 | 1540 | 3 1/8 | 9 1/2 | 2.69 | 4 1/2″ | 71,000 | 14,200 |
| 1 | 3 | .270 | 3′-9″ | 9,000 | 1800 | 3 1/4 | 10 | 2.99 | 4″ | 77,000 | 15,400 |
| 1 1/16 | 3 1/4 | .313 | 3′-2″ | 10,500 | 2100 | 3 1/2 | 11 | 3.67 | 3 1/4″ | 91,000 | 18,200 |
| 1 1/8 | 3 1/2 | .360 | 2′-9″ | 12,000 | 2400 | 3 3/4 | 12 | 4.36 | 2 3/4″ | 105,000 | 21,000 |
| 1 1/4 | 3 3/4 | .418 | 2′-5″ | 13,500 | 2700 | ...... | ...... | ...... | ...... | ...... | ...... |

* Four strand rope weighs about 5% more than 3-strand rope; the tensile strength is about the same as that of 3-strand rope.

**STRESS DUE TO LENGTH OF VERTICAL ROPE.**—In a very long rope the weight of the rope itself adds a considerable amount to the stress in the rope, and decreases the load that the rope can safely carry. The weight of the rope, as a whole, can be diminished by making it up of a series of ropes of constantly decreasing diameters, the smallest rope being at the lower end. Table 2 shows the length of rope, hanging vertical, that will develop the maximum stress in the rope. The figures are based on the values of Table 1.

**Table 2.—Length of Vertical Rope Required to Develop Maximum Stress**

| Circumference, in. | Length in feet to Develop Maximum Stress | | Circumference, in. | Length in feet to Develop Maximum Stress | | Circumference, in. | Length in feet to Develop Maximum Stress | |
|---|---|---|---|---|---|---|---|---|
| | Breaking | Working | | Breaking | Working | | Breaking | Working |
| ¾ | 3000 | 600 | 3½ | 3340 | 666 | 7 | 2800 | 560 |
| 1 | 3440 | 688 | 4 | 3120 | 624 | 8 | 2720 | 542 |
| 1½ | 3520 | 701 | 4½ | 3090 | 618 | 9 | 2625 | 535 |
| 2 | 3300 | 664 | 5 | 2895 | 579 | 10 | 2570 | 514 |
| 2½ | 3315 | 663 | 5½ | 2980 | 596 | 11 | 2480 | 496 |
| 3 | 3315 | 663 | 6 | 2870 | 574 | 12 | 2410 | 482 |

**MANILA ROPE FOR HOISTING OR TRANSMISSION.**—Rope for hoisting or power transmission is subject to severe usage. Ordinary rope chafes and grinds to powder in the center, while the exterior may look as though it was little worn. In bending a rope over a sheave, the strands and the yarns of these strands slide a small distance upon each other, causing friction, and wearing the rope internally. This condition can be alleviated by proper lubrication of the rope. The selection of the lubricant depends on the use of the rope. For indoor use, the lubricant should be one that will not dry out the rope, nor rub off easily. For outdoor use, a graphite lubricant of fine flake graphite should be worked into the yarns while the rope is being made, so that each fiber is impregnated.

In manufacturing rope, the fibers are first spun into a yarn, this yarn being twisted in a direction called "right hand." From 20 to 80 of these yarns, depending on the size of the rope, then are put together and twisted in the opposite direction, or "left hand," into a strand. Three of these strands, for a 3-strand, or four for a 4-strand rope, are then twisted together, the twist being again in the "right hand" direction. When the strand is twisted, it untwists each of the threads, and when the three strands are twisted together into rope, it untwists the strands, but again twists up the threads. It is this opposite twist that keeps the rope in its proper form. A weight hung on the end of a rope tends to untwist the rope, and make it longer. In untwisting, the threads twist up, and the weight will revolve until the strain of the untwisting strands just equals the strain of the threads being twisted tighter. In making a rope it is impossible to make these strains exactly balance each other. It is this fact that makes it necessary to take out the "turns" in a new rope, that is, untwist it when it is put to work. The proper twist that should be put in the threads has been ascertained approximately by experience.

The amount of work that the rope will do varies greatly. It depends not only on the quality of the fiber and the method of laying up the rope, but also on weather when the rope is used, the blocks or sheaves over which it is run, and the strain in proportion to the strain put upon the rope. The principal wear comes in practice from defective or badly set sheaves, from excess of load and exposure to storms.

Loads put on the rope should not exceed those given in the tables, for the most economical wear. Indications of excessive load will be the twist coming out of the rope, or one of the strands slipping out of its proper position. A certain amount of twist comes out in use in the first day or two, but the rope then should remain substantially the same. If it does not, the load is too great for the durability of the rope. Wear of the rope on the outside, while it is good on the inside, indicates chafing in running over pulleys or sheaves. If the blocks are very small, it will increase the sliding of the strands and threads, and result in a more rapid internal wear. For further information on rope for power transmission, see p. 24–48.

**TECHNICAL TERMS RELATING TO CORDAGE** most frequently used are:

**Yarn.**—Fibers twisted together. Yarns are laid up left-handed into *strands.*

**Thread.**—Two or more *small yarns* twisted together.

**String.**—The same as a thread but of a little larger *yarns.*

**Strand.**—Two or more *large yarns* twisted together. Strands are laid up right-handed into rope.

**Cord.**—Several threads twisted together.

**Rope.**—Several *strands* twisted together.

**Hawser.**—A rope of three *strands.* Hawsers are laid up left-handed into a cable.

**Shroud-laid.**—A rope of four *strands.*

**Cable.**—Three hawsers twisted together.

A rope is:

**Laid** by twisting strands together in making the rope.

**Spliced** by joining to another rope by interweaving the strands.

**Whipped.**—By winding a string around the end to prevent untwisting.

**Served.**—When covered by winding a yarn continuously and tightly around it.

**Parceled.**—By wrapping with canvas.

**Seized.**—When two parts are bound together by a yarn, thread or string.

**Payed.**—When painted, tarred or greased to resist wet.

**Haul.**—To pull on a rope.

**Taut.**—Drawn tight or strained.

SPLICING OF ROPE.—The splice in a transmission rope is not only the weakest part of the rope but is the first part to fail when the rope is worn out. If the rope is larger at the splice, the projecting part will wear on the pulleys and the rope fail from the cutting off of the strands. The following directions are given for splicing a 4-strand rope:

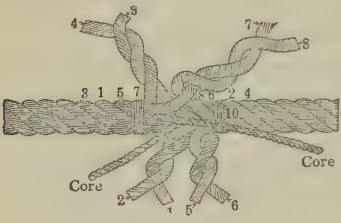


FIG. 1

Tie a piece of twine, 9 and 10, around the rope to be spliced, about 6 ft. from each end, and unlay the strands of each end back to the twine. Butt the ropes together and twist each corresponding pair of strands loosely to prevent them being tangled. (See Fig. 1.) The twine 10 is now cut, and the strand 8 unlaid and strand 7 carefully laid in its place for a distance of 4 1/2 ft. from the junction. The strand 6 is next unlaid about 1 1/2 ft. and strand 5 laid in its place. The ends of the cores are now cut off so they just meet. Unlay strand 1 a distance of 4 1/2 ft., laying strand 2 in its place. Unlay strand 3 about 1 1/2 ft., laying in strand 4.

Cut all the strands off to a length of about 20 in. for convenience in manipulation. The rope now assumes the form shown in Fig. 2 with the meeting points of the strands three feet apart.

Each pair of strands is successively subjected to the following operation:

From the point of meeting of the strands 8 and 7, unlay each one three turns; split both the strand 8 and the strand 7 in halves as far back as they are now unlaid and "whip" the end of each half strand with a small piece of twine.

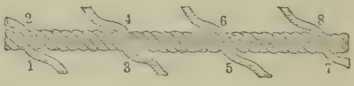


FIG. 2

The half of the strand 7 is now laid in three turns and the half of 8 also laid in three turns. The half strands now meet and are tied in a simple knot, 11, Fig. 3, making the rope at this point its original size.

The rope is now opened with a marlin spike and the half strand of 7 worked around the half strand of 8 by passing the end of the half strand 7 through the rope, as shown, drawn taut, and again worked around this half strand until it reaches the half strand 13 that was not laid in. This half strand 13 is now split, and the half strand 7 drawn through the opening thus made, and then tucked under the two adjacent strands, as shown in Fig. 4. The other half of the strand 8 is now wound around the other half strand 7 in the same manner. After each pair of strands has been treated in this manner, the ends are cut off at 12, leaving them about 4 in. long. After a few days wear they will draw into the body of the rope or wear off.

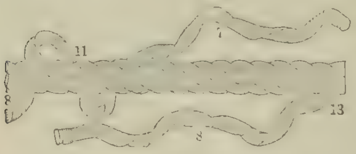


FIG. 3

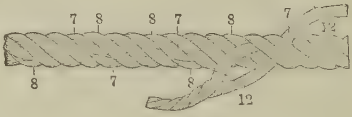


FIG. 4

KNOTS.—The principle of a knot is that no two parts, which would move in the same direction if the rope were to slip, should lay alongside of and touching each other. (See Fig. 5.)

The Bowline is one of the most useful knots; it will not slip, and after being stressed is easily untied. Commence by making a bight in the rope; then put the end through the bight and under the standing part as shown in *G*; then pass the end again through the bight, and haul tight.

The Square or Reef Knot must not be mistaken for the "granny" knot that slips under a stress. Knots *H*, *K*, and *M* are easily untied after being under stress. The knot *M* is useful when the rope passes through an eye and is held by the knot, as it will not slip and is easily untied after being stressed.

The timber hitch *S* looks as though it would give way, but it will not; the greater the stress the tighter it will hold. The wall knot looks complicated, but is easily made by proceeding as follows: Form a bight with strand 1 and pass strand 2 around the end of it, and strand 3 around the end of 2 and then through the bight of 1 as shown at *Z*. Haul the ends taut; the appearance is as shown at *AA*. The end of the strand 1 is now laid over the center of the knot, strand 2 laid over 1, and strand 3 over 2; the end of 3 is passed through the bight of 1 as shown at *BB*. Haul all the strands taut as shown at *CC*.

**Varieties of Knots.**—A great number of knots have been devised of which a few only are illustrated; those selected are the most frequently used. In Fig. 5, they are shown before being drawn taut, in order to show the position of the parts. The names usually given to them are:

A. Bight of a rope.
B. Simple or overhand knot.
C. Figure 8 knot.
D. Double knot.
E. Boat knot.
F. Bowline, first step.
G. Bowline, second step.
H. Bowline completed.
I. Square or reef knot.
J. Sheet bend or weaver's knot.
K. Sheet bend with a toggle.
L. Carrick bend.
M. Stevedore knot completed.
N. Stevedore knot commenced.
O. Slip knot.
P. Flemish loop.
Q. Chain knot with toggle.
R. Half-hitch.
S. Timber-hitch.
T. Clove-hitch.
U. Rolling-hitch.
V. Timber-hitch and half-hitch.
W. Blackwall-hitch.
X. Fisherman's bend.
Y. Round turn and half-hitch.
Z. Wall knot commenced.
AA. Wall knot completed.
BB. Wall knot crown commenced.
CC. Wall knot crown completed.

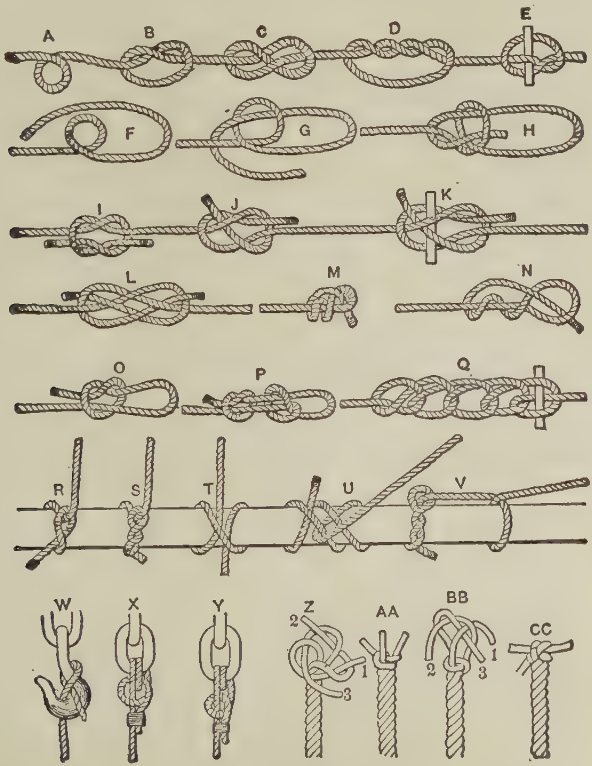


Fig. 5. Varieties of Knots

**EFFICIENCY OF KNOTS** as a percentage of the full strength of the rope, and the factor of safety when used with the stresses given below are as follows:

| Kind of Knot. | Efficiency, % | Factor of Safety |
|---|---|---|
| Eye splice over an iron thimble | 90 | 6.3 |
| Short splice in the rope | 80 | 5.6 |
| Timber hitch, round turn, half-hitch | 65 | 4.5 |
| Bowline slip knot, clove hitch | 60 | 4.2 |
| Square knot, weaver's knot, sheet bend | 50 | 3.5 |
| Flemish loop, overhand knot | 45 | 3.1 |
| Full strength of dry rope, average of four tests | 100 | 7.0 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Diameter of rope, in. | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 5/8 | 1 3/4 |
| Working stress, lb. | 1000 | 1250 | 1500 | 1900 | 2200 | 2600 | 3000 |

## 2. CHAINS

**CLASSES OF CHAIN.**—**Crane Chain**, for slings, hoists, steam shovels, and marine uses, and where an all-iron chain is desired.

**Proof Coil**, for railroad cars, construction and forestry work.

The A.S.T.M. standard specifications (A56–30) for iron and steel chain provide:

**Crane Chain** shall be made from wrought iron free from any admixture of iron scrap or steel, and lap fire welded.

**Proof Coil** may be made from open-hearth steel or all reworked steel and iron scrap. It may be made by fire- or electric welding. Chemical requirements are: Wrought-iron crane chain, Mn (max.), 0.10% steel proof coil, P (max.), 0.05%, S (max.), 0.05%.

**Physical Properties and Tests.**—All chain shall be subjected to the proof test load, shown in Table 3 for the respective size chain, without showing any defects. Test specimens, of not less than 2 ft., cut from the finished chain, shall conform to the minimum requirements for the break test loads specified in Table 4. The elongation, determined in a gage length of from 12 to 18 in. to the nearest link, shall not be less than 15% for both crane and proof coil chain.

**Standard Weights and Dimensions.**—The nominal length of 100 links, the nominal weight per 100 ft., and the nominal dimensions of links are specified in Table 4, with a permissible variation for length and weight of chain of ± 4%. The length is determined by measuring to the inside of the end links, and after the proof test, and with a load not exceeding 10% of the proof test load specified in Table 3 to take up slack. The diameter of the bar in all chain is not to be less than the nominal size in the body and welds. The diameter of the weld shall not exceed the diameter of the bar by more than 25%.

**CAUSES OF CHAIN FAILURE.**—**1. Brittle Failure** caused by cold-working of the metal surface. The metal fractures with no elongation, or with less elongation than in the ordinary tensile failure mentioned under (6). This type of fracture usually is caused by a brittle surface skin developed on the metal by impacts received in service. Brittle

**Table 3.—Proof and Break Test Loads**

(A.S.T.M. Specification A56–30)

| Nominal Size of Chain Bar, in. | Proof Test Load, lb. | | Break Test Load, lb. | | Safe Working Load, lb. | |
|---|---|---|---|---|---|---|
| | Crane Chain, Wrought Iron | Proof Coil, Iron or Steel | Crane Chain, Wrought Iron | Proof Coil, Iron or Steel | Crane Chain, Wrought Iron | Proof Coil, Iron or Steel |
| 1/4 | 1,767 | 1,700 | 3,535 | 3,400 | 1,060 | 850 |
| 5/16 | 2,760 | 2,650 | 5,520 | 5,300 | 1,655 | 1,325 |
| 3/8 | 3,975 | 3,850 | 7,950 | 7,700 | 2,385 | 1,925 |
| 7/16 | 5,415 | 5,250 | 10,830 | 10,500 | 3,250 | 2,625 |
| 1/2 | 7,072 | 6,850 | 14,145 | 13,700 | 4,240 | 3,425 |
| 9/16 | 8,947 | 8,650 | 17,895 | 17,300 | 5,370 | 4,325 |
| 5/8 | 11,047 | 10,700 | 22,095 | 21,400 | 6,630 | 5,350 |
| 3/4 | 15,900 | 15,350 | 31,800 | 30,700 | 9,540 | 7,675 |
| 7/8 | 21,622 | 20,900 | 43,245 | 41,800 | 12,960 | 10,450 |
| 1 | 28,275 | 27,350 | 56,550 | 54,700 | 16,950 | 13,675 |
| 1 1/8 | 33,400 | ...... | 66,800 | ...... | 20,040 | ...... |
| 1 1/4 | 41,250 | ...... | 82,500 | ...... | 24,750 | ...... |
| 1 3/8 | 49,900 | ...... | 99,800 | ...... | 29,910 | ...... |
| 1 1/2 | 59,350 | ...... | 118,700 | ...... | 35,600 | ...... |
| 1 5/8 | 69,750 | ...... | 139,500 | ...... | 41,800 | ...... |
| 1 3/4 | 80,800 | ...... | 161,600 | ...... | 48,450 | ...... |
| 1 7/8 | 92,750 | ...... | 185,500 | ...... | 55,300 | ...... |
| 2 | 105.500 | ...... | 211,100 | ...... | 63,300 | ...... |

**Table 4.—Nominal Weights and Dimensions of Chain**

(A.S.T.M. Specification A56–30)

| Nominal Size of Chain Bar, in. | Actual Size of Material, in. | Nominal Length of 100 Links, in. | | Nominal Weight per 100 ft., lb. | | Nominal Dimensions of Link, in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Crane Chain | Proof Coil | Crane Chain | Proof Coil | Crane Chain | | | | Proof Coil | | | |
| | | | | | | Outside | | Inside | | Outside | | Inside | |
| | | | | | | Length | Width | Length | Width | Length | Width | Length | Width |
| 1/4 | 9/32 | 86 | 100 | 78 | 70 | 1 27/64 | 1 | 55/64 | 7/16 | 1 9/16 | 1 1/16 | 1 | 1/2 |
| 5/16 | 11/32 | 100 | 111 | 115 | 105 | 1 11/16 | 1 3/16 | 1 | 1/2 | 1 51/64 | 1 3/16 | 1 7/64 | 1/2 |
| 3/8 | 13/32 | 109 | 123 | 166 | 158 | 1 29/32 | 1 7/16 | 1 3/32 | 5/8 | 2 3/64 | 1 7/16 | 1 15/64 | 5/8 |
| 7/16 | 15/32 | 122 | 138 | 220 | 210 | 2 5/32 | 1 5/8 | 1 7/32 | 11/16 | 2 5/16 | 1 11/16 | 1 3/8 | 3/4 |
| 1/2 | 17/32 | 134 | 150 | 275 | 265 | 2 13/32 | 1 13/16 | 1 11/32 | 3/4 | 2 9/16 | 1 7/8 | 1 1/2 | 13/16 |
| 9/16 | 19/32 | 156 | 175 | 350 | 335 | 2 3/4 | 1 15/16 | 1 9/16 | 3/4 | 2 15/16 | 2 1/16 | 1 3/4 | 7/8 |
| 5/8 | 21/32 | 169 | 188 | 430 | 410 | 3 | 2 3/16 | 1 11/16 | 7/8 | 3 3/16 | 2 5/16 | 1 7/8 | 1 |
| 3/4 | 25/32 | 188 | 213 | 615 | 580 | 3 7/16 | 2 9/16 | 1 7/8 | 1 | 3 11/16 | 2 11/16 | 2 1/8 | 1 1/8 |
| 7/8 | 29/32 | 225 | 250 | 820 | 780 | 4 1/16 | 3 1/16 | 2 1/4 | 1 1/4 | 4 5/16 | 3 3/16 | 2 1/2 | 1 3/8 |
| 1 | 1 1/32 | 256 | 275 | 1045 | 1000 | 4 5/8 | 3 7/16 | 2 9/16 | 1 3/8 | 4 13/16 | 3 9/16 | 2 3/4 | 1 1/2 |
| 1 1/8 | 1 5/32 | 288 | ...... | 1310 | ...... | 5 3/16 | 3 15/16 | 2 7/8 | 1 5/8 | ...... | ...... | ...... | ...... |
| 1 1/4 | 1 9/32 | 306 | ...... | 1600 | ...... | 5 5/8 | 4 5/16 | 3 1/16 | 1 3/4 | ...... | ...... | ...... | ...... |
| 1 3/8 | 1 13/32 | 363 | ...... | 1930 | ...... | 6 7/16 | 4 11/16 | 3 5/8 | 1 7/8 | ...... | ...... | ...... | ...... |
| 1 1/2 | 1 17/32 | 387 | ...... | 2335 | ...... | 6 15/16 | 5 1/16 | 3 7/8 | 2 | ...... | ...... | ...... | ...... |
| 1 5/8 | 1 21/32 | 425 | ...... | 2740 | ...... | 7 9/16 | 5 7/16 | 4 1/4 | 2 1/8 | ...... | ...... | ...... | ...... |
| 1 3/4 | 1 25/32 | 475 | ...... | 3180 | ...... | 8 5/16 | 5 13/16 | 4 3/4 | 2 1/4 | ...... | ...... | ...... | ...... |
| 1 7/8 | 1 29/32 | 525 | ...... | 3650 | ...... | 9 1/16 | 6 5/16 | 5 1/4 | 2 1/2 | ...... | ...... | ...... | ...... |
| 2 | 2 1/32 | 575 | ...... | 4100 | ...... | 9 13/16 | 6 11/16 | 5 3/4 | 2 5/8 | ...... | ...... | ...... | ...... |

failures show a more or less crystalline fracture, especially where the brittleness is present in an extreme degree. In service, fractures caused by cold-working the surface occur most frequently in the sides or near the shoulder of the link. Where the surface hardening is extreme, the fracture extends directly across the metal of the link in an approximately plane surface. Brittle failures usually are found in chain which has failed under the impact of suddenly-applied loads.

**2. Failure in the Weld** covers any failure involving partial or total separation of the welded surfaces. In old chain, welded surfaces often show signs of progressive deterioration and oxidation.

**3. Failure Due to Repeated Severe Bending or Deformation of the Links** is similar to that produced in a bar by repeated bending back and forth. This is common in long-link chains, as conveyor chains. The fractured surface usually is crystalline both in wrought iron and steel.

**4. Fatigue Failure** produced by many repetitions of stress exceeding the fatigue limit of the metal. In fatigue tests on chain links, the fracture usually occurs at about the quarter point of the link. The fracture extends directly across the side of the link. Fatigue failure occurs in chain in machines or appliances used for repeated liftings of a given load, but is not common in crane chains and sling chains. There is no elongation of the metal. The fractured surface is usually plane and crystalline.

**5. Brittle Failure Caused by Defects in the Metal.**—Occasional types of brittle fracture are caused by defects in the original iron or steel. Such failures are comparatively rare. The proof test by the manufacturer detects brittleness from this cause, and the links are removed. Brittle fractures, due to overheating during welding, are rare.

**6. Ordinary Tensile Failure,** accompanied by full elongation of the metal of the links, usually occurs in new chain of good quality when the chain is required to carry a load greater than its ultimate tensile strength. The metal of the links elongates greatly and the chain "pulls stiff" before final failure. The fracture usually occurs in the sides of the links near the welded end. The metal "necks down" and pulls apart as in an ordinary tension test. The fractured surfaces are generally silky in the case of steel, or fibrous in the case of wrought iron.

**USE OF CHAIN.**—The diameter of a sheave or winding drum used with a short link chain should be from 25 to 30 times the diameter of the stock used for the link, to minimize the bending stresses in the chain as it passes over the sheave or drum. Where the application requires a longer link the chain manufacturer should be consulted as to the proper size of drum or sheave. Sheaves and drums used should be grooved to receive the chain. These grooves should be of such shape and size as to allow the chain to work smoothly without twisting. Chain should not be wound on sheaves, shafts, or drums without proper grooves.

**Splicing of Broken Chain.**—A dangerous practice, which should be prohibited, is that

### Table 5.—Safe Working Loads, Pounds, for Iron Sling Chains

(Chain Institute Standard, 1931)

| Size of Chain, in. | Single Sling Chain | Double Sling Chains | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | Angle α of Chains. See Fig. 6 | | | | | | |
| | | 60° | 45° | 30° | 20° | 15° | 10° | 5° |
| 3/8 | 2,700 | 4,700 | 3,800 | 2,700 | 1,850 | 1,450 | 940 | 470 |
| 7/16 | 3,450 | 5,900 | 4,900 | 3,450 | 2,350 | 1,750 | 1,200 | 600 |
| 1/2 | 4,500 | 7,800 | 6,350 | 4,500 | 3,100 | 2,300 | 1,570 | 780 |
| 5/8 | 6,900 | 12,000 | 9,750 | 6,900 | 4,700 | 3,550 | 2,400 | 1,200 |
| 3/4 | 10,100 | 17,500 | 14,000 | 10,100 | 6,900 | 5,200 | 3,500 | 1,750 |
| 7/8 | 14,000 | 24,000 | 19,500 | 14,000 | 9,600 | 7,250 | 4,900 | 2,400 |
| 1 | 18,600 | 32,000 | 26,000 | 18,600 | 12,700 | 9,650 | 6,500 | 3,200 |
| 1 1/8 | 23,400 | 40,000 | 33,000 | 23,400 | 16,000 | 12,000 | 8,000 | 4,000 |
| 1 1/4 | 28,800 | 50,000 | 40,500 | 28,800 | 19,700 | 15,000 | 10,000 | 5,000 |
| 1 3/8 | 34,500 | 60,000 | 49,000 | 34,500 | 23,500 | 17,800 | 12,000 | 6,000 |
| 1 1/2 | 40,800 | 70,000 | 57,500 | 40,800 | 28,000 | 21,000 | 14,000 | 7,000 |
| 1 5/8 | 46,500 | 80,000 | 66,000 | 46,500 | 31,800 | 24,000 | 16,000 | 8,000 |
| 1 3/4 | 52,500 | 91,000 | 74,000 | 52,500 | 36,000 | 27,000 | 18,000 | 9,100 |
| 2 | 66,600 | 115,000 | 94,000 | 66,600 | 45,600 | 34,500 | 23,000 | 11,500 |

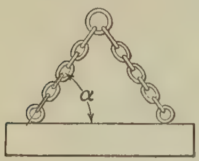


Fig 6. Sling Chain Angles

of splicing a broken chain by inserting a bolt between two links, with the head of the bolt and the nut sustaining the load, or of passing one link through another and inserting a bolt or nail to hold it.

**Kinked, Knotted, or Twisted Chain.**—A lift never should be made with a kink, knot, or twist in the chain. The chain should be perfectly straight. Twisting and dragging chain over pipe and other materials injures the chain.

**Safe Loads.**—Some plants stamp on a metal tag attached to each chain the safe vertical load which may be lifted with that chain. A better way is to stamp the safe load or a reference number for the chain load on the ring or hook. Stamp marks should not be placed on chain links as they may form points of weakness in the surface of the links when they are in cold worked condition, which may start a crack. Some chain users weld to the ring or connecting link of the chain a flat link on which the safe load or number may be marked. This flat link does not form a part of the chain to which the load is applied. A greater load can be lifted safely with a given size of chain or sling when the chain or all of the legs of the sling are in a vertical position. Safe working loads for iron sling chains are given in Table 5.

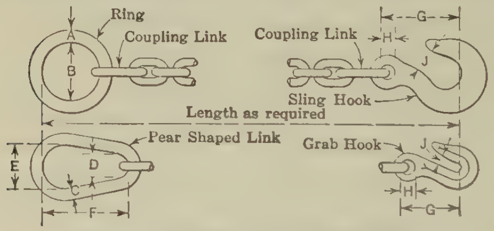


Fig. 7. Dimensions of Single Sling Chains

**IRON SPECIFICATIONS.**—**Grade A Iron Chain** shall be made from all-puddled iron, free from any admixture of iron or steel scrap. The original muck bars shall be twice

### Table 6.—Single Sling Chain Dimensions, Inches (Fig. 7)

(Chain Institute Standard, 1931)

| Chain Size | Coupling Links | Rings | | Pear-shaped End Links | | | | Sling Hooks | | | Grab Hooks | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *J* | *G* | *H* | *J* |
| 3/8 | 7/16 | 3/4 | 4 | 5/8 | 1 1/2 | 3 | 6 | 3 3/4 | 7/8 | 1 1/4 | 2 13/16 | 5/8 | 15/32 |
| 7/16 | 1/2 | 3/4 | 4 | 3/4 | 1 1/2 | 3 | 6 | 4 3/8 | 1 | 1 9/16 | 3 3/8 | 3/4 | 17/32 |
| 1/2 | 5/8 | 7/8 | 4 | 7/8 | 1 3/4 | 3 | 6 | 5 1/2 | 1 1/8 | 1 13/16 | 4 | 7/8 | 19/32 |
| 5/8 | 3/4 | 1 1/8 | 5 | 1 | 2 | 4 | 8 | 6 1/2 | 1 1/4 | 2 1/8 | 4 3/4 | 1 | 25/32 |
| 3/4 | 7/8 | 1 3/8 | 6 | 1 1/4 | 2 | 4 | 8 | 8 1/2 | 1 3/4 | 2 11/16 | 5 3/4 | 1 1/8 | 29/32 |
| 7/8 | 1 | 1 1/2 | 6 | 1 1/2 | 3 | 6 | 11 | 10 1/8 | 2 | 3 | 6 7/8 | 1 1/4 | 1 1/32 |
| 1 | 1 1/8 | 1 3/4 | 7 | 1 3/4 | 3 | 6 | 12 | 10 1/8 | 2 | 3 | 8 | 1 3/8 | 1 5/32 |
| 1 1/8 | 1 1/4 | 1 7/8 | 8 | 1 7/8 | 3 1/2 | 7 | 13 | 12 | 2 | 3 3/4 | ...... | ...... | ...... |
| 1 1/4 | 1 1/2 | 2 1/8 | 9 | 2 1/8 | 3 1/2 | 7 | 14 | 12 | 2 | 3 3/4 | ...... | ...... | ...... |
| 1 3/8 | 1 5/8 | 2 3/8 | 10 | 2 3/8 | 4 | 8 | 15 | 15 3/8 | 2 3/8 | 4 1/2 | ...... | ...... | ...... |
| 1 1/2 | 1 3/4 | 2 3/4 | 12 | 2 1/2 | 4 | 8 | 16 | 16 3/4 | 2 1/2 | 5 | ...... | ...... | ...... |

**Table 7.—Double Sling Chain Dimensions, Inches** (Fig. 8)

(Chain Institute Standard, 1931)

| Chain Size | Coupling Links | Rings | | Pear-shaped End Links | | | | Sling Hooks | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *J* |
| 3/8 | 7/16 | 7/8 | 4 | 3/4 | 1 1/2 | 3 | 6 | 3 3/4 | 7/8 | 1 1/4 |
| 7/16 | 1/2 | 1 | 4 | 7/8 | 1 3/4 | 3 | 6 | 4 3/8 | 1 | 1 9/16 |
| 1/2 | 5/8 | 1 1/8 | 5 | 1 | 2 | 4 | 8 | 5 1/2 | 1 1/8 | 1 13/16 |
| 5/8 | 3/4 | 1 3/8 | 6 | 1 1/4 | 2 | 4 | 8 | 6 1/2 | 1 1/4 | 2 1/8 |
| 3/4 | 7/8 | 1 3/4 | 7 | 1 1/2 | 3 | 6 | 11 | 8 1/2 | 1 3/4 | 2 11/16 |
| 7/8 | 1 | 1 7/8 | 8 | 1 3/4 | 3 | 6 | 12 | 10 1/8 | 2 | 3 |
| 1 | 1 1/8 | 2 1/8 | 9 | 1 7/8 | 3 1/2 | 7 | 13 | 10 1/8 | 2 | 3 |
| 1 1/8 | 1 1/4 | 2 3/8 | 10 | 2 1/8 | 3 1/2 | 7 | 14 | 12 | 2 | 3 3/4 |
| 1 1/4 | 1 1/2 | 2 3/4 | 12 | 2 3/8 | 4 | 8 | 15 | 12 | 2 | 3 3/4 |
| 1 3/8 | 1 5/8 | 3 | 12 | 2 1/2 | 4 | 8 | 16 | 15 3/8 | 2 3/8 | 4 1/2 |
| 1 1/2 | 1 3/4 | 3 1/4 | 12 | 2 3/4 | 5 | 8 | 16 | 16 3/4 | 2 1/2 | 5 |

piled and rerolled. All bars shall be the full length of the pile. The physical properties of the iron shall be such, when made into chain, as to conform to the breaking loads and elongation called for by specifications A56–30 for iron crane chain. (See p. 6–31.)

**Grade B Iron Chain** shall be rolled from a box pile, top, bottom, and sides of which shall have been rolled from muck bar, and the filler, which consists of muck bar, shall be the full length of the pile. The physical properties shall be as specified for Grade A chain.

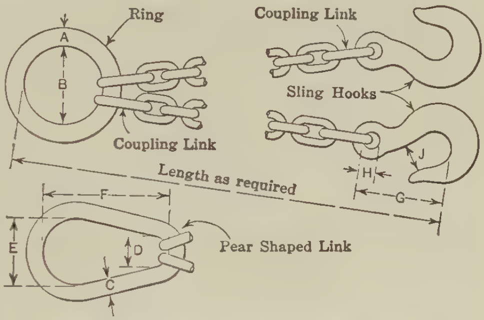


Fig. 8. Dimensions of Double Sling Chains

**LUBRICATION.**—To lessen wear of the chain links when wound on a drum and passed over sheaves, they should be lubricated at frequent and regular intervals with grease or heavy oil mixed with graphite. When lubricating, the chain should be slack and the oil brushed well in between the ends of the links. Grease and oil should not be used in places such as foundries where grit and sand particles are present, resulting in severe abrasive action. Sling chains should not be lubricated, as the lubricant would be transferred to the bodies of men handling the chain, introducing a hazard. The lubricant may also increase the slipping hazard.

**STORAGE.**—When chains are not in use they should be kept in a place suitable for storage. They should not be exposed to corrosive fumes or liquids.

**RINGS, HOOKS, AND END LINKS.**—When hooks, rings, or special end links are used with chain in sustaining loads, care should be taken to select the attachments of the type, grade, and size suitable for service with the size of chain used. Specifications for sling hooks, grab hooks, end rings, and pear-shaped end links given in Tables 6 and 7 are approved by the Chain Institute. It is recommended that they be followed. Forged steel hooks always are preferable, and especially for lifting heavy loads. Cast steel hooks frequently contain such defects as blow holes which cannot be detected by inspection. Hooks of very large capacity frequently are made of laminated steel plate.

**METHOD OF EXAMINATION.**—It is important that all hoisting chains be examined at frequent and regular intervals. The examination should include inspection of the links for stretch, wear, gouge marks, open welds, and especially a careful inspection of the surface to ascertain the need of annealing or normalizing, as indicated by the peened, hammered, or polished surface of the links. The examination should be made by a person conversant with the defects and flaws liable to be found, and who has the knowledge and authority to decide when the chain should be removed from service for repair, annealing, normalizing, or replacement.

**BENT HOOKS.**—Hooks which have become bent, due to overloads, should not be straightened and put back into service. When a hook once is bent the stress in the metal so tends to weaken it that its strength afterward is doubtful. Fig. 9 shows a safety hook on which the locking pawl prevents loads from becoming unhooked, and the point of the hook from fouling.

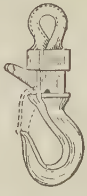


Fig. 9. Safety Hook

**STUD LINK CHAIN.**—Stud link chain, or studded chain, is used only as an anchor chain, and operates over a windlass. It should be made in accordance with the rules of the American Bureau of Shipping.

## 3. WIRE ROPE

The following notes and tables are compiled from data furnished by the American Steel & Wire Co., Cleveland. For further information see catalogs of the American Steel & Wire Co., John A. Roebling's Sons Co., and A. Leschens & Sons Co.

Wire ropes, which have almost entirely superseded chains and manila rope for haulage and hoisting purposes, are made with a varying number of wires to the strand, and a varying number of strands to the rope, according to the service in which they are to be used and the degree of flexibility required. Five grades of rope are usually manufactured, as regards the material used, *viz.*: Iron, cast steel, mild plow steel, plow-steel, and special grades called "Monitor," "Blue center" plow steel, etc. Haulage rope, for mines, docks, etc., usually consists of 6 strands of 7 wires each laid around a hemp core. Hoisting rope, fore levators, mines, coal and ore hoists, conveyors, derricks, steam shovels, dredges, logging, etc., consists of 6 strands of 19 wires each, with a single hemp core. A more flexible rope, for crane service, etc., consists of 6 strands of 37 wires each wound around a single hemp core.

In general, the flexibility of the rope is increased by increasing the number of wires in the strands. The most flexible standard rope made consists of 6 61-wire strands and one hemp core. Other varieties comprise flattened strand ropes for haulage, hoisting, and transmission, non-spinning rope for the suspension of loads at the end of a single line, steel clad rope for severe conditions of service, guy and rigging ropes, and hawsers for towing or mooring.

**BREAKING STRENGTH OF WIRE ROPE.**—The various manufacturers have adopted standard figures for the strength of all sizes and qualities of wire rope. Formerly it was the custom to test the individual wires and to consider their combined strength as the strength of the rope as a whole. These strengths were greater than the actual strength obtained by breaking the finished rope. The figures given in the tables herewith represent actual breaks of the various ropes, and range from 95 to 80% or less of the combined strength of the single wires, depending on the construction.

**LAY OF WIRE ROPE.**—Regular lay of wire rope comprises wires in the strands laid to the left, the strands being laid to the right, known as right-lay rope; or wires laid to the right, and strands laid to the left, known as left-lay rope. Standard rope is made regular lay, right lay. In Lang lay rope the wires in the strands and the strands themselves are laid in the rope in the same direction, either right or left.

Lang lay rope is somewhat more flexible than ordinary rope, and as tne wires are laid more axially in the rope, longer surfaces are exposed to wear, and the endurance thereby is increased. Lang lay rope is used only to a limited extent in the U. S., largely where the ropes drag or are subjected to much friction. As Lang lay rope untwists more readily than regular lay, care must be exercised in its use. It is not as good an all-around rope as regular lay.

**SHEAVES AND DRUMS.**—Drums and sheaves of the largest practicable diameter are recommended in all wire rope installations. See Table 8. If possible, drums should be lagged, and where feasible, a grooved drum on hoists is more desirable than a flat drum. The grooves should give ample clearance between successive windings; thus a drum for $^3/_4$-in. rope should have the grooves at least $^7/_8$-in. apart on centers. The grooves should be made smooth in order not to cut the rope, and they should be of slightly larger radius than the rope in order to avoid wedging or pinching it. See Table 9. Overwinding, that is, the winding of the rope in more than one layer, is to be avoided, if possible, by making the drum large enough to take all the rope in a single layer. Overwinding will rapidly

**Table 8.—Diameter of Sheaves and Drums for Wire Rope**

Diam. of Sheave = (Diam. of Rope × $C$)

| Type of Rope | Values of $C$ | |
|---|---|---|
| | Average Recommended | Minimum |
| 6 × 7 | 72 | 42 |
| 18 × 7 non-spinning | 51 | 34 |
| 6 × 19 | 45 | 30 |
| 6 × 21 | 45 | 30 |
| 6 × 25 | 45 | 30 |
| 6 × 37 | 27 | 18 |
| 8 × 19 | 31 | 21 |

**Table 9.—Proper Clearances for Sheave Grooves**

(John A. Roebling's Sons Co., Trenton, N. J.)

| Normal Rope, Diam., in. | Minimum Clearance, in. | | Normal Rope Diameter, in. | Minimum Clearance, in. | | Normal Rope, Diam., in. | Minimum Clearance, in. | | Normal Rope Diameter, in. | Minimum Clearance, in. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Before Replacement or Remachining | New or Remachined | | Before Replacement or Remachining | New or Remachined | | Before Replacement or Remachining | New or Remachined | | Before Replacement or Remachining | New or Remachined |
| 1/4 | 1/64 | 1/32 | 9/16 | 1/32 | 1/16 | 1 1/8 | 1/32 | 1/16 | 1 3/4 | 3/64 | 1/8 |
| 5/16 | 1/64 | 1/32 | 5/8 | 1/32 | 1/16 | 1 1/4 | 1/32 | 1/16 | 1 7/8 | 3/64 | 1/8 |
| 3/8 | 1/32 | 1/16 | 3/4 | 1/32 | 1/16 | 1 3/8 | 1/32 | 1/8 | 2 | 3/64 | 1/8 |
| 7/16 | 1/32 | 1/16 | 7/8 | 1/32 | 1/16 | 1 1/2 | 1/32 | 1/8 | 2 1/8 | 3/64 | 1/8 |
| 1/2 | 1/32 | 1/16 | 1 | 1/32 | 1/16 | 1 5/8 | 1/32 | 1/8 | 2 1/4 | 3/64 | 1/8 |

destroy the rope, and the extra cost of the larger drum will be more than compensated by the greater life of the rope. The best possible alignment of sheaves and drums should be made to avoid undue wear on the sides of the sheaves and the rope. In general, the lead sheaves over which the rope runs from the drum should be aligned with the center of the drum, or if the drum is not entirely filled, with the center of the portion on which the rope is wound. The distance between the drum and lead sheave should be such as to cause an angle not exceeding 1° 30′ between the line from the center of the sheave to the center of the drum, and the line from the center of the sheave to the outer side of the drum. When the sheaves become worn, they should be replaced or the grooves turned before they are used with a new wire rope, otherwise the rope will not work properly. When changing an old rope for a new one, the sheaves should be examined, and if scored or cut from the old rope, they should be turned, relined or renewed, so that the new rope will not receive undue wear. For many purposes, particularly mine service, the grooves can advantageously be lined with well-seasoned, hardwood blocks set on end, which can be renewed when worn. Large sheaves, running at high velocity, should be lined with leather set on end, or with india-rubber. This is the practice for power transmission between distant points, where the rope frequently runs at a velocity of 4000 feet per minute.

**REVERSE BENDING.**—Reverse bending, that is, bending the wire rope first in one direction over sheaves and then in the opposite direction, is to be avoided wherever possible. This practice will wear out a rope more quickly than any other known method. A little care in design usually will eliminate all situations which call for reverse bending, and it is even desirable to change existing constructions if necessary to remove this condition. The expense of rope renewals will more than equal the cost of change, as a rule.

**HANDLING WIRE ROPE.**—Wire rope must not be coiled or uncoiled like hemp rope. When received in a coil it should be rolled on the ground like a hoop and straightened out before being put on the sheaves. If on a reel, it should be mounted on a spindle or a flat turntable and properly unwound. Kinking or untwisting must be avoided.

**PROTECTION AND LUBRICATION OF WIRE ROPE.**—Wire rope should be protected by a suitable lubricant, both externally and internally, to prevent rust and keep it pliable. The lubricant should be suited to the conditions under which the rope is used. It should not contain acids or other corrosive agents, and should readily penetrate the strands. It should be self spreading, adhesive, and water repelling. The higher the speed of the rope, the lower should be the viscosity of the lubricant. For stationary ropes for low speed work, as on shovels, cranes, or derricks, a heavy petroleum oil or grease, viscosity approximately 1000 sec. at 210° F., applied hot, is advised. For heavy haulage, as slope and logging ropes, a lubricant of viscosity of 500 to 800 sec. at 210° F. should be used. For high speed hoisting, as in mine hoists, a lubricant of viscosity of 400 to 500 sec. at 100° F. is recommended, with automatic lubrication. In all cases, the viscosity should be such that the lubricant will cover the rope, penetrate thoroughly, withstand pressure, but yet not so thin as to run off the rope. Crude oils should not be used. New lubricants only should be applied. Lubricants should be applied hot, as the penetration thereby is increased, and when cooled the lubricant exists as a plastic filler to lubricate the wires and to exclude moisture. The lubricant is best applied by passing the rope through a bath of heated oil. Placing a drum of heated oil over the rope and allowing the oil to flow on the rope is wasteful of oil. When hand lubrication is necessary a piece of sheepskin with the wool coat on the inside is recommended as a wiper, the wool being saturated with oil, and held around the rope as it passes through the wiper. While in storage, wire rope should be kept dry and protected from the weather, acid, and acid fumes. When a used rope is stored, it should be thoroughly cleaned before oiling and then treated as a new rope. For special cases, the manufacturers of wire rope should be consulted as to the type and method of lubrication.

**EXPOSURE TO HEAT.**—Where wire rope is exposed to intense heat, as in foundry or steel mill service, a soft iron often is substituted for the hemp core. Asbestos also sometimes is used, but it is not recommended.

## Varieties and Uses of Wire Rope

**GALVANIZED WIRE ROPE** is used for "standing" service, where no bending is required, as guys, ship's rigging, towing and mooring lines, etc. It is not adapted to hoisting service, as the galvanized coating is abraded and worn away by the sheaves, thus promoting corrosion. The more commonly used types of galvanized wire rope are described below.

**Galvanized Standing and Guy Ropes** are made both of iron and cast steel wire. The iron rope comprises 6 strands, 7 wires to the strand, with a hemp core. The cast steel rope comprises 6 strands with either 7 or 19 wires to the strand and a hemp core. Both types also are made with a wire strand center, which increases the breaking strength about 7.5%. These ropes also are known as yacht rigging rope. See Table 10.

**Table 10.—Galvanized Wire Standing and Guy Rope**
Also known as Yacht Rigging Rope
6 Strands, 7 Wires per Strand, 1 Hemp Core

| | | | Iron | | Cast Steel | | | | | Iron | | Cast Steel | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diameter, in. | Circumference, in. | Approximate weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | Circumference of Equal Strength Manila Rope | Breaking Strength, Tons (2000 lb.) | Circumference of Equal Strength Manila Rope | Diameter, in. | Circumference, in. | Approximate weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | Circumference of Equal Strength Manila Rope | Breaking Strength, Tons (2000 lb.) | Circumference of Equal Strength Manila Rope |
| 1 3/4 | 5 1/2 | 4.60 | 37.0 | 11 | ...... | ...... | 3/4 | 2 1/4 | 0.84 | 7.10 | 4 3/4 | 14.8 | 7 |
| 1 11/16 | 5 1/4 | 4.27 | 34.7 | 10 1/2 | ...... | ...... | 5/8 | 2 | .59 | 5.30 | 4 1/2 | 10.4 | 6 |
| 1 5/8 | 5 | 3.96 | 32.4 | 10 | ...... | ...... | 9/16 | 1 3/4 | .48 | 4.32 | 3 3/4 | 8.5 | 5 1/4 |
| 1 1/2 | 4 3/4 | 3.38 | 27.7 | 9 1/2 | ...... | ...... | 1/2 | 1 1/2 | .38 | 3.43 | 3 | 6.8 | 4 3/4 |
| 1 7/16 | 4 1/2 | 3.10 | 25.6 | 9 | ...... | ...... | 15/32 | 1 3/8 | .35 | ...... | ...... | 6 | 4 1/2 |
| 1 3/8 | 4 1/4 | 2.84 | 23.7 | 8 1/2 | ...... | ...... | 7/16 | 1 1/4 | .29 | 2.64 | 2 1/2 | 5.3 | 4 1/4 |
| 1 1/4 | 4 | 2.34 | 19.9 | 8 | 40.0 | 13 | 3/8 | 1 1/8 | .21 | 1.95 | 2 1/4 | 3.9 | 3 3/8 |
| 1 3/16 | 3 3/4 | 2.12 | 18.1 | 7 1/2 | 36.4 | 12 | 5/16 | 1 | .15 | 1.36 | 2 | 2.8 | 3 |
| 1 1/8 | 3 1/2 | 1.90 | 16.5 | 6 1/2 | 32.8 | 11 | 9/32* | 7/8 | .125 | 1.20 | 1 3/4 | ...... | ...... |
| 1 1/16 | 3 1/4 | 1.70 | 14.8 | 6 | 29.4 | 10 | 1/4* | 3/4 | .09 | .99 | 1 1/2 | ...... | ...... |
| | 3 | 1.50 | 13.2 | 5 3/4 | 26.1 | 9 | 7/32* | 5/8 | .063 | .79 | 1 1/4 | ...... | ...... |
| 7/8 | 2 3/4 | 1.15 | 10.2 | 5 1/4 | 20.0 | 8 1/2 | 3/16* | 1/2 | .04 | .61 | 1 1/8 | ...... | ...... |
| 13/16 | 2 1/2 | 0.99 | 8.86 | 5 | 17.3 | 8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

* 5 strands, 7 wires per strand.

**Table 11.—Galvanized Steel Bridge Strand**

| Diameter, in. | 1 1/2 | 1 3/8 | 1 1/4 | 1 1/8 | 1 |
|---|---|---|---|---|---|
| Weight per ft., lb. | 4.70 | 3.90 | 3.38 | 2.82 | 2.01 |
| Area, sq. in. | 1.337 | 1.105 | 0.956 | 0.798 | 0.569 |
| Minimum ultimate strength, lb. | 270,000 | 223,000 | 194,000 | 161,000 | 115,000 |

Minimum modulus of elasticity up to 50% of ultimate strength, 24,000,000.

**Table 12.—Galvanized Plow Steel Bridge Cable**
6 Strands, 7, 19, or 37 Wires to the Strand

| Diameter, in. | Approx. Circumference, in. | Approx. Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | Diameter, in. | Approx. Circumference, in. | Approx. Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) |
|---|---|---|---|---|---|---|---|
| 3 | 9 3/8 | 15.75 | 360 | 1 7/8 | 5 7/8 | 6.25 | 144 |
| 2 3/4 | 8 5/8 | 13.25 | 310 | 1 3/4 | 5 1/2 | 5.36 | 124 |
| 2 5/8 | 8 1/4 | 12.10 | 283 | 1 5/8 | 5 1/8 | 4.62 | 106 |
| 2 1/2 | 7 7/8 | 11.00 | 256 | 1 1/2 | 4 3/4 | 3.80 | 90 |
| 2 3/8 | 7 1/2 | 9.90 | 232 | 1 3/8 | 4 3/8 | 3.30 | 75 |
| 2 1/4 | 7 1/8 | 8.85 | 208 | 1 1/4 | 3 7/8 | 2.70 | 62 |
| 2 1/8 | 6 5/8 | 7.90 | 185 | 1 1/8 | 3 1/2 | 2.20 | 54 |
| 2 | 6 1/4 | 7.00 | 164 | 1 | 3 1/8 | 1.75 | 42 |

**Galvanized Steel Strand** is made with 7 or 19 steel wires twisted into a single strand. The standard grade is used for guys for smokestacks and poles and for supporting trolley wires and operating railroad signals. The Siemens-Martin high strength, and extra high strength grades combine high strength with toughness and light weight. They are used to guy electric railway, telegraph and telephone poles, to support trolley and messenger wires, and for purposes where high strength is required. See Table 13.

**Table 13.—Galvanized Steel Strand**

| Diameter of Rope, in. | No. of Wires | Approximate Size of Wire, in. | Approximate Weight per 1000 ft., lb. | Breaking Strength, lb. Standard | Siemens-Martin | High Strength | Extra High Strength |
|---|---|---|---|---|---|---|---|
| 5/64 | 7 | 0.026 | 13.0 | 300 | ...... | ...... | ...... |
| 3/32 | 7 | .031 | 20.0 | 400 | ...... | ...... | ...... |
| 7/64 | 7 | .035 | 25.0 | 450 | ...... | ...... | ...... |
| 1/8 | 7 | .041 | 31.8 | 540 | 910 | 1,330 | 1,830 |
| 9/64 | 7 | .047 | 40.0 | 700 | ...... | ...... | ...... |
| 5/32 | 7 | .052 | 51.3 | 870 | ...... | ...... | ...... |
| 3/16 | 7 | .062 | 72.9 | 1,150 | 1,900 | 2,850 | 3,990 |
| 7/32 | 7 | .072 | 98.3 | 1,540 | ...... | ...... | ...... |
| 1/4 | 7 | .080 | 121.0 | 1,900 | 3,150 | 4,750 | 6,650 |
| 9/32 | 7 | .093 | 164.0 | 2,570 | 4,250 | 6,400 | 8,950 |
| 5/16 | 7 | .104 | 205.0 | 3,200 | 5,350 | 8,000 | 11,200 |
| 3/8 | 7 | .120 | 273.0 | 4,250 | 6,950 | 10,800 | 15,400 |
| 3/8 | 19 | ..... | 288 | ...... | 6,800 | 11,500 | 17,250 |
| 7/16 | 7 | 0.145 | 399 | 5,700 | 9,350 | 14,500 | 20,800 |
| 7/16 | 19 | ..... | 395 | ...... | 9,000 | 15,000 | 22,500 |
| 1/2 | 7 | 0.165 | 517 | 7,400 | 12,100 | 18,800 | 26,900 |
| 1/2 | 19 | .100 | 504 | 7,620 | 12,700 | 19,100 | 26,700 |
| 9/16 | 7 | .188 | 671 | 9,600 | 15,700 | 24,500 | 35,000 |
| 9/16 | 19 | .110 | 637 | 9,640 | 16,100 | 24,100 | 33,700 |
| 5/8 | 7 | .207 | 813 | 11,600 | 19,100 | 29,600 | 42,400 |
| 5/8 | 19 | .125 | 796 | 11,000 | 18,100 | 28,100 | 40,200 |
| 3/4 | 7 | .250 | 1,200 | 16,700 | ...... | ...... | ...... |
| 3/4 | 19 | .150 | 1,155 | 16,000 | 26,200 | 40,800 | 58,300 |
| 7/8 | 19 | .175 | 1,581 | 21,900 | 35,900 | 55,800 | 79,700 |
| 1 | 19 | .200 | 2,073 | 28,700 | 47,100 | 73,200 | 104,500 |

**Table 14.—Airplane Strand and Cord**

| Diam., in. | 7-Wire and 19-Wire |  | 6-Strand, 7-Wire, Cotton Center |  | 7-Strand, 7-Wire |  | 7-Strand, 19-Wire |  |
|---|---|---|---|---|---|---|---|---|
|  | Approx. Weight, lb. per 1000 ft. | Breaking Strength, lb. | Approx. Weight, lb. per 1000 ft. | Breaking Strength, lb. | Approx. Weight, lb. per 1000 ft. | Breaking Strength, lb. | Approx. Weight, lb. per 1000 ft. | Breaking Strength, lb. |
| 1/32 | 2.2* | 185 | ...... | ...... | ...... | ...... | ...... | ...... |
| 1/16 | 8.8† | 500 | 6.2 | 400 | 6.9 | 480 | ...... | ...... |
| 5/64 | 14.0† | 780 | 10.0 | 480 | 10.9 | 550 | ...... | ...... |
| 3/32 | 20.0† | 1,100 | 14.0 | 780 | 16.0 | 920 | ...... | ...... |
| 7/64 | 28.0† | 1,600 | 19.0 | 830 | ...... | ...... | ...... | ...... |
| 1/8 | 35.0† | 2,100 | 24.0 | 1,150 | 27.0 | 1350 | 27 | 2,000 |
| 5/32 | 57.0† | 3,200 | 38.0 | 2,200 | 43.0 | 2600 | 42 | 2,800 |
| 3/16 | 75.0† | 4,600 | 55.0 | 2,750 | 61.0 | 3200 | 60 | 4,200 |
| 7/32 | 106.0† | 6,100 | 73.0 | 4,000 | 81.0 | 4600 | 79 | 5,600 |
| 1/4 | 136.0† | 8,000 | 96.0 | 5,000 | 106.0 | 5800 | 103 | 7,000 |
| 9/32 | ...... | ...... | 121.0 | 6,200 | ...... | ...... | 131 | 8,000 |
| 5/16 | 214.0† | 12,500 | 148.0 | 7,900 | 166.0 | 9200 | 161 | 9,800 |
| 11/32 | ...... | ...... | ...... | ...... | ...... | ...... | 195 | 12,500 |
| 3/8 | ...... | ...... | 214.0 | 11,800 | ...... | ...... | 232 | 14,400 |
| 7/16 | ...... | ...... | 300.0 | 14,200 | ...... | ...... | ...... | ...... |
| 1/2 | ...... | ...... | 391.0 | 20,000 | ...... | ...... | ...... | ...... |

* 7 wire. † 19 wire.

**TRANSMISSION OR HAULAGE ROPE.**—Usually made of 6 7-wire strands and one hemp core, in all five grades. It is made of large wires and is the least flexible of any standard rope. Cast steel rope is particularly adapted to mine haulage work, including tail rope and endless haulage systems, gravity hoists, and coal and ore dock haulage, roads, operating small grip cars, etc. The sizes 3/8 to 5/8 in. inclusive, are used for sand lines in oil wells, and from 5/8 to 1 in. for oil-well drilling. In general, it can be used for severe service, and where the flexibility required is a minimum. The plow-steel rope is advised for situations similar to those for which the cast steel ropes are used, but where it is necessary to secure increased strength, without altering the working conditions. The wires are harder and capable of standing greater wear than any of the foregoing ropes. Monitor plow-steel rope is the strongest and stiffest of all and is used for work demanding the greatest strength and lightest rope possible. Sheaves for this rope should, if possible, be somewhat larger than for other grades. For working loads, strength, etc., of these ropes, see Tables 15 and 16.

**Standard Hoisting Rope.**—Composed of 6 19-wire strands and a hemp core; made in the following grades: Iron, cast steel, mild plow steel, plow steel and Monitor plow steel. The wires are smaller than those in transmission ropes of the same size, and the rope is more flexible. It will not stand as much abrasion as transmission rope. The hemp-center rope is very pliable and will wind on moderate size drums and sheaves without injury. If possible, drums and sheaves should be larger than shown in Table 8. The cast steel, mild plow steel, plow steel and Monitor plow steel ropes are adapted to mine hoisting, logging, quarrying, elevators, derricks, hay presses, dredges, cableways, inclined planes, coal hoists, conveyors, ballast unloaders, skip hoists, and similar applications. See Table 16.

**Extra Flexible Hoisting Rope.**—Consists of 8 19-wire strands and one hemp core. The greater flexibility permits its use on smaller sheaves and drums, such as usually are found on derricks. It is not advisable to use it where there is much overwinding, as it will flatten much more quickly than the 6 × 19 standard rope. It is made in the five grades of iron, cast steel, mild plow steel, plow steel, and Monitor plow steel. Its uses are the same as those of standard hoisting rope. See Table 17.

**Special Flexible Hoisting Rope.**—Consists of 6 37-wire strands and one hemp core. It is extremely flexible, and is especially adapted to high-speed service on cranes, or where sheaves are small. It is made in the grades cast steel, mild plow steel, plow steel, and Monitor plow steel. It will not stand as much abrasion as the 6 × 19 rope, but it is particularly efficient, as over 50% of the wires are in the inner layers and are protected from abrasion. The cast steel ropes are used for general hoisting work where the sheaves are small, while the plow steel varieties are recommended for crane service. The Monitor plow steel rope is largely used on dredges for both main and spud ropes. See Table 18.

**Table 15.—Transmission or Haulage Rope**

6 Strands, 7 Wires per Strand, 1 Hemp Core

| Diameter, in. | Approximate Circumference, in. | Approximate Weight, lb. per ft. | Breaking Strength, Tons (2000 lb.) | | | | | Diameter, in. | Approximate Circumference, in. | Approximate Weight, lb. per ft. | Breaking Strength, Tons (2000 lb.) | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel | | | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel |
| 1 1/2 | 4 3/4 | 3.38 | 29.7 | 62.5 | 68.7 | 75.0 | 86.5 | 9/16 | 1 3/4 | 0.48 | 4.49 | 9.4 | 10.3 | 11.3 | 13.0 |
| 1 3/8 | 4 1/4 | 2.84 | 25.2 | 53.0 | 58.2 | 63.5 | 73.5 | 1/2 | 1 1/2 | .38 | 3.57 | 7.5 | 8.2 | 9.0 | 10.3 |
| 1 1/4 | 4 | 2.34 | 21.0 | 44.5 | 48.7 | 53.0 | 61.0 | 7/16 | 1 1/4 | .29 | 2.76 | 5.8 | 6.3 | 6.9 | 7.9 |
| 1 1/8 | 3 1/2 | 1.90 | 17.2 | 36.4 | 40.0 | 43.6 | 50.0 | 3/8 | 1 1/8 | .21 | 2.05 | 4.3 | 4.7 | 5.15 | 5.9 |
| 1 | 3 | 1.50 | 13.7 | 29.0 | 31.9 | 34.8 | 40.0 | 5/16 | 1 | .15 | 1.43 | 3.1 | 3.35 | 3.65 | 4.2 |
| 7/8 | 2 3/4 | 1.15 | 10.5 | 22.4 | 24.6 | 26.8 | 30.8 | 9/32 | 7/8 | .12 | 1.17 | 2.52 | 2.72 | 2.95 | 3.4 |
| 3/4 | 2 1/4 | 0.84 | 7.86 | 16.5 | 18.1 | 19.8 | 22.8 | 1/4 | 3/4 | .094 | ...... | 2.00 | 2.15 | 2.35 | 2.7 |
| 5/8 | 2 | .59 | 5.52 | 11.5 | 12.6 | 13.8 | 16.0 | . | .... | .... | ..... | ...... | ...... | ...... | ...... |

**Table 16.—Standard Hoisting Rope**

6 Strands, 19 Wires per Strand, 1 Hemp Core

| Diameter, in. | Approximate Circumference, in. | Approximate Weight, lb. per ft. | Breaking Strength Tons (2000 lb.) | | | | | Diameter, in. | Approximate Circumference, in. | Approximate Weight, lb. per ft. | Breaking Strength Tons (2000 lb.) | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel | | | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel |
| 2 3/4 | 8 5/8 | 12.10 | 95.0 | 212.0 | 235.0 | 256.0 | 294.0 | 1 1/8 | 3 1/2 | 2.03 | 17.2 | 37.0 | 41.5 | 46.0 | 53.0 |
| 2 1/2 | 7 7/8 | 10.00 | 79.1 | 176.0 | 195.0 | 214.0 | 246.0 | 1 | 3 | 1.60 | 13.7 | 29.5 | 33.0 | 36.5 | 42.0 |
| 2 1/4 | 7 1/8 | 8.10 | 64.8 | 144.0 | 160.0 | 176.0 | 202.0 | 7/8 | 2 3/4 | 1.23 | 10.6 | 22.8 | 25.4 | 28.0 | 32.2 |
| 2 1/8 | 6 5/8 | 7.22 | ...... | 128.0 | 143.0 | 157.0 | 181.0 | 3/4 | 2 1/4 | 0.90 | 7.86 | 16.8 | 18.7 | 20.6 | 23.7 |
| 2 | 6 1/4 | 6.40 | 51.8 | 114.0 | 127.0 | 140.0 | 161.0 | 5/8 | 2 | .63 | 5.52 | 11.8 | 13.1 | 14.4 | 16.6 |
| 1 7/8 | 5 3/4 | 5.63 | 45.8 | 100.0 | 112.0 | 123.0 | 142.0 | 9/16 | 1 3/4 | .51 | 4.49 | 9.6 | 10.6 | 11.7 | 13.5 |
| 1 3/4 | 5 1/2 | 4.90 | 40.1 | 88.0 | 98.0 | 108.0 | 124.0 | 1/2 | 1 1/2 | .40 | 3.57 | 7.7 | 8.5 | 9.4 | 10.8 |
| 1 5/8 | 5 | 4.23 | 34.8 | 76.0 | 85.0 | 94.0 | 108.0 | 7/16 | 1 1/4 | .31 | 2.76 | 6.0 | 6.6 | 7.3 | 8.4 |
| 1 1/2 | 4 3/4 | 3.60 | 29.7 | 65.0 | 72.5 | 80.5 | 92.5 | 3/8 | 1 1/8 | .23 | 2.05 | 4.5 | 5.0 | 5.5 | 6.3 |
| 1 3/8 | 4 1/4 | 3.03 | 25.2 | 55.0 | 61.5 | 68.0 | 78.5 | 5/16 | 1 | .16 | 1.43 | 3.2 | 3.5 | 3.9 | 4.5 |
| 1 1/4 | 4 | 2.50 | 21.0 | 46.0 | 51.0 | 56.5 | 65.0 | 1/4 | 3/4 | .10 | 0.97 | 2.1 | 2.3 | 2.5 | 2.9 |

### Table 17.—Extra Flexible Hoisting Rope

8 Strands, 19 Wires per Strand, 1 Hemp Core

| Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | | | Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel | | | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel |
| 1 1/2 | 4 3/4 | 3.26 | ...... | 57.5 | 63.3 | 69.0 | 79.5 | 5/8 | 2 | 0.57 | 4.86 | 10.4 | 11.4 | 12.5 | 14.4 |
| 1 3/8 | 4 1/4 | 2.74 | ...... | 48.6 | 53.4 | 58.3 | 67.0 | 9/16 | 1 3/4 | .46 | 4.06 | 8.5 | 9.4 | 10.3 | 11.8 |
| 1 1/4 | 4 | 2.27 | ...... | 40.4 | 44.4 | 48.4 | 55.7 | 1/2 | 1 1/2 | .36 | 3.28 | 6.8 | 7.5 | 8.2 | 9.5 |
| 1 1/8 | 3 1/2 | 1.84 | ...... | 32.8 | 36.0 | 39.4 | 45.2 | 7/16 | 1 1/4 | .28 | 2.4 | 5.3 | 5.8 | 6.3 | 7.3 |
| 1 | 3 | 1.45 | 12.2 | 26.0 | 28.6 | 31.2 | 35.8 | 3/8 | 1 1/8 | .20 | 1.6 | 3.94 | 4.33 | 4.7 | 5.45 |
| 7/8 | 2 3/4 | 1.11 | 9.4 | 20.0 | 22.0 | 24.0 | 27.6 | 5/16 | 1 | .14 | 1.1 | 2.79 | 3.07 | 3.34 | 3.85 |
| 3/4 | 2 1/4 | 0.82 | 6.95 | 14.8 | 16.3 | 17.8 | 20.5 | 1/4 | 3/4 | .09 | 0.8 | 1.79 | 1.97 | 2.15 | 2.47 |

### Table 18.—Special Flexible Steel Hoisting Rope

6 Strands, 37 Wires per Strand, 1 Hemp Core

| Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | | Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel | | | | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel |
| 3 1/2 | 11 | 19.00 | 323.0 | 357.0 | 392.0 | 451.0 | 1 3/8 | 4 1/4 | 2.93 | 53.5 | 59.0 | 64.6 | 74.3 |
| 3 1/4 | 10 1/4 | 16.37 | 281.0 | 311.0 | 341.0 | 392.0 | 1 1/4 | 4 | 2.42 | 44.3 | 48.9 | 53.5 | 61.5 |
| 3 | 9 1/2 | 13.95 | 241.0 | 267.0 | 293.0 | 337.0 | 1 1/8 | 3 1/2 | 1.96 | 36.0 | 39.7 | 43.5 | 49.9 |
| 2 3/4 | 8 5/8 | 11.72 | 204.0 | 226.0 | 248.0 | 285.0 | 1 | 3 | 1.55 | 28.6 | 31.5 | 34.4 | 39.5 |
| 2 1/2 | 7 7/8 | 9.69 | 170.0 | 188.0 | 206.0 | 237.0 | 7/8 | 2 3/4 | 1.19 | 22.0 | 24.2 | 26.5 | 30.5 |
| 2 3/8 | 7 1/2 | 8.24 | ...... | ...... | 187.0 | 215.0 | 3/4 | 2 1/4 | 0.87 | 16.4 | 18.1 | 19.8 | 22.8 |
| 2 1/4 | 7 1/8 | 7.85 | 139.0 | 153.0 | 168.0 | 194.0 | 5/8 | 2 | .61 | 11.6 | 12.8 | 14.0 | 16.1 |
| 2 1/8 | 6 3/4 | 7.00 | 125.0 | 138.0 | 151.0 | 174.0 | 9/16 | 1 3/4 | .49 | 9.5 | 10.5 | 11.5 | 13.2 |
| 2 | 6 1/4 | 6.20 | 111.0 | 123.0 | 135.0 | 155.0 | 1/2 | 1 1/2 | .39 | 7.7 | 8.4 | 9.2 | 10.6 |
| 1 7/8 | 5 3/4 | 5.45 | 98.0 | 108.5 | 119.0 | 137.0 | 7/16 | 1 1/4 | .30 | 6.0 | 6.6 | 7.2 | 8.3 |
| 1 3/4 | 5 1/2 | 4.75 | 86.0 | 95.0 | 104.0 | 119.5 | 3/8 | 1 1/8 | .22 | 4.4 | 4.9 | 5.3 | 6.1 |
| 1 5/8 | 5 | 4.09 | 74.3 | 82.0 | 89.8 | 103.3 | 5/16 | 1 | .15 | 3.1 | 3.4 | 3.8 | 4.4 |
| 1 1/2 | 4 3/4 | 3.49 | 63.5 | 70.0 | 76.7 | 88.2 | 1/4 | 3/4 | .10 | 2.0 | 2.2 | 2.4 | 2.8 |

### Table 19.—Flattened Strand Haulage or Transmission Rope

| Diam. of Rope, in. | Type C | | | | Type D | | | | Type E | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Weight per foot, lb. | Breaking Strength, Tons (2000 lb.) | | | Weight per foot, lb. | Breaking Strength, Tons (2000 lb.) | | | Weight per foot, lb. | Breaking Strength, Tons (2000lb.) |
| | | Cast Steel | Mild Plow Steel | Monitor Plow Steel | | Cast Steel | Mild Plow Steel | Monitor Plow Steel | | Cast Steel |
| 1 1/2 | 3.49 | 54.0 | 60.0 | 75.0 | 4.05 | 67.0 | 74.0 | 93.0 | 3.49 | 54.0 |
| 1 3/8 | 2.93 | 46.0 | 50.0 | 63.0 | 3.40 | 57.0 | 62.0 | 79.0 | 2.93 | 46.0 |
| 1 1/4 | 2.42 | 39.0 | 42.0 | 52.0 | 2.81 | 48.0 | 52.0 | 65.0 | 2.42 | 39.0 |
| 1 1/8 | 1.96 | 31.0 | 35.0 | 43.0 | 2.28 | 39.0 | 43.0 | 54.0 | 1.96 | 31.0 |
| 1 | 1.55 | 24.8 | 27.5 | 34.5 | 1.80 | 31.0 | 34.0 | 43.0 | 1.55 | 24.8 |
| 7/8 | 1.19 | 19.2 | 21.2 | 26.6 | 1.39 | 24.0 | 26.5 | 33.2 | 1.19 | 19.2 |
| 3/4 | 0.87 | 14.3 | 15.6 | 19.7 | 1.01 | 17.8 | 19.5 | 24.6 | 0.87 | 14.3 |
| 5/8 | .61 | 10.0 | 10.8 | 13.8 | 0.70 | 12.4 | 13.5 | 17.2 | .61 | 10.0 |
| 1/2 | .39 | 6.5 | 7.1 | 8.8 | .45 | 8.1 | 8.8 | 11.0 | .39 | 6.5 |
| 3/8 | .22 | 3.7 | 4.0 | 5.1 | .25 | 4.6 | 5.0 | 6.35 | .22 | 3.7 |

**FLATTENED STRAND ROPE.**—Flattened strand ropes are used where an increased wearing surface is desired above that obtained with a round strand rope. The wearing surface in contact with sheave or drum is from 100 to 150% greater than in round-strand ropes.

**Flattened-strand Haulage or Transmission Rope** is made in three types, each with a hemp core. Type C has 5 9-wire strands, the center wire being of elliptical cross-section; type D has 6 8-wire strands, the center wire being of triangular cross-section; type E has 5 11-wire strands, the three center wires being of smaller diameter than the others and laid alongside of each other in the same plane. These ropes also are made with a wire strand or wire rope center, which increases the breaking strength about 7.5%. See Table 19.

**Flattened-strand Hoisting Rope** is made in two types each with a hemp core. Type A consists of 5 strands, 28 wires per strand, the center wire being of elliptical cross-section, and the inner layer of surrounding wires being of smaller diameter than the outer. Type B consists of 6 strands, 25 wires per strand, the center wire being of triangular cross-section, and the inner layer of surrounding wires being of smaller diameter than the outer layer. Type A sometimes is used on elevators and on some types of coal handling equipment. It is not adapted to general hoisting, especially when used for heavy service over flat sheaves or drums, or where overwinding occurs. Type B is a very rigid rope that holds its shape well under stress. It easily can be passed over sheaves of moderate size. Both types are made in the iron, cast steel, mild plow steel, plow steel and Monitor plow steel grades. When made with a steel rope or steel strand center, the breaking strength is increased about 7.5%. See Table 20.

**Non-spinning Hoisting Rope.**—Comprises 18 7-wire strands and one hemp core, 6 strands, Lang lay, being laid around the core to the left, and 12 strands, regular lay being laid to the right around them. A free object suspended from the end of a rope of this character will not rotate and endanger the lives of persons below it. Furthermore, the attention required to handle and guide the load is decreased. This rope is recommended for back haul or single-line derricks, and for shaft sinking and mine hoisting, where the bucket swings without guides. This rope works best where it does not overwind on the

**Table 20.—Flattened Strand Hoisting Rope**

| Diam. of Rope, in. | Type A | | | | | | Type B | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Weight per foot, lb. | Breaking Strength, Tons (2000 lb.) | | | | | Weight per foot, lb. | Breaking Strength, Tons (2000 lb.) | | | |
| | | Iron | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel | | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel |
| 2 3/4 | ...... | ...... | ...... | ...... | ...... | ...... | 13.6 | 233.0 | 257.0 | ...... | 323.0 |
| 2 1/2 | ...... | ...... | ...... | ...... | ...... | ...... | 11.2 | 193.0 | 214.0 | ...... | 270.0 |
| 2 1/4 | ...... | ...... | ...... | ...... | ...... | ...... | 9.1 | 158.0 | 176.0 | 193.0 | 222.0 |
| 2 | ...... | ...... | ...... | ...... | ...... | ...... | 7.2 | 125.0 | 139.0 | 154.0 | 177.0 |
| 1 3/4 | ...... | ...... | ...... | ...... | ...... | ...... | 5.51 | 96.0 | 107.0 | 118.0 | 136.0 |
| 1 5/8 | ...... | ...... | ...... | ...... | ...... | ...... | 4.75 | 83.0 | 93.0 | 103.0 | 118.0 |
| 1 1/2 | 3.49 | ...... | 53.0 | 64.0 | 70.0 | 81.0 | 4.05 | 71.0 | 79.0 | 88.0 | 101.0 |
| 1 3/8 | 2.93 | ...... | 49.0 | 54.0 | 60.0 | 69.0 | 3.40 | 60.0 | 67.0 | 74.0 | 86.0 |
| 1 1/4 | 2.42 | ...... | 41.0 | 45.0 | 50.0 | 57.0 | 2.81 | 50.5 | 56.0 | 62.0 | 71.5 |
| 1 1/8 | 1.96 | ...... | 33.0 | 37.0 | 41.0 | 47.0 | 2.28 | 40.0 | 45.5 | 50.5 | 58.0 |
| 1 | 1.55 | ...... | 26.0 | 29.0 | 32.0 | 37.0 | 1.80 | 32.0 | 36.0 | 40.0 | 46.0 |
| 7/8 | 1.19 | ...... | 20.1 | 22.4 | 24.8 | 28.5 | 1.39 | 25.0 | 27.9 | 30.0 | 35.4 |
| 3/4 | 0.87 | 7.1 | 14.8 | 16.5 | 18.2 | 20.9 | 1.01 | 18.4 | 20.5 | 22.6 | 26.0 |
| 5/8 | .61 | 5.0 | 10.4 | 11.6 | 12.7 | 14.6 | 0.70 | 12.9 | 14.4 | 15.8 | 18.2 |
| 9/16 | .49 | 3.25 | 8.5 | 9.4 | 10.3 | 11.9 | .57 | 10.5 | 11.6 | 12.8 | 14.8 |
| 1/2 | .39 | 1.85 | 6.8 | 7.5 | 8.2 | 9.5 | .45 | 8.4 | 9.3 | 10.3 | 11.8 |
| 3/8 | .22 | ...... | 4.0 | 4.4 | 4.8 | 5.6 | .25 | 4.9 | 5.5 | 6.0 | 6.9 |

**Table 21.—Non-spinning Hoisting Rope**

18 Strands, 7 Wires per Strand, 1 Hemp Core

| Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | | Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel | | | | Cast Steel | Mild Plow Steel | Plow Steel | Monitor Plow Steel |
| 1 3/4 | 5 1/2 | 5.3 | ...... | ...... | ...... | 115.0 | 7/8 | 2 3/4 | 1.32 | 21.1 | 23.5 | 25.9 | 29.8 |
| 1 5/8 | 5 | 4.57 | ...... | ...... | ...... | 100.0 | 3/4 | 2 1/4 | 0.97 | 15.6 | 17.3 | 19.0 | 21.9 |
| 1 1/2 | 4 3/4 | 3.89 | ...... | ...... | ...... | 85.5 | 5/8 | 2 | .68 | 10.9 | 12.1 | 13.3 | 15.3 |
| 1 3/8 | 4 1/4 | 3.27 | ...... | ...... | ...... | 72.5 | 9/16 | 1 3/4 | .55 | 8.9 | 9.8 | 10.8 | 12.5 |
| 1 1/4 | 4 | 2.70 | 42.5 | 47.2 | 52.3 | 60.0 | 1/2 | 1 1/2 | .43 | 7.1 | 7.9 | 8.7 | 10.0 |
| 1 1/8 | 3 1/2 | 2.19 | 34.2 | 38.4 | 42.5 | 49.0 | 7/16 | 1 1/4 | .33 | 5.5 | 6.1 | 6.7 | 7.8 |
| 1 | 3 | 1.73 | 27.3 | 30.5 | 33.8 | 38.8 | 3/8 | 1 1/8 | .24 | 4.1 | 4.6 | 5.1 | 5.8 |

drum. The best fastening is an open or closed socket, but the wire rope makers recommend that fastenings be attached at the factory. This rope should not be as heavily loaded as ordinary hoisting rope. It is made in the grades of cast steel, mild plow steel, plow-steel, and Monitor plow-steel. See Table 21.

**ELEVATOR ROPE.**—Table 22 gives the characteristics of elevator hoisting rope, both standard and Seale construction. Seale construction consists of a single center wire, surrounded by a circle of wires of the same diameter, with a circle of smaller filler wires interposed between the center and outer wires.

**MARLIN-CLAD ROPE.**—This is a round strand rope, each strand being given an external serving of tarred marlin. (See Table 23). It is particularly adapted for ship rigging, cargo hoists, power transmission (see page 24–53), grain shovels, etc. It is considerably stronger than manila rope of equal diameter, a 1-in. plow-steel marlin-clad rope having a strength of 26,400 lb., while that of a 1-in. manila rope is 7500 lb. Marlin-clad rope weighs approximately 30% less than manila rope of the same strength.

**Marlin-clad Hawser Rope** consists of 5 separate ropes around a hemp core. Each rope comprises 5 strands, 7 wires per strand and one hemp core, covered with an external serving of tarred marlin. It is made in plow steel, mild plow steel and cast steel grades. See Table 24.

**STEEL-CLAD HOISTING ROPE.**— The regular grades of hoisting ropes, as well as the special flexible, are made, if desired, with a flat strip of steel wound spirally around each strand. These strips give additional wearing surface without sacrificing the flex-

**Table 22.—Elevator Rope**

| Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | Seale Patent Rope | | Diameter, in. | Approximate Circumference, in. | Approximate Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | Seale Patent Rope | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Iron | Traction Steel | Approx. Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | | Iron | Traction Steel | Approx. Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) |
| 1 | 3 | 1.60 | 13.7 | 27.0 | 1.45 | 25.0 | 7/16 | 1 1/4 | 0.31 | 2.76 | 5.5 | 0.28 | 5.0 |
| 7/8 | 2 3/4 | 1.23 | 10.6 | 21.0 | 1.11 | 19.0 | 3/8 | 1 1/8 | .23 | 2.05 | 4.1 | .20 | 3.7 |
| 3/4 | 2 1/4 | 0.90 | 7.86 | 16.0 | 0.82 | 14.0 | 5/16 | 1 | .16 | 1.43 | ...... | ...... | ...... |
| 5/8 | 2 | .63 | 5.52 | 11.5 | .57 | 10.0 | 1/4 | 3/4 | .10 | 0.97 | ...... | ...... | ...... |
| 9/16 | 1 3/4 | .51 | 4.49 | 9.25 | .46 | 8.0 | 3/16 | 9/16 | .06 | .55 | ...... | ...... | ...... |
| 1/2 | 1 1/2 | .40 | 3.57 | 7.25 | .36 | 6.5 | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

**Table 23.—Marlin-clad Hoisting Rope**

5 19-wire Strands, 1 Hemp Core. Each strand served with Tarred Marlin

| Diam. before Serving, in. | Approx. Diam. after Serving, in. | Approx. Circum. after Serving, in. | Approx. Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | | Diam. before Serving, in. | Approx. Diam. after Serving, in. | Approx. Circum. after Serving, in. | Approx. Weight per ft., lb. | Breaking Strength, Tons (2000 lb.) | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Cast Steel | Mild Plow Steel | Plow Steel | | | | | Cast Steel | Mild Plow Steel | Plow Steel |
| 1 3/4 | 2 1/8 | 6 5/8 | 5.00 | 81.0 | 90.0 | 99.0 | 3/4 | 1 1/8 | 3 1/2 | 1.10 | 15.4 | 17.1 | 18.9 |
| 1 5/8 | 2 | 6 1/4 | 4.29 | 70.0 | 78.0 | 86.0 | 5/8 | 1 | 3 1/8 | 0.81 | 10.8 | 12.0 | 13.2 |
| 1 1/2 | 1 7/8 | 5 7/8 | 3.69 | 60.0 | 66.5 | 74.0 | 9/16 | 7/8 | 2 3/4 | .62 | 8.8 | 9.7 | 10.7 |
| 1 3/8 | 1 3/4 | 5 1/2 | 3.14 | 50.5 | 56.5 | 62.5 | 1/2 | 13/16 | 2 1/2 | .51 | 7.1 | 7.8 | 8.6 |
| 1 1/4 | 1 5/8 | 5 1/8 | 2.58 | 42.0 | 47.0 | 52.0 | 7/16 | 3/4 | 2 3/8 | .42 | 5.5 | 6.1 | 6.7 |
| 1 1/8 | 1 1/2 | 4 3/4 | 2.12 | 34.0 | 38.0 | 42.0 | 3/8 | 11/16 | 2 1/8 | .36 | 4.1 | 4.6 | 5.0 |
| 1 | 1 3/8 | 4 5/16 | 1.70 | 27.0 | 30.3 | 33.5 | 5/16 | 5/8 | 2 | .28 | 2.9 | 3.2 | 3.6 |
| 7/8 | 1 1/4 | 3 7/8 | 1.32 | 21.0 | 23.3 | 25.7 | 1/4 | 9/16 | 1 3/4 | .21 | 1.9 | 2.1 | 2.3 |

**Table 24.—Marlin-clad Hawser Rope**

| | | | | |
|---|---|---|---|---|
| Diameter, each of 5 component ropes, in. | 5/8 | 1/2 | 7/16 | 3/8 |
| Approx. outside diam. of hawser, in., after serving | 2 5/8 | 2 | 1 7/8 | 1 3/4 |
| Approx. circumference, in., after serving | 8 1/4 | 6 1/4 | 5 7/8 | 5 1/2 |
| Approx. weight per foot, lb. | 3.80 | 2.60 | 2.30 | 2.10 |
| Breaking strength, tons (2000 lb.) Plow steel | 57.5 | 37.5 | 28.7 | 21.4 |
| Mild plow steel | 52.5 | 34.2 | 26.2 | 19.6 |
| Cast steel | 47.9 | 31.2 | 24.2 | 17.9 |

ibility. When the flat winding is worn through, a complete rope remains with unimpaired strength. These ropes are designed for severe conditions of service, and an additional service of 50 to 100% over that of the unprotected rope frequently is obtained. Tables 15 to 18 may be used for the strength of steel-clad rope, by referring to the diameter of the rope, as it would be were no wrapping applied. The standard steel-clad hoisting rope is made in sizes (diam. of bare rope) of 1/2 to 2 in. See Table 16. The special flexible steel-clad rope is made in sizes (diam. of bare rope) of 7/8 to 2 1/4 in. See Table 18. The steel wrapping is not considered as adding any strength to the rope but merely serving to increase its life. This rope is made in two grades of plow steel and Monitor plow steel.

**FLAT ROPE.**—Flat rope consists of a number of "flat-rope" strands, twisted alternately right and left, placed side by side and sewed with soft Swedish iron or steel wire, to form a flat rope of the desired width and thickness. The soft sewing wires wear much quicker than the rope wire, and have to be replaced from time to time, at which time worn strands can also be renewed. Flat rope is used principally for hoisting heavy loads out of deep shafts, it requiring a reel but little larger than the width of the rope, whereas round rope necessitates the use of a large drum. Its use is recommended where saving of machinery space is an object. It does not twist or spin in the shaft. It is also used for operating spouts on coal and ore docks, and for raising and lowering emergency gates on canals and similar machinery. For details of methods of fastening it to drums, the manufacturers should be consulted. Drums and sheaves should be as large as possible. A rule for the diameter of the drum is $D = ct$, where $D$ is diameter of drum at bottom, in., $t$ = thickness of rope, in., and $c$ = 100 for drums and 160 for sheaves. Sheaves should be crowned at the center and have deep flanges to guide the rope. See Table 25.

**TOWING AND RUNNING ROPES** are made of 6 12-, 24-, or 37-wire strands with a hemp core in each strand. See Table 26. They are lighter than hemp hawsers of equal strength and are flexible enough to work around the same bitts as hemp rope of equivalent strength. The following is a comparison of hawsers of equal strength:

| | Tarred Russian Hemp | White Manila Hawser | 4-in. Galv. Steel Hawser | Studded Link Chain |
|---|---|---|---|---|
| Weight per fathom, lb. | 39 | 18 | 13 | 68 |
| Circumference, in. | 13 | 10 | 4 | 1 1/8 diam. |

**TRACK STRAND FOR AERIAL TRAMWAYS.**—Composed of several successive layers of wires wrapped around a single wire core, the number of wires varying with the diameter of the cable. The cable is made in plow steel and cast steel grades. See Table 27.

**LOCKED WIRE CABLE.**—Locked wire cable and locked coil track cable, of the general form shown in Fig. 10, are used as track cables for aerial tramways. They differ only in the number and size of wires used, and both are made of cast steel. The locked wire cable is the more flexible of the two. These cables are smoother than the track cable described above. See Table 28.

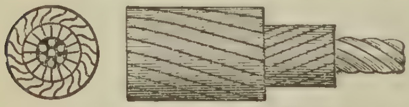


Fig. 10. Locked Wire Cable

**Table 25.—Steel Flat Rope**

| Thickness, in. | Width, in. | Wt. per ft., lb. | Breaking Strength, Tons (2000 lb.) | | Thickness, in. | Width, in. | Wt. per ft., lb. | Breaking Strength, Tons (2000 lb.) | | Thickness, in. | Width, in. | Wt. per ft., lb. | Breaking Strength, Tons (2000 lb.) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Cast Steel | Plow Steel | | | | Cast Steel | Plow Steel | | | | Cast Steel | Plow Steel |
| 1/4 | 1 1/2 | 0.69 | 14.0 | 16.9 | 3/8 | 5 | 3.03 | 63.8 | 78.1 | 5/8 | 5 | 5.04 | 104.8 | 129.4 |
| 1/4 | 2 | 0.88 | 18.0 | 21.7 | 3/8 | 5 1/2 | 3.42 | 72.2 | 88.5 | 5/8 | 5 1/2 | 5.59 | 116.4 | 143.8 |
| 1/4 | 2 1/2 | 1.15 | 22.0 | 26.5 | 3/8 | 6 | 3.63 | 76.5 | 93.7 | 5/8 | 6 | 6.14 | 128.1 | 158.0 |
| 1/4 | 3 | 1.34 | 26.0 | 31.3 | 1/2 | 2 1/2 | 2.13 | 45.8 | 56.4 | 5/8 | 7 | 7.23 | 151.3 | 187.0 |
| 5/16 | 2 | 1.17 | 24.0 | 29.2 | 1/2 | 3 | 2.47 | 53.4 | 65.8 | 5/8 | 8 | 8.32 | 174.6 | 216.0 |
| 5/16 | 2 1/2 | 1.47 | 30.0 | 36.5 | 1/2 | 3 1/2 | 2.82 | 61.0 | 75.2 | 3/4 | 5 | 6.50 | 134.2 | 166.0 |
| 5/16 | 3 | 1.75 | 36.0 | 43.8 | 1/2 | 4 | 3.16 | 68.6 | 84.6 | 3/4 | 6 | 7.31 | 150.9 | 186.8 |
| 5/16 | 3 1/2 | 2.03 | 42.0 | 51.2 | 1/2 | 4 1/2 | 3.82 | 76.2 | 94.0 | 3/4 | 7 | 8.13 | 167.7 | 207.5 |
| 5/16 | 4 | 2.29 | 48.0 | 58.5 | 1/2 | 5 | 4.16 | 91.5 | 112.8 | 3/4 | 8 | 9.70 | 184.5 | 228.3 |
| 3/8 | 2 | 1.25 | 25.5 | 31.2 | 1/2 | 5 1/2 | 4.50 | 99.1 | 122.2 | 7/8 | 5 | 7.50 | 157.0 | 192.7 |
| 3/8 | 2 1/2 | 1.64 | 34.0 | 41.6 | 1/2 | 6 | 4.85 | 106.7 | 131.6 | 7/8 | 6 | 8.56 | 179.4 | 220.2 |
| 3/8 | 3 | 1.84 | 38.3 | 46.8 | 1/2 | 7 | 5.85 | 122.0 | 150.4 | 7/8 | 7 | 9.63 | 201.9 | 247.8 |
| 3/8 | 3 1/2 | 2.23 | 46.7 | 57.2 | 5/8 | 3 1/2 | 3.40 | 69.9 | 86.3 | 7/8 | 8 | 10.69 | 224.3 | 275.3 |
| 3/8 | 4 | 2.44 | 51.0 | 62.4 | 5/8 | 4 | 3.95 | 81.5 | 100.6 | .... | ..... | ...... | ...... | ...... |
| 3/8 | 4 1/2 | 2.83 | 59.5 | 72.8 | 5/8 | 4 1/2 | 4.50 | 93.1 | 115.0 | .... | ..... | ...... | ...... | ...... |

### Table 26.—Galvanized Steel Hawser, Towing and Running Rope

6 Strands and 1 Hemp Core

| Diameter, in. | Circumference, in. | 37-wire Strand* Special Plow Steel One Hemp Core | | 24-wire Strand† 7 Hemp Cores | | | 12-wire Strand‡ 7 Hemp Cores | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Wt. per foot, lb. | Approx. Breaking Strength, Tons (2000 lb.) | Wt. per foot, lb. | Approx. Breaking Strength, Tons (2000 lb.) | | Wt. per foot, lb. | Approx. Breaking Strength, Tons (2000 lb.) | | |
| | | | | | Cast Steel | Plow Steel | | Iron | Cast Steel | Plow Steel |
| 2 3/8 | 7 1/2 | 8.74 | 173.3 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 2 5/16 | 7 1/4 | 8.29 | 164.6 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 2 1/4 | 7 1/8 | 7.85 | 156.2 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 2 1/8 | 6 3/4 | 7.00 | 140.2 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 2 1/16 | 6 1/2 | 6.59 | 132.6 | 5.87 | 98.0 | 118.0 | 4.47 | ........ | 66.6 | 81.3 |
| 2 | 6 1/4 | 6.20 | 125.0 | 5.52 | 92.0 | 112.0 | 4.20 | ........ | 62.7 | 76.9 |
| 1 15/16 | 6 | 5.82 | 117.8 | 5.18 | 86.6 | 105.0 | 3.94 | ........ | 59.0 | 72.2 |
| 1 13/16 | 5 3/4 | 5.09 | 103.4 | 4.53 | 76.2 | 92.3 | 3.45 | ........ | 51.7 | 63.4 |
| 1 3/4 | 5 1/2 | 4.75 | 96.5 | 4.23 | 71.2 | 86.2 | 3.22 | ........ | 48.3 | 59.3 |
| 1 11/16 | 5 1/4 | 4.41 | 89.8 | 3.93 | 66.3 | 80.2 | 2.99 | ........ | 45.0 | 55.3 |
| 1 5/8 | 5 | 4.09 | 83.4 | 3.64 | 61.6 | 74.5 | 2.77 | ........ | 41.8 | 51.5 |
| 1 1/2 | 4 3/4 | 3.49 | 71.2 | 3.11 | 52.6 | 63.6 | 2.36 | ........ | 35.8 | 44.3 |
| 1 7/16 | 4 1/2 | 3.20 | 65.5 | 2.85 | 48.4 | 58.5 | 2.17 | ........ | 33.0 | 40.7 |
| 1 3/8 | 4 1/4 | 2.93 | 60.0 | 2.61 | 44.4 | 53.6 | 1.99 | ........ | 30.2 | 37.4 |
| 1 1/4 | 4 | 2.42 | 49.7 | 2.16 | 36.7 | 44.4 | 1.64 | ........ | 25.1 | 31.0 |
| 1 3/16 | 3 3/4 | 2.19 | 44.9 | 1.95 | 33.2 | 40.1 | 1.48 | ........ | 22.8 | 28.1 |
| 1 1/8 | 3 1/2 | 1.96 | 40.3 | 1.75 | 29.9 | 36.0 | 1.33 | ........ | 20.5 | 25.3 |
| 1 1/16 | 3 1/4 | 1.75 | 36.0 | 1.56 | 26.7 | 32.2 | 1.19 | ........ | 18.3 | 22.6 |
| 1 | 3 | 1.55 | 31.9 | 1.38 | 23.7 | 28.5 | 1.05 | 8.1 | 16.3 | 20.0 |
| 7/8 | 2 3/4 | 1.19 | 24.6 | 1.06 | 18.3 | 22.0 | 0.80 | 6.3 | 12.5 | 15.4 |
| 13/16 | 2 1/2 | 1.02 | 21.3 | 0.91 | 15.8 | 19.1 | .69 | 5.46 | 10.8 | 13.3 |
| 3/4 | 2 1/4 | 0.87 | 18.3 | .78 | 13.6 | 16.4 | .59 | 4.7 | 9.26 | 11.3 |
| 5/8 | 2 | ........ | ........ | .54 | 9.59 | 11.6 | .41 | 3.35 | 6.46 | 7.9 |
| 9/16 | 1 3/4 | ........ | ........ | ........ | ........ | ........ | .33 | 2.75 | 5.26 | 6.43 |
| 1/2 | 1 1/2 | ........ | ........ | .35 | 6.37 | 7.63 | .26 | 2.18 | 4.23 | 5.1 |
| 7/16 | 1 1/4 | ........ | ........ | ........ | ........ | ........ | .20 | 1.69 | 3.29 | 4.01 |
| 3/8 | 1 1/8 | ........ | ........ | .194 | 3.67 | 4.4 | .15 | 1.26 | 2.47 | 3.01 |
| 5/16 | 1 | ........ | ........ | ........ | ........ | ........ | .10 | 0.88 | 1.76 | 2.13 |

* For deep sea and lake towing. †Hawser and mooring lines. ‡Running rope.

### Table 27.—Track Strand for Aerial Tramways

| Diameter, in. | No. of Wires | Weight per 100 ft., lb. | Breaking Strength, (2000 lb.) Tons | | Diameter, in. | No. of Wires | Weight per 100 ft., lb. | Breaking Strength, (2000 lb.) Tons | | Diameter, in. | No. of Wires | Weight per 100 ft., lb. | Breaking Strength, (2000 lb.) Tons | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Cast Steel | Plow Steel | | | | Cast Steel | Plow Steel | | | | Cast Steel | Plow Steel |
| 2 1/2 | 91 | 1272 | 285.0 | 335.0 | 1 5/8 | 61 | 536 | 124.0 | 146.0 | 7/8 | 19 | 155 | 37.6 | 44.4 |
| 2 1/4 | 91 | 1019 | 233.0 | 266.0 | 1 1/2 | 37 | 455 | 108.4 | 127.5 | 3/4 | 19 | 112 | 27.6 | 32.5 |
| 2 1/8 | 91 | 928 | 204.0 | 240.0 | 1 3/8 | 37 | 379 | 88.8 | 105.0 | 5/8 | 19 | 79 | 19.2 | 22.3 |
| 2 | 61 | 813 | 185.0 | 218.0 | 1 1/4 | 37 | 310 | 71.8 | 84.6 | 9/16 | 19 | 63 | 15.0 | 18.0 |
| 1 7/8 | 61 | 711 | 161.0 | 189.0 | 1 1/8 | 37 | 258 | 60.0 | 70.7 | 1/2 | 19 | 49 | 12.6 | 15.3 |
| 1 3/4 | 61 | 621 | 145.8 | 171.0 | 1 | 19 | 198 | 49.2 | 58.0 | ..... | .... | .... | ....... | ...... |

### Table 28.—Locked Coil Track and Locked Wire Cable

| Diameter, in. | Approximate Circumference, in. | Approximate Weight per foot, lb. | | Approximate Breaking Strength, Tons (2000 lb.) | | Diameter, in. | Approximate Circumference, in. | Approximate Weight per foot, lb. | | Approximate Breaking Strength, Tons (2000 lb.) | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Locked Coil | Locked Wire | Locked Coil | Locked Wire | | | Locked Coil | Locked Wire | Locked Coil | Locked Wire |
| 2 1/4 | 7 1/8 | ....... | 12.65 | ....... | 195 | 1 1/8 | 3 1/2 | 3.16 | 3.16 | 54 | 54 |
| 2 | 6 1/4 | 10.0 | 10.00 | 158 | 158 | 1 | 3 | 2.50 | 2.50 | 42 | 42 |
| 1 7/8 | 5 7/8 | 8.79 | 8.79 | 138 | 138 | 7/8 | 2 3/4 | 1.92 | 1.92 | 32 | 32 |
| 1 3/4 | 5 1/2 | 7.66 | 7.66 | 125 | 125 | 3/4 | 2 1/4 | ....... | 1.41 | ....... | 22 |
| 1 5/8 | 5 1/8 | 6.60 | 6.60 | 108 | 108 | 5/8 | 2 | ....... | 0.98 | ....... | 16 |
| 1 1/2 | 4 3/4 | 5.63 | 5.63 | 93 | 93 | 9/16 | 1 3/4 | ....... | .79 | ....... | 13 |
| 1 3/8 | 4 1/4 | 4.73 | 4.73 | 78 | 78 | 1/2 | 1 1/2 | ....... | .63 | ....... | 10 |
| 1 1/4 | 4 | 3.91 | 3.91 | 65 | 65 | ....... | ....... | ....... | ....... | ....... | ....... |

## Wire Rope Fittings

**Wire Rope Splices and Fittings.** (See also page 23–10.)—Three types of splices are used with wire rope—the end-to-end, the endless, and the thimble splice. The first two may be either long or short splices. The end-to-end splice is used to attach one rope to another by interlocking the strands, as shown on page 6–46. Ropes of similar construction can be so spliced, but a right lay rope cannot be spliced into a left lay rope. In making a splice a certain amount of rope is used up. The additional length to be allowed for a splice is shown in Table 29.

The endless splice is made in the same manner as the end-to-end splice.

The thimble splice is the forming of a loop in the end of a rope and then interlocking the strands, the loop usually containing a thimble or other fitting. Dimensions of thimbles are given in Table 30.

**Shackles** are used to connect ropes whose ends are equipped with thimble sockets, turnbuckles, etc. See Table 31. Select the size of shackle whose strength is equal to that of the rope to be used with it.

### Table 29.—Length of Wire Rope to be Allowed for Splices

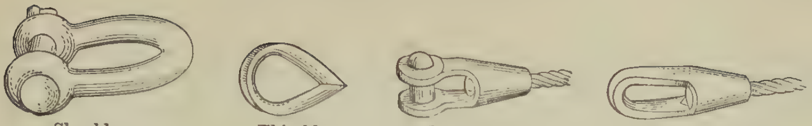
| Diameter of rope, in. | 1/4–3/8 | 1/2–5/8 | 3/4–7/8 | 1–1 1/8 | 1 1/4–1 3/8 | 1 1/2 |
|---|---|---|---|---|---|---|
| Allowance, Short Splice, ft. | 15 | 20 | 24 | 28 | 32 | 36 |
| " Long Splice, ft. | 30 | 40 | 50 | 60 | 70 | 80 |
| " Thimble Splice, ft. | 1 | 1 1/2 | 2 | 3 | 3 1/2 | 4 |

### Table 30.—Dimensions and Weight of Galvanized Wire Rope Thimbles

| | Standard Thimbles for Iron and Cast Steel Ropes, Guys and Stationary Ropes | | | | Heavy Thimbles for Plow Steel and Monitor Steel Ropes and Material Handling Ropes | | | | Standard Cast Steel Hawser Thimbles | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diam. of Rope, in. | Width of Opening, in. | Length of Opening, in. | Max. Diam. of Thimble Pin, in. | Weight, lb. | Width of Opening, in. | Length of Opening, in. | Max. Diam. of Thimble Pin, in. | Weight, lb. | Circumference of Hawser, in. | Width of Opening, in. | Length of Opening, in. | Weight, lb. |
| 1/4 | 3/4 | 1 1/4 | 11/16 | 0.034 | 7/8 | 1 5/8 | 13/16 | 0.12 | ...... | ...... | ...... | ........ |
| 5/16 | 13/16 | 1 5/16 | 3/4 | .056 | 1 | 1 7/8 | 15/16 | .15 | ...... | ...... | ...... | ........ |
| 3/8 | 1 | 1 1/2 | 15/16 | .081 | 1 1/8 | 2 1/8 | 1 1/16 | .22 | ...... | ...... | ...... | ........ |
| 7/16 | 1 1/16 | 1 5/8 | 1 | .103 | 1 1/4 | 2 1/2 | 1 3/16 | .30 | ...... | ...... | ...... | ........ |
| 1/2 | 1 1/8 | 1 3/4 | 1 1/16 | .115 | 1 1/2 | 2 3/4 | 1 7/16 | .50 | ...... | ...... | ...... | ........ |
| 9/16 | 1 3/8 | 2 1/4 | 1 1/4 | .356 | 1 1/2 | 2 3/4 | 1 7/16 | .50 | ...... | ...... | ...... | ........ |
| 5/8 | 1 3/8 | 2 1/4 | 1 1/4 | .356 | 1 3/4 | 3 1/4 | 1 5/8 | .75 | 2 | 3 | 4 1/2 | 4 |
| 3/4 | 1 9/16 | 2 3/8 | 1 1/2 | .494 | 2 | 3 3/4 | 1 7/8 | 1.30 | 2 3/8 | 3 | 4 1/2 | 4 |
| 7/8 | 1 3/4 | 2 1/2 | 1 5/8 | .58 | 2 1/4 | 4 1/4 | 2 1/8 | 1.60 | 2 3/4 | 3 3/4 | 5 3/4 | 7 |
| 1 | 2 | 3 | 1 7/8 | .75 | 2 1/2 | 4 1/2 | 2 3/8 | 2.80 | 3 1/8 | 3 3/4 | 5 3/4 | 7 |
| 1 1/8 | 2 5/16 | 3 3/8 | 2 1/8 | .96 | 2 7/8 | 5 | 2 3/4 | 4.50 | 3 1/2 | 4 1/4 | 6 3/4 | 9 |
| 1 1/4 | 2 5/8 | 3 3/4 | 2 1/2 | 1.61 | 2 7/8 | 5 1/8 | 2 3/4 | 4.50 | 3 7/8 | 4 1/4 | 6 3/4 | 9 |
| 1 3/8 | 2 7/8 | 4 | 2 3/4 | 2.67 | 3 1/2 | 6 1/4 | 3 3/8 | 7.50 | 4 3/8 | 4 1/2 | 7 1/8 | 13 |
| 1 1/2 | 2 7/8 | 4 | 2 3/4 | 2.67 | 3 1/2 | 6 1/4 | 3 3/8 | 7.50 | 4 3/4 | 5 | 8 | 20 |
| 1 5/8 | 3 1/8 | 4 3/4 | 3 | 3.08 | 4 | 8 | 3 7/8 | 11.75 | 5 1/8 | 5 | 8 | 26 |
| 1 3/4 | 3 1/8 | 4 3/4 | 3 | 3.08 | 4 1/2 | 9 | 4 3/8 | 18.00 | 5 1/2 | 5 | 8 | 26 |
| 1 7/8 | 3 3/8 | 5 | 3 1/4 | 5.48 | 5 | 10 | 4 7/8 | 22.00 | 5 7/8 | 6 | 9 1/2 | 36 |
| 2 | 3 3/8 | 5 | 3 1/4 | 5.48 | 6 | 12 | 5 7/8 | 30.00 | 6 1/4 | 6 | 9 1/2 | 36 |
| 2 1/8 | ...... | ...... | ...... | ........ | 7 | 14 | 6 7/8 | 52.00 | 6 3/4 | 7 | 11 | 53 |
| 2 1/4 | ...... | ...... | ...... | ........ | 7 | 14 | 6 7/8 | 52.00 | 7 1/8 | 7 | 11 | 53 |

### Table 31.—Iron Guy Shackles

| Size, in. (Diameter of Iron) | Maximum Strength, lb. | Length Inside, in. | Width Between Eyes, in. | Diameter of Pin, in. | Approximate Weight, lb. | Size, in. (Diameter of Iron) | Maximum Strength, lb. | Length Inside, in. | Width Between Eyes, in. | Diameter of Pin, in. | Approximate Weight, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3/8 | 10,890 | 1 3/8 | 5/8 | 1/2 | 0.30 | 1 1/8 | 90,200 | 4 1/2 | 1 7/8 | 1 1/4 | 6.80 |
| 7/16 | 15,200 | 1 3/4 | 13/16 | 9/16 | .48 | 1 1/4 | 92,040 | 5 | 2 | 1 3/8 | 9.40 |
| 1/2 | 18,390 | 1 7/8 | 13/16 | 5/8 | .70 | 1 3/8 | 94,100 | 5 1/2 | 2 1/8 | 1 1/2 | 12.20 |
| 9/16 | 24,800 | 1 7/8 | 7/8 | 11/16 | .90 | 1 1/2 | 103,800 | 6 | 2 1/4 | 1 5/8 | 16.40 |
| 5/8 | 33,400 | 2 1/4 | 1 3/16 | 3/4 | 1.40 | 1 5/8 | 155,542 | 6 1/2 | 2 1/2 | 1 3/4 | 19.00 |
| 3/4 | 43,400 | 3 | 1 9/32 | 7/8 | 2.20 | 1 3/4 | 172,400 | 7 | 2 3/4 | 1 7/8 | 24.00 |
| 7/8 | 55,200 | 3 1/2 | 1 3/8 | 1 | 3.40 | 2 | 235,620 | 8 | 3 1/4 | 2 1/8 | 38.20 |
| 1 | 74,900 | 4 | 1 3/4 | 1 1/8 | 5.00 | ..... | .......... | ...... | ...... | ...... | ....... |



Shackle Thimble Open Socket Closed Socket

FIG. 11. Wire Rope Fastenings

**Clips** are fastenings usually used to form a thimble loop where the use of the rope is temporary. The U-bolt of the clip should rest on the short end of the rope so that the flat bears on the tension side of the loop. Otherwise the rope will be injured by the clip putting a crimp in the tension side. Not less than 2, and preferably 4 or 6 clips should be used, especially on the larger diameters of rope.

**Clamps** are temporary fastenings, usually used to form loops without thimbles. From 2 to 6 clamps should be used for each end fastening, and alternate clips and clamps are preferable.

**Turnbuckles** are necessary where slack in rope is to be taken up or a uniform tension maintained. The turnbuckles should always be equal in strength to that of the rope used with them. See Table 32 for dimensions and strength. Where a greater take-up is required than that given in the table, two or more turnbuckles should be used. Turnbuckles are usually made with the styles of ends shown in Fig. 12.

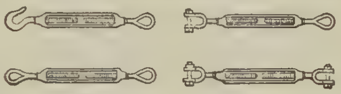


FIG. 12.. Turnbuckles

**Sockets** are the strongest type of rope fastening. Dimensions of closed and open sockets are shown in Table 33. Bridge sockets of both open and closed type are also

**Table 32.—Standard Turnbuckles**

| Diameter of Screw, in. | Take-up, in. | Approx. Breaking Strength, lb.* | Approx. Weight Eye Stub, Turn-buckle, lb. | Eye Stub, Inside Diameter of Eye, in. | Solid Eye Stub | | Clevis Stub | |
|---|---|---|---|---|---|---|---|---|
| | | | | | Outside Diameter of Eye, in. | Thick-ness of Eye, in. | Diameter of Pin, in. | Width Between Jaws, in. |
| 3/8 | 6 | 4,000 | 1.00 | 3/4 | ......... | ......... | 7/16 | 5/8 |
| 1/2 | 9 | 7,500 | 1.75 | 3/4 | ......... | ......... | 9/16 | 3/4 |
| 5/8 | 9 | 12,000 | 3.25 | 1 | ......... | ......... | 3/4 | 7/8 |
| 3/4 | 12 | 18,000 | 5.50 | 1 1/4 | 1 7/8 | 3/4 | 7/8 | 1 |
| 7/8 | 12 | 25,000 | 8.00 | 1 5/16 | 2 1/8 | 7/8 | 1 | 1 1/8 |
| 1 | 12 | 33,000 | 11.00 | 1 1/2 | 2 9/16 | 1 | 1 1/8 | 1 1/4 |
| 1 1/8 | 18 | 41,500 | 18.00 | 1 11/16 | 2 3/4 | 1 1/8 | 1 1/8 | 1 1/4 |
| 1 1/4 | 18 | 53,500 | 23.25 | 1 7/8 | 3 | 1 1/4 | 1 3/8 | 1 1/2 |
| 1 1/2 | 18 | 78,000 | 34.50 | 2 1/4 | 3 1/2 | 1 1/2 | 1 5/8 | 1 3/4 |
| 1 3/4 | 24 | 105,000 | 57.75 | 2 5/8 | 4 1/4 | 1 3/4 | 2 | 2 |
| 2 | 24 | 138,000 | 79.00 | 3 | 4 3/4 | 2 | 2 1/4 | 2 1/4 |
| 2 1/4 | 24 | 181,000 | 132.50 | 3 3/4 | 6 13/16 | 2 1/4 | ......... | ......... |
| 2 1/2 | 24 | 223,000 | 140.00 | 3 3/4 | 6 13/16 | 2 1/2 | ......... | ......... |

* The recommended working load = 1/5 approximate breaking strength.

**Table 33.—Sockets**

| Diameter of Rope, in. | Length of Basket, in. | Open Sockets | | | | | Closed Sockets | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Overall Length, in. | Width between Jaws, in. | Diameter of Pin, in. | Basket to Center Line of Pin | Approximate Weight, lb. | Overall Length, in. | Length of Opening, in. | Width of Opening, in. | Approximate Weight, lb. |
| 1/4-5/16 | 2 | 4 5/16 | 11/16 | 11/16 | 1 9/16 | 0.9 | 4 1/4 | 1 13/16 | 11/16 | 0.5 |
| 3/8 | 2 | 4 5/8 | 13/16 | 13/16 | 1 3/4 | 1.1 | 4 5/8 | 2 1/16 | 15/16 | 0.8 |
| 1/2-9/16 | 2 1/2 | 5 9/16 | 1 | 1 | 2 | 2.3 | 5 1/2 | 2 5/16 | 1 1/8 | 1.6 |
| 5/8 | 3 | 6 3/4 | 1 1/4 | 1 3/16 | 2 1/2 | 3.8 | 6 3/8 | 2 9/16 | 1 3/8 | 3.0 |
| 3/4 | 3 1/2 | 7 15/16 | 1 1/2 | 1 3/8 | 3 | 6.0 | 7 5/8 | 3 1/16 | 1 5/8 | 4.5 |
| 7/8 | 4 | 9 1/4 | 1 3/4 | 1 5/8 | 3 1/2 | 10.0 | 8 7/8 | 3 5/8 | 1 7/8 | 7.0 |
| 1 | 4 1/2 | 10 9/16 | 2 | 2 | 4 | 15.5 | 10 | 4 1/8 | 2 1/4 | 11.0 |
| 1 1/8 | 5 | 11 13/16 | 2 1/4 | 2 1/4 | 4 1/2 | 22.0 | 11 1/8 | 4 5/8 | 2 1/2 | 16.0 |
| 1 1/4 | 5 1/2 | 13 3/16 | 2 1/2 | 2 1/2 | 5 | 32.0 | 12 5/16 | 5 3/16 | 2 3/4 | 22.0 |
| 1 3/8 | 5 1/2 | 13 3/16 | 2 1/2 | 2 1/2 | 5 | 32.0 | 12 5/16 | 5 3/16 | 2 3/4 | 22.0 |
| 1 1/2 | 6 | 15 1/8 | 3 | 2 3/4 | 6 | 46.0 | 14 1/8 | 6 3/16 | 3 1/8 | 28.0 |
| 1 5/8 | 6 1/2 | 16 1/4 | 3 | 3 | 6 1/2 | 55.0 | 15 3/8 | 6 3/4 | 3 1/4 | 36.0 |
| 1 3/4 | 7 1/2 | 18 1/4 | 3 1/2 | 3 1/2 | 7 | 82.0 | 17 1/2 | 7 13/16 | 3 17/32 | 53.0 |
| 1 7/8 | 7 1/2 | 18 1/4 | 3 1/2 | 3 1/2 | 7 | 82.0 | 17 1/2 | 7 13/16 | 3 17/32 | 53.0 |
| 2–2 1/8 | 8 1/2 | 21 1/2 | 4 | 3 3/4 | 9 | 120.0 | 19 3/4 | 8 13/16 | 3 25/32 | 80.0 |
| 2 1/4–2 3/8 | 9 | 23 1/2 | 4 1/2 | 4 1/4 | 10 | 120.0 | 21 5/8 | 9 3/4 | 4 9/32 | 105.0 |

made for bridge cables and locked tramway and cableway strand. For dimensions, see catalog of American Steel & Wire Co. Sockets develop the full strength of the rope, and are stronger than any rope that may be inserted in them. See below for directions for attaching sockets.

**ATTACHING SOCKETS TO WIRE ROPE.**—The rope should be seized at the end before cutting off and also at a distance equal to the length of the socket basket from the end, the first seizing being removed after the rope is cut off. The seizing in large ropes should be several inches long and securely wrapped with a special seizing iron to prevent untwisting of the lay of the rope, with consequent unequal tension on the strands after the socket is applied. The hemp center is cut out, back to the seizing and the wires broomed out straight, *i.e.*, untwisted, not necessarily straightened. The wires should be cleaned with benzine, naphtha or gasoline, and then dipped for 30 seconds in a bath of commercial muriatic acid. They should next be dipped in boiling water containing a small amount of soda to neutralize the acid, and then inserted in the basket of the socket. The socket should line up with the axis of the rope. After sealing its base with clay, putty or similar substance, the socket is filled with molten zinc, at a temperature not exceeding 875° F. If socketing is properly done, one or more strands of the rope will break before the socket fails. See also Bureau of Mines Bulletin No. 75, page 124.

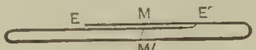


FIG. 13

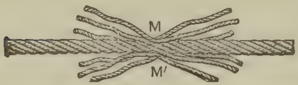


FIG. 14

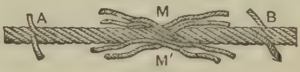


FIG. 15

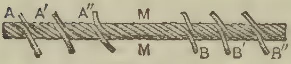


FIG. 16

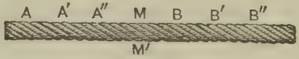


FIG. 17

Splicing Wire Rope.

**TO SPLICE A WIRE ROPE.**—The tools required will be a small marline spike, cutting nippers, and either clamps or a small hemp-rope sling with which to wrap around and untwist the rope. If a bench-vise is accessible it will be found convenient. In splicing a rope, a certain length is used up in making the splice. See Table 29.

Having measured, carefully, the length the rope should be after splicing, and marked the points *M* and *M′*, Fig. 13, unlay the strands from each end *E* and *E′* to *M* and *M′* and cut off the center at *M* and *M′*, and then:

(1) Interlock the six unlaid strands of each end alternately and draw them together so that the points *M* and *M′* meet, as in Fig. 14.

(2) Unlay a strand from one end, and following the unlay closely, lay into the seam or groove it opens the strand opposite it belonging to the other end of the rope, until within a length equal to three or four times the length of one lay of the rope, and cut the other strand to about the same length from the point of meeting as at *A*, Fig. 15.

(3) Unlay the adjacent strand in the opposite direction, and following the unlay closely, lay in its place the corresponding opposite strand, cutting the ends as described before at *B*, Fig. 15.

There are now four strands laid in place terminating at *A* and *B*, with the eight remaining at *MM′*, as in Fig. 15. It will be well after laying each pair of strands to tie them temporarily at the points *A* and *B*.

Pursue the same course with the remaining four pairs of opposite strands, stopping each pair about eight or ten turns of the rope short of the preceding pair, and cutting the ends as before.

We now have all the strands laid in their proper places with their respective ends passing each other, as in Fig. 16. All methods of rope-splicing are identical to this point: Their variety consists in the method of tucking the ends. The one given below is the one most generally practiced.

Clamp the rope firmly in a vise at a point to the left of *A*, Fig. 16, and by a hand-clamp applied near *A*, open up the rope by untwisting sufficiently to cut the core at *A*. The core is seized with the nippers, and drawn out slowly. The core is followed closely by the strand, which is crowded in the place of the core, until it is all laid in. Cut the core where the strand ends, and push the end back into its place. Remove the clamps and let the rope close together around it. Draw out the core in the opposite direction and lay the other strand in the center of the rope, in the same manner. Repeat the operation at the five remaining points and hammer the rope lightly at the points where the ends pass each other at *A*, *A′*, *B*, *B′*, etc., with small wooden mallets, and the splice is complete, as shown in Fig. 17. If a clamp and vise are not obtainable, two rope slings and short wooden levers may be used to untwist and open up the rope.

A rope spliced as above will be nearly as strong as the original rope and smooth everywhere. After running a few days, the splice, if well made, cannot be found except by close examination. The above instructions have been adopted by the leading rope manufacturers of the U. S.

# RIVETS, NAILS AND WASHERS

### Table 1.—Weight of 100 Cone Head Rivets, in Pounds

(Champion Rivet Co., Cleveland)

| Length, Under Head, in. | Rivet Diameter, in. | | | | | | |
|---|---|---|---|---|---|---|---|
| | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 |
| 1 1/2 | 22.76 | 35.00 | .......... | .......... | .......... | .......... | .......... |
| 1 5/8 | 23.84 | 36.54 | .......... | .......... | .......... | .......... | .......... |
| 1 3/4 | 24.92 | 38.08 | .......... | .......... | .......... | .......... | .......... |
| 1 7/8 | 25.99 | 39.61 | .......... | .......... | .......... | .......... | .......... |
| 2 | 27.07 | 41.15 | 58.30 | 79.20 | .......... | .......... | .......... |
| 2 1/8 | 28.15 | 42.69 | 59.35 | 81.84 | .......... | .......... | .......... |
| 2 1/4 | 29.23 | 44.23 | 61.40 | 84.47 | .......... | .......... | .......... |
| 2 3/8 | 30.30 | 45.76 | 63.45 | 87.10 | .......... | .......... | .......... |
| 2 1/2 | 31.38 | 47.30 | 65.50 | 89.74 | 121.70 | 159.11 | 200.35 |
| 2 5/8 | 32.46 | 48.84 | 67.55 | 92.37 | 125.08 | 163.29 | 205.40 |
| 2 3/4 | 33.54 | 50.38 | 69.60 | 95.00 | 128.46 | 167.46 | 210.44 |
| 2 7/8 | 34.61 | 51.91 | 71.65 | 97.64 | 131.84 | 171.64 | 215.48 |
| 3 | 35.69 | 53.45 | 73.70 | 100.27 | 135.22 | 175.81 | 220.52 |
| 3 1/8 | 36.77 | 54.99 | 75.75 | 103.90 | 138.60 | 179.98 | 225.57 |
| 3 1/4 | 37.85 | 56.53 | 77.80 | 105.53 | 141.98 | 184.16 | 230.61 |
| 3 3/8 | 38.92 | 58.06 | 79.85 | 108.17 | 145.36 | 188.33 | 235.65 |
| 3 1/2 | 39.96 | 59.60 | 81.90 | 110.80 | 148.74 | 192.51 | 240.69 |
| 3 5/8 | 41.08 | 61.14 | 83.95 | 113.43 | 152.12 | 196.68 | 245.74 |
| 3 3/4 | 42.16 | 62.68 | 86.00 | 116.07 | 155.50 | 200.86 | 250.78 |
| 3 7/8 | 43.23 | 64.21 | 88.05 | 118.70 | 158.88 | 205.03 | 255.82 |
| 4 | 44.31 | 65.75 | 90.10 | 121.33 | 162.26 | 209.20 | 260.86 |
| 4 1/8 | 45.39 | 67.29 | 92.15 | 123.97 | 165.64 | 213.38 | 265.90 |
| 4 1/4 | 46.47 | 68.83 | 94.20 | 126.60 | 169.01 | 217.55 | 270.95 |
| 4 3/8 | 47.54 | 70.36 | 96.25 | 129.23 | 172.40 | 221.73 | 275.99 |
| 4 1/2 | 48.60 | 71.90 | 98.30 | 131.86 | 175.78 | 225.90 | 281.03 |
| 4 5/8 | 49.70 | 73.44 | 100.35 | 134.50 | 179.16 | 230.07 | 286.08 |
| 4 3/4 | 50.78 | 74.98 | 102.40 | 137.13 | 182.54 | 234.25 | 291.12 |
| 4 7/8 | 51.85 | 76.51 | 104.45 | 139.76 | 185.92 | 238.42 | 296.16 |
| 5 | 52.93 | 78.05 | 106.50 | 142.40 | 189.30 | 242.60 | 301.20 |
| 5 1/8 | 54.01 | 79.59 | 108.55 | 145.03 | 192.68 | 247.87 | 306.25 |
| 5 1/4 | 55.09 | 81.13 | 110.60 | 147.66 | 196.06 | 250.94 | 311.29 |
| 5 3/8 | 56.16 | 82.66 | 122.65 | 150.29 | 199.44 | 255.12 | 316.33 |
| 5 1/2 | 57.24 | 84.20 | 114.70 | 152.93 | 202.82 | 259.29 | 321.37 |
| 5 5/8 | 58.32 | 85.74 | 116.75 | 155.56 | 206.20 | 263.47 | 326.42 |
| 5 3/4 | 59.40 | 87.28 | 118.80 | 158.19 | 209.58 | 267.64 | 331.46 |
| 5 7/8 | 60.47 | 88.81 | 120.85 | 160.83 | 212.96 | 271.81 | 336.50 |
| 6 | 61.55 | 90.35 | 122.90 | 163.46 | 216.34 | 275.99 | 341.54 |
| 6 1/4 | 63.71 | 93.43 | 127.00 | 168.73 | 223.10 | 284.34 | 351.63 |
| 6 1/2 | 65.86 | 96.50 | 131.10 | 174.00 | 229.86 | 292.68 | 361.71 |
| 6 3/4 | 68.02 | 99.58 | 135.20 | 179.26 | 236.62 | 301.03 | 371.80 |
| 7 | 70.17 | 102.65 | 139.30 | 184.53 | 243.38 | 309.38 | 381.88 |
| Weight of 100 Heads | 9.83 | 16.55 | 24.50 | 37.08 | 54.10 | 75.63 | 99.50 |

### Table 2.—Weight, in Pounds, of 100 Button Head Rivets

(Champion Rivet Co., Cleveland)

| Length Under Head, in. | Rivet Diameter, in. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 5/8 |
| 1/2 | 7.19 | 13.18 | 21.55 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 3/4 | 8.58 | 15.34 | 24.63 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 1 | 9.96 | 17.49 | 27.70 | 39.88 | 54.36 | ........ | ........ | ........ | ........ | ........ |
| 1 1/4 | 11.35 | 19.65 | 30.78 | 43.98 | 59.63 | ........ | ........ | ........ | ........ | ........ |
| 1 1/2 | 12.73 | 21.80 | 33.85 | 48.08 | 64.89 | 86.66 | ........ | ........ | ........ | ........ |
| 1 3/4 | 14.12 | 23.96 | 36.93 | 52.18 | 70.16 | 93.42 | ........ | ........ | ........ | ........ |
| 2 | 15.50 | 26.11 | 40.00 | 56.28 | 75.42 | 100.18 | 132.59 | 167.18 | 206.35 | 253.64 |
| 2 1/4 | 16.89 | 28.27 | 43.08 | 60.38 | 80.69 | 106.94 | 140.94 | 177.27 | 218.36 | 267.75 |
| 2 1/2 | 18.27 | 30.42 | 46.15 | 64.48 | 85.96 | 113.70 | 149.28 | 187.35 | 230.38 | 281.86 |
| 2 3/4 | 19.66 | 32.58 | 49.23 | 68.58 | 91.22 | 120.46 | 157.63 | 197.44 | 242.39 | 295.97 |
| 3 | 21.04 | 34.73 | 52.30 | 72.68 | 96.49 | 127.22 | 165.98 | 207.52 | 254.41 | 310.08 |
| 3 1/4 | 22.43 | 36.89 | 55.38 | 76.78 | 101.75 | 133.98 | 174.33 | 217.61 | 266.42 | 324.19 |
| 3 1/2 | 23.81 | 39.04 | 58.45 | 80.88 | 107.02 | 140.74 | 182.68 | 227.69 | 278.44 | 338.31 |
| 3 3/4 | 25.20 | 41.20 | 61.53 | 84.98 | 112.29 | 147.50 | 191.03 | 237.78 | 290.45 | 352.41 |
| 4 | 26.58 | 43.35 | 64.60 | 89.08 | 117.55 | 154.26 | 199.37 | 247.86 | 302.47 | 366.52 |
| 4 1/4 | 27.97 | 45.51 | 67.68 | 93.18 | 122.82 | 161.02 | 207.72 | 257.95 | 314.48 | 380.63 |
| 4 1/2 | 29.35 | 47.66 | 70.75 | 97.28 | 128.08 | 167.78 | 216.07 | 268.03 | 326.50 | 394.74 |
| 4 3/4 | 30.74 | 49.82 | 73.83 | 101.38 | 133.35 | 174.54 | 224.42 | 278.12 | 338.51 | 408.86 |
| 5 | 32.12 | 51.97 | 76.90 | 105.48 | 138.62 | 181.30 | 232.77 | 288.20 | 350.52 | 422.96 |
| 5 1/2 | 34.89 | 56.28 | 83.05 | 113.68 | 149.15 | 194.82 | 249.46 | 308.37 | 374.56 | 451.18 |
| 6 | 37.66 | 60.59 | 89.20 | 121.88 | 159.68 | 208.34 | 266.16 | 328.54 | 398.59 | 479.40 |
| 6 1/2 | 40.43 | 64.90 | 95.35 | 130.08 | 170.22 | 221.86 | 282.85 | 348.71 | 422.61 | 507.62 |
| 7 | 43.20 | 69.21 | 101.50 | 138.28 | 180.75 | 235.38 | 299.55 | 368.88 | 446.64 | 535.84 |
| 7 1/2 | 45.97 | 73.52 | 107.65 | 146.48 | 191.28 | 248.90 | 316.25 | 389.05 | 470.67 | 564.06 |
| 8 | 48.74 | 77.83 | 113.80 | 154.68 | 201.81 | 262.42 | 332.96 | 409.22 | 494.70 | 592.28 |
| 8 1/2 | 51.51 | 82.14 | 119.95 | 162.88 | 212.34 | 275.94 | 349.66 | 429.39 | 518.73 | 620.50 |
| 9 | 54.28 | 86.45 | 126.10 | 171.08 | 222.88 | 289.46 | 366.35 | 449.56 | 542.76 | 648.72 |
| 9 1/2 | 57.05 | 90.76 | 132.25 | 179.28 | 233.41 | 302.98 | 383.05 | 469.73 | 566.79 | 676.94 |
| 10 | 59.82 | 95.07 | 138.40 | 187.48 | 243.92 | 316.50 | 399.74 | 489.90 | 590.82 | 705.16 |
| 10 1/2 | 62.59 | 99.38 | 144.55 | 195.68 | 254.45 | 330.02 | 416.44 | 510.07 | 614.85 | 733.38 |
| 11 | 65.36 | 103.69 | 150.70 | 203.88 | 264.98 | 343.54 | 433.14 | 530.24 | 638.88 | 761.60 |
| 12 | 70.90 | 112.31 | 163.00 | 220.28 | 286.05 | 370.58 | 466.53 | 570.58 | 686.94 | 818.04 |
| 13 | 76.44 | 120.93 | 175.30 | 236.68 | 307.11 | 397.62 | 499.92 | 610.92 | 735.00 | 874.48 |
| 14 | 81.98 | 129.55 | 187.60 | 253.08 | 328.18 | 424.66 | 533.31 | 651.26 | 783.06 | 920.92 |
| 15 | 87.52 | 138.17 | 199.90 | 269.48 | 349.23 | 451.70 | 566.73 | 691.60 | 831.12 | 987.36 |
| Weight of 100 Heads | 4.42 | 8.87 | 15.40 | 23.48 | 33.30 | 46.10 | 65.80 | 86.50 | 110.23 | 140.76 |

### Table 3.—Approximate Weight, in Pounds, of 1000 Small Round or Button Head Rivets

(Townsend Co., New Brighton, Pa.)

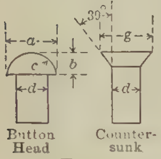
| Length of Shank, in. | Rivet Diameter, in.* | | | | | |
|---|---|---|---|---|---|---|
| | 7/16 | 3/8 | 5/16 | 1/4 | 3/16 | 1/8 |
| 1/16 | ..... | ..... | ..... | ..... | ..... | 0.87 |
| 1/8 | ..... | ..... | ..... | ..... | ..... | 1.09 |
| 3/16 | ..... | ..... | ..... | ..... | 3.66 | 1.31 |
| 1/4 | ..... | ..... | ..... | 8.71 | 4.14 | 1.52 |
| 5/16 | ..... | ..... | 17.01 | 9.57 | 4.62 | 1.74 |
| 3/8 | ..... | 29.39 | 18.35 | 10.43 | 5.10 | 1.96 |
| 1/2 | 49.31 | 33.25 | 21.03 | 12.14 | 6.07 | 2.40 |
| 5/8 | 54.57 | 37.12 | 23.72 | 13.86 | 7.03 | 2.83 |
| 3/4 | 59.84 | 40.98 | 26.40 | 15.58 | 7.99 | 3.27 |
| 7/8 | 65.09 | 44.85 | 29.09 | 17.29 | 8.95 | 3.70 |
| 1 | 70.36 | 48.71 | 31.78 | 19.01 | 9.91 | 4.14 |
| 1 1/8 | 75.62 | 52.27 | 34.46 | 20.73 | 10.87 | 4.58 |
| 1 1/4 | 80.88 | 56.44 | 37.15 | 22.45 | 11.83 | 5.01 |
| 1 3/8 | 86.14 | 60.30 | 39.84 | 24.17 | 12.79 | 5.45 |
| 1 1/2 | 91.41 | 64.17 | 42.53 | 25.88 | 13.76 | 5.89 |
| 1 5/8 | 96.67 | 68.03 | 45.21 | 27.60 | 14.72 | 6.32 |
| 1 3/4 | 101.94 | 71.90 | 47.90 | 29.32 | 15.68 | 6.76 |
| 1 7/8 | 107.20 | 75.77 | 50.58 | 31.04 | 16.64 | 7.19 |
| 2 | 112.46 | 79.63 | 53.27 | 32.75 | 17.60 | 7.53 |
| 2 1/4 | 122.99 | 87.36 | 58.64 | 36.18 | 19.52 | 8.50 |
| 2 1/2 | 133.51 | 95.09 | 64.02 | 39.62 | 21.45 | 9.38 |
| 2 3/4 | 144.04 | 102.82 | 69.39 | 43.05 | 23.37 | 10.25 |
| 3 | 154.56 | 110.55 | 74.76 | 46.49 | 25.29 | 11.12 |
| 3 1/2 | 175.61 | 126.01 | 85.51 | 53.36 | 29.14 | ...... |
| 4 | 196.66 | 141.47 | 96.25 | 60.23 | 32.98 | ...... |
| 5 | 238.76 | 172.39 | 117.74 | 73.97 | 40.67 | ...... |

* These rivets are also made in gages from No. 1 to No. 16, inclusive.

### Table 4.—Dimensions of Aluminum Structural Rivets (Fig. 1)

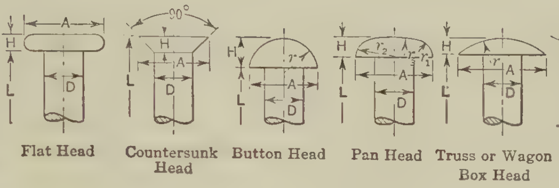
General proportions: $a = 1.75\,d$; $b = 0.75\,d$; $c = 0.885\,d$; $f = 0.50\,d$; $g = 1.81\,d$

| Dimensions, in. | Diameter, in. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 |
| *a*...... | 0.438 | 0.547 | 0.656 | 0.766 | 0.875 | 1.094 | 1.313 | 1.531 | 1.750 | 1.969 | 2.188 |
| *b*...... | 0.188 | 0.234 | 0.281 | 0.328 | 0.375 | 0.469 | 0.563 | 0.656 | 0.750 | 0.844 | 0.938 |
| *c*...... | 0.221 | 0.277 | 0.332 | 0.387 | 0.442 | 0.553 | 0.664 | 0.774 | 0.885 | 0.995 | 1.106 |
| *f*...... | 0.125 | 0.156 | 0.188 | 0.219 | 0.250 | 0.313 | 0.375 | 0.438 | 0.500 | 0.563 | 0.625 |
| *g*...... | 0.452 | 0.566 | 0.679 | 0.792 | 0.905 | 1.131 | 1.357 | 1.583 | 1.810 | 2.036 | 2.262 |



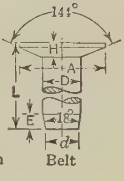
Button Head Countersunk

FIG. 1



Flat Head Countersunk Head Button Head Pan Head Truss or Wagon Box Head

FIG. 2



Belt

FIG. 3

### Table 5.—Dimensions of Small Rivets (See Fig. 2)

(American Standards Assoc. Standard B18a, 1927)

| Body Diam., *D*, in. | | | Head Diam., *A*, in. | | | | | Head Height, *H*, in. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal | Max. | Min. | Flat | Csk. | Button | Pan | Truss | Flat | Csk. | Button | Pan | Truss |
| 3/32 | 0.096 | 0.090 | 0.190 | 0.176 | 0.166 | 0.163 | 0.238 | 0.032 | 0.040 | 0.071 | 0.054 | 0.032 |
| 1/8 | .127 | .121 | .250 | .231 | .219 | .215 | .313 | .042 | .053 | .094 | .072 | .042 |
| 5/32 | .158 | .152 | .312 | .289 | .273 | .268 | .390 | .052 | .066 | .117 | .089 | .052 |
| 3/16 | .191 | .182 | .374 | .346 | .327 | .321 | .468 | .062 | .079 | .140 | .107 | .062 |
| 7/32 | .222 | .213 | .440 | .407 | .385 | .378 | .550 | .073 | .094 | .165 | .126 | .073 |
| 1/4 | .253 | .244 | .500 | .463 | .438 | .429 | .625 | .083 | .106 | .188 | .143 | .083 |
| 9/32 | .285 | .273 | .562 | .520 | .492 | .482 | .703 | .094 | .119 | .211 | .161 | .094 |
| 5/16 | .317 | .305 | .624 | .577 | .546 | .535 | .780 | .104 | .133 | .234 | .178 | .104 |
| 11/32 | .348 | .336 | .686 | .635 | .600 | .589 | .858 | .114 | .146 | .257 | .196 | .114 |
| 3/8 | .380 | .365 | .750 | .694 | .656 | .644 | .938 | .125 | .159 | .281 | .215 | .125 |
| 7/16 | .443 | .428 | .874 | .808 | .765 | .750 | 1.093 | .146 | .186 | .328 | .250 | .146 |

Notes:—Length of all types of rivets made as ordered. Radius of head of button head rivets $r = 0.885\,D$. Radii of head of pan head rivets, $r_1 = 0.314\,D$; $r_2 = 0.85\,D$; $r_3 = 3.43\,D$. Radius of head of truss rivets, $r = 2.512\,D$. Rivets are to be made of open-hearth steel of 45,000 to 55,000 lb. per sq. in. tensile strength; yield point of 0.5 × tensile strength; minimum elongation in 8 in., 1,500,000 ÷ tensile strength; manganese 0.30 to 0.50%; phosphorus $< 0.04\%$; sulphur $< 0.05\%$; rivets to test between Nos. 20 and 26 scleroscope (Shore).

### Table 6.—Coopers' Rivets (See Fig. 3)

(American Standards Assoc. Standard B18g, 1928)

| Size No.* | Diameter, in. | | | Length, in. | | | | Size No.* | Diameter, in. | | | Length, in. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Body *D*† | Head *A* | Tip *d* | Body *l* | Over-all, *L* | Head *H* | Chamfer *E* | | Body *D*† | Head *A* | Tip *d* | Body *l* | Over-all, *L* | Head *H* | Chamfer *E* |
| 1 lb. | 0.109 | 0.245 | 0.098 | 0.187 | 0.219 | 0.032 | 0.043 | 6 lb. | 0.203 | 0.456 | 0.182 | 0.406 | 0.466 | 0.060 | 0.081 |
| 1 1/2 " | .127 | .285 | .114 | .218 | .256 | .038 | .050 | 8 " | .238 | .535 | .214 | .500 | .571 | .071 | .095 |
| 2 " | .141 | .317 | .126 | .250 | .292 | .042 | .056 | 10 " | .250 | .562 | .225 | .531 | .606 | .075 | .100 |
| 2 1/2 " | .148 | .333 | .133 | .281 | .325 | .044 | .059 | 12 " | .259 | .582 | .233 | .531 | .608 | .077 | .103 |
| 3 " | .156 | .351 | .140 | .312 | .358 | .046 | .062 | 14 " | .271 | .609 | .243 | .562 | .643 | .081 | .108 |
| 4 " | .165 | .371 | .148 | .343 | .392 | .049 | .066 | 16 " | .281 | .632 | .252 | .593 | .677 | .084 | .112 |

* Size No. is trade name or weight of 1000 rivets. † Tolerances are the same as for Tinners' rivets. See Table 8.

### Table 7.—Belt Rivets (See Fig. 3)

(American Standards Assoc. Standard B18g, 1928)

| Size No.* | Diameter, in. | | | Length, in. | | | Size No.* | Diameter, in. | | | Length, in. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Body † *D* | Head *A* | Tip *d* | Overall *L* | Head *H* | Chamfer, *E* | | Body † *D* | Head *A* | Tip *d* | Overall *L* | Head *H* | Chamfer, *E* |
| 7 | 0.180 | 0.504 | 0.162 | From 3/8″ to 3/4″ by 1/8″ increments | 0.054 | 0.072 | 11 | 0.120 | 0.336 | 0.108 | From 3/8″ to 3/4″ by 1/8″ increments | 0.036 | 0.048 |
| 8 | .165 | .462 | .148 | | .049 | .066 | 12 | .109 | .305 | .098 | | .032 | .043 |
| 9 | .148 | .414 | .133 | | .044 | .059 | 13 | .095 | .266 | .085 | | .028 | .038 |
| 10 | .134 | .375 | .120 | | .040 | .053 | | | | | | | |

* Stubs Iron Wire Gage No. of stock used in body. † Body diameter tolerances are as follows: +0.002, −0.004.

## Table 8.—Tinners' Rivets

(American Standards Assoc. Standard B18g, 1928) Dimensions in Inches

| Size No.* | Head | | Body | | Size No.* | Head | | Body | | Size No.* | Head | | Body | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Diam. | Thick. | Diam.† | Length | | Diam. | Thick. | Diam.† | Length | | Diam. | Thick. | Diam.† | Length |
| 8 oz. | 0.207 | 0.027 | 0.089 | 0.16 | 2 1/2 lb. | 0.333 | 0.044 | 0.148 | 0.28 | 10 lb. | 0.535 | 0.071 | 0.238 | 0.47 |
| 12 " | .236 | .031 | .105 | .19 | 3 " | .360 | .048 | .160 | .31 | 12 " | .582 | .077 | .259 | .50 |
| 1 lb. | .249 | .033 | .111 | .20 | 4 " | .396 | .052 | .176 | .34 | 14 " | .639 | .085 | .284 | .52 |
| 1 1/2 " | .292 | .039 | .130 | .23 | 6 " | .456 | .060 | .203 | .39 | 16 " | .675 | .090 | .300 | .53 |
| 2 " | .324 | .043 | .144 | .27 | 8 " | .504 | .067 | .224 | .44 | | | | | |

* Size No. is the trade name or weight of 1000 rivets. † Body diameter tolerances are as follows: 8 oz.–2 1/2 lb., +0.002, −0.004; 3–10 lb., + 0.003, −0.006; 12–16 lb., +0.004, −0.008.

## Table 9.—Standard Rivet Burrs

(Reed and Prince Co., Worcester, Mass.)

| Size | Outside Diam., in. | Thickness, in. | No. in 1 lb. | Size | Outside Diam., in. | Thickness, in. | No. in 1 lb. |
|---|---|---|---|---|---|---|---|
| 3/8 in. | 7/8 | 0.072 | 100 | No. 6 | 1/2 | 0.058 | 371 |
| 11/32 in. | 3/4 | .072 | 140 | 3/16 in. | 1/2 | .058 | 362 |
| 5/16 in. | 3/4 | .072 | 134 | No. 7 | 1/2 | .049 | 424 |
| 9/32 in. | 11/16 | .065 | 176 | 5/32 in. | 1/2 | .042 | 488 |
| No. 3 | 11/16 | .065 | 172 | No. 9 | 7/16 | .042 | 632 |
| 1/4 in. | 5/8 | .065 | 210 | No. 10 | 13/32 | .035 | 870 |
| No. 4 | 5/8 | .065 | 229 | 1/8 in. | 13/32 | .035 | 855 |
| 7/32 in. | 9/16 | .058 | 289 | | | | |

## Table 10.—Dimensions and Weights of Wrought and Light Steel Washers

| Outside Diam., in. | Size of Hole, in. | Size of Bolt, in. | Standard Wrought Washers | | | Light Steel Washers | | Standard Wrought Washers | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Thickness | | Approx. No. per 100 lb | Gage No. | Approx. No. per 100 lb. | Outside Diam., in. | Size of Hole, in. | Size of Bolt, in. | Thickness | | Approx. No. per 100 lb. |
| | | | In. | Gage No. | | | | | | | In. | Gage No. | |
| 9/16 | 1/4 | 3/16 | 3/64 | 18 | 39,400 | 20 | 50,000 | 2 3/4 | 1 1/4 | 1 1/8 | 11/64 | 8 | 473 |
| 3/4 | 5/16 | 1/4 | 1/16 | 16 | 15,600 | 18 | 20,000 | 3 | 1 3/8 | 1 1/4 | 11/64 | 8 | 364 |
| 7/8 | 3/8 | 5/16 | 1/16 | 16 | 11,250 | 18 | 14,285 | 3 1/4 | 1 1/2 | 1 3/8 | 3/16 | 7 | 275 |
| 1 | 7/16 | 3/8 | 5/64 | 14 | 6,800 | 16 | 8,330 | 3 1/2 | 1 5/8 | 1 1/2 | 3/16 | 7 | 256 |
| 1 1/4 | 1/2 | 7/16 | 5/64 | 14 | 4,300 | 16 | 5,555 | 3 3/4 | 1 3/4 | 1 5/8 | 3/16 | 7 | 220 |
| 1 3/8 | 9/16 | 1/2 | 7/64 | 12 | 2,600 | 16 | 4,545 | 4 | 1 7/8 | 1 3/4 | 3/16 | 7 | 197 |
| 1 1/2 | 5/8 | 9/16 | 7/64 | 12 | 2,250 | 16 | 3,845 | 4 1/4 | 2 | 1 7/8 | 3/16 | 7 | 174 |
| 1 3/4 | 11/16 | 5/8 | 1/8 | 10 | 1,300 | 16 | 2,775 | 4 1/2 | 2 1/8 | 2 | 3/16 | 7 | 160 |
| 2 | 13/16 | 3/4 | 5/32 | 9 | 900 | 16 | 2,175 | 4 3/4 | 2 3/8 | 2 1/4 | 7/32 | 5 | 122 |
| 2 1/4 | 15/16 | 7/8 | 11/64 | 8 | 782 | 16 | 1,755 | 5 | 2 5/8 | 2 1/2 | 15/64 | 4 | 106 |
| 2 1/2 | 1 1/16 | 1 | 11/64 | 8 | 508 | 16 | 1,430 | ...... | ...... | ...... | ...... | ...... | ...... |

## Table 11.—Dimensions and Weights of S.A.E. Washers

| Outside Diam., in. | Size of Hole, in. | Size of Bolt, in. | Thickness, in. | Approx. Weight, lb. per 1000 Pcs. | Outside Diam., in. | Size of Hole, in. | Size of Bolt, in. | Thickness, in. | Approx. Weight, lb. per 1000 Pcs. |
|---|---|---|---|---|---|---|---|---|---|
| 3/8 | 5/32 | No. 6 | 3/64 | 1.2 | 1 1/16 | 17/32 | 1/2 | 3/32 | 17.5 |
| 7/16 | 3/16 | No. 8 | 3/64 | 1.6 | 1 3/16 | 19/32 | 9/16 | 3/32 | 22.0 |
| 1/2 | 7/32 | 3/16 | 1/16 | 3.0 | 1 5/16 | 21/32 | 5/8 | 3/32 | 26.5 |
| 5/8 | 9/32 | 1/4 | 1/16 | 4.5 | 1 3/8 | 23/32 | 11/16 | 3/32 | 28.0 |
| 11/16 | 11/32 | 5/16 | 1/16 | 5.3 | 1 1/2 | 13/16 | 3/4 | 1/8 | 43.3 |
| 13/16 | 13/32 | 3/8 | 1/16 | 7.0 | 1 3/4 | 15/16 | 7/8 | 1/8 | 60.0 |
| 15/16 | 15/32 | 7/16 | 1/16 | 10.0 | 2 | 1 1/16 | 1 | 1/8 | 78.5 |

## Table 12.—Dimensions and Weights of Malleable Iron Washers

(St. Louis Malleable Casting Co., St. Louis, Mo.)

| Size of Bolt, in.. | 3/8 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 3/4 | 2 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Standard Pattern | | | | | | | | | | | |
| Diam., in...... | 2 1/2 | 2 1/2 | 2 3/4 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 6 | 7 1/2 |
| Thickness, in... | 1/4 | 1/4 | 5/16 | 7/16 | 7/16 | 1/2 | 1/2 | 5/8 | 9/16 | 3/4 | 3/4 | 3/4 |
| Wt. per 100, lb.. | 18 | 17 | 26 | 37 1/2 | 54 | 76 | 108 | 136 | 155 | 182 | 225 | 410 |
| | Light Pattern | | | | | | | | | | | |
| Diam., in...... | 2 | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 7 | 7 1/2 |
| Thickness, in... | 3/16 | 1/4 | 5/16 | 3/8 | 3/8 | 7/16 | 1/2 | 9/16 | 9/16 | 5/8 | 3/4 | 3/4 |
| Wt. per 100, lb.. | 9 | 12 | 21 1/2 | 28 | 48 | 58 1/2 | 85 | 125 | 146 | 182 | 306 | 420 |

## Table 13.—Sizes, Lengths and Number to the Pound of Standard Steel Wire Nails

(American Steel & Wire Co., Cleveland)

| Size | Length, in. | Common Nails and Brads | Clinch | Fence | Smooth and Barbed Finishing | Fine | Barrel | Smooth and Barbed Box, Casing and Siding‡ | Flooring Brads | Boat Nails | | Barbed Car Nails | | Hinge Nails | | Slating | Barbed Roofing | Shingles | Coolers | Clout | Wire Spikes | Length, in. | Size |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | Light | Heavy | Light | Heavy | Light | Heavy | | | | | | | | |
| ..... | 5/8 | ..... | ..... | ..... | ..... | ..... | 1615 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5/8 | ..... |
| ..... | 3/4 | ..... | ..... | ..... | ..... | ..... | 1346 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 714 | ..... | ..... | 1160 | ..... | 3/4 | ..... |
| ..... | 7/8 | ..... | ..... | ..... | ..... | ..... | 906 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 469 | ..... | ..... | 808 | ..... | 7/8 | ..... |
| 2d | 1 | 876 | 710 | ..... | 1351 | 1351* | 775 | 1010 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 411 | 411 | ..... | 1084 | 705 | ..... | 1 | 2d |
| 3d fine | 1 1/8 | ..... | ..... | ..... | ..... | 778† | 700 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 365 | ..... | 848 | 628 | ..... | 1 1/8 | 3d fine |
| 3d common | 1 1/4 | 568 | 429 | ..... | 807 | ..... | 568 | 635 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 225 | 251 | 429 | ..... | 423 | ..... | 1 1/4 | 3d common |
| ..... | 1 3/8 | ..... | ..... | ..... | ..... | ..... | 400 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 230 | 345 | ..... | 390 | ..... | 1 3/8 | ..... |
| 4d | 1 1/2 | 316 | 274 | ..... | 584 | ..... | 367 | 473 | ..... | 82 | 44 | 274 | 165 | 82 | 50 | 187 | 176 | 274 | 488 | 350 | ..... | 1 1/2 | 4d |
| 5d | 1 3/4 | 271 | 235 | 142 | 500 | ..... | ..... | 406 | ..... | ..... | ..... | 142 | 118 | ..... | ..... | 142 | 151 | 235 | 364 | ..... | ..... | 1 3/4 | 5d |
| 6d | 2 | 131 | 157 | 124 | 309 | ..... | ..... | 236 | 157 | 62 | 32 | 124 | 103 | 62 | 38 | 103 | 103 | 204 | 275 | ..... | ..... | 2 | 6d |
| 7d | 2 1/4 | 161 | 139 | 92 | 238 | ..... | ..... | 210 | 139 | ..... | ..... | 92 | 76 | ..... | ..... | ..... | ..... | ..... | 212 | ..... | ..... | 2 1/4 | 7d |
| 8d | 2 1/2 | 106 | 99 | 82 | 189 | ..... | ..... | 145 | 99 | 50 | 26 | 82 | 69 | 50 | 30 | ..... | ..... | ..... | 142 | ..... | ..... | 2 1/2 | 8d |
| 9d | 2 3/4 | 96 | 90 | 62 | 172 | ..... | ..... | 132 | 90 | ..... | ..... | 62 | 54 | ..... | ..... | ..... | ..... | ..... | 130 | ..... | ..... | 2 3/4 | 9d |
| 10d | 3 | 69 | 69 | 50 | 121 | ..... | ..... | 94 | 69 | 22 | 14 | 57 | 50 | 25 | 12 | ..... | ..... | ..... | 104 | ..... | 41 | 3 | 10d |
| 12d | 3 1/4 | 63 | 62 | 40 | 113 | ..... | ..... | 88 | 54 | 20 | 13 | 50 | 42 | 23 | 11 | ..... | ..... | ..... | ..... | ..... | 38 | 3 1/4 | 12d |
| 16d | 3 1/2 | 49 | 49 | 30 | 90 | ..... | ..... | 71 | 43 | 18 | 12 | 43 | 35 | 22 | 10 | ..... | ..... | ..... | ..... | ..... | 30 | 3 1/2 | 16d |
| 20d | 4 | 31 | 37 | 23 | 62 | ..... | ..... | 52 | 31 | 16 | 10 | 31 | 26 | 19 | 9 | ..... | ..... | ..... | ..... | ..... | 23 | 4 | 20d |
| 30d | 4 1/2 | 24 | ..... | ..... | ..... | ..... | ..... | 46 | ..... | ..... | ..... | 28 | 24 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 17 | 4 1/2 | 30d |
| 40d | 5 | 18 | ..... | ..... | ..... | ..... | ..... | 35 | ..... | ..... | ..... | 21 | 18 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 13 | 5 | 40d |
| 50d | 5 1/2 | 14 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 17 | 15 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 10 | 5 1/2 | 50d |
| 60d | 6 | 11 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 15 | 13 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 9 | 6 | 60d |
| ..... | 7 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 6 | 7 | ..... |
| ..... | 8 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 4 | 8 | ..... |
| ..... | 9 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 3 1/2 | 9 | ..... |
| ..... | 10 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 3 | 10 | ..... |
| ..... | 12 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2 1/2 | 12 | ..... |

* 2d, light, 1560. † 3d, light, 1015. 3 3/8 lb. of 4d common or 2 3/8 lb. of 3d common will lay 1000 shingles. 3 1/4 lb. of 3d, fine, will put on 1000 laths, 4 nails to the lath. ‡ Siding nails from 5d to 10d inclusive.

### Table 14.—Length and Number of Cut Nails to the Pound

| Size | Length | Common | Clinch | Fence | Finishing | Fine | Barrel | Casing | Brads | Tobacco | Cut Spikes |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3/4 | 3/4 | ....... | ....... | ....... | ....... | ....... | 800 | ....... | ....... | ....... | ....... |
| 7/8 | 7/8 | ....... | ....... | ....... | ....... | ....... | 500 | ....... | ....... | ....... | ....... |
| 2d | 1 | 800 | ....... | ....... | 1100 | 1000 | 376 | ....... | ....... | ....... | ....... |
| 3d | 1 1/4 | 480 | ....... | ....... | 720 | 760 | 224 | ....... | ....... | ....... | ....... |
| 4d | 1 1/2 | 288 | ....... | ....... | 523 | 368 | 180 | 398 | ....... | ....... | ....... |
| 5d | 1 3/4 | 200 | ....... | ....... | 410 | ....... | ....... | ....... | ....... | 130 | ....... |
| 6d | 2 | 168 | 95 | 84 | 268 | ....... | ....... | 224 | 126 | 96 | ....... |
| 7d | 2 1/4 | 124 | 74 | 64 | 188 | ....... | ....... | ....... | 98 | 82 | ....... |
| 8d | 2 1/2 | 88 | 62 | 48 | 146 | ....... | ....... | 128 | 75 | 68 | ....... |
| 9d | 2 3/4 | 70 | 53 | 36 | 130 | ....... | ....... | 110 | 65 | ....... | ....... |
| 10d | 3 | 58 | 46 | 30 | 102 | ....... | ....... | 91 | 55 | ....... | 28 |
| 12d | 3 1/4 | 44 | 42 | 24 | 76 | ....... | ....... | 71 | 40 | ....... | ....... |
| 16d | 3 1/2 | 34 | 38 | 20 | 62 | ....... | ....... | 54 | 27 | ....... | 22 |
| 20d | 4 | 23 | 33 | 16 | 54 | ....... | ....... | 40 | ....... | ....... | 14 1/2 |
| 30d | 4 1/2 | 18 | 20 | ....... | ....... | ....... | ....... | 33 | ....... | ....... | 12 1/2 |
| 40d | 5 | 14 | ....... | ....... | ....... | ....... | ....... | 27 | ....... | ....... | 9 1/2 |
| 50d | 5 1/2 | 10 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | 8 |
| 60d | 6 | 8 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | 6 |

### Table 15.—Number of Nails in Keg of 150 Pounds

| Length, Inches | 1/4 in. | 5/16 in. | 3/8 in. | Length, Inches | 1/4 in. | 5/16 in. | 3/8 in. | 7/16 in. | 1/2 in. |
|---|---|---|---|---|---|---|---|---|---|
| 3 | 2250 | ........ | ....... | 7 | 1161 | 662 | 482 | 445 | 306 |
| 3 1/2 | 1890 | 1208 | ....... | 8 | ........ | 635 | 455 | 384 | 256 |
| 4 | 1650 | 1135 | ....... | 9 | ........ | 573 | 424 | 300 | 240 |
| 4 1/2 | 1464 | 1064 | ....... | 10 | ........ | ......... | 391 | 270 | 222 |
| 5 | 1380 | 930 | 742 | 11 | ........ | ......... | ......... | 249 | 203 |
| 6 | 1292 | 868 | 570 | 12 | ........ | ......... | ......... | 236 | 180 |

### Table 16.—Approximate Number of Boat Spikes in Keg of 200 Pounds

(American Steel & Wire Co., Cleveland)

| Size, in. Square | Length, in. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 14 |
| 5/8 | ........ | ........ | ........ | ........ | 214 | 190 | 176 | ....... | 144 | 122 |
| 1/2 | ........ | ........ | ........ | ........ | 324 | 286 | 258 | 244 | 220 | 192 |
| 7/16 | ........ | ........ | ........ | 480 | 438 | ....... | 378 | ....... | ....... | ....... |
| 3/8 | 1114 | 930 | 816 | 690 | 622 | 532 | 492 | ....... | 434 | ....... |
| 5/16 | 1776 | 1342 | 1124 | 978 | 858 | 776 | 706 | ....... | ....... | ....... |
| 1/4 | 2576 | 2134 | 1778 | 1488 | 1382 | ....... | ....... | ....... | ....... | ....... |

## Section 7

# STRENGTH OF MATERIALS

By Edward R. Kent
Revised by Jasper O. Draffin

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# STRENGTH OF MATERIALS

By Edward R. Kent
Revised by Jasper O. Draffin

## SIMPLE STRESSES

### 1. DEFINITIONS

**STRESS OR TOTAL STRESS** is the resultant internal force that resists change in the size or shape of a body acted on by external forces. If the external forces acting on the body increase to the extent that the maximum stress that can be developed is unable to balance the external forces, the body will break or rupture. Stresses are measured in the same units as the forces which produce them, *i.e.*, pounds, tons, etc.

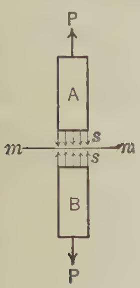

Fig. 1

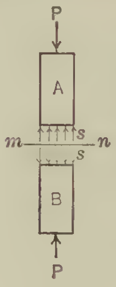

Fig. 2

**UNIT STRESS** is the stress per unit of area. Its value at any point of a section is the stress on an elementary part of the area, including the point, divided by the elementary area, and generally varies from point to point of the section. When the stress $P$ is uniformly distributed over an area $A$, the unit stress $S$ at any point of the area is $P/A$. Unit stresses are expressed in pounds per square inch, tons per square foot, kilograms per square centimeter, etc.

**TENSILE STRESS OR TENSION** is the internal force that resists the action of external forces tending to increase the length of a body. Tension is developed in a bar when the external forces $P$ act on it in the directions away from its ends. See Fig. 1. The tendency is to separate the bar into two parts $A$ and $B$. To maintain equilibrium each part is acted on at section *mn* by tensile stresses, $s$. The resultant of $s$ is equal and opposite in direction to the resultant of the forces, $P$. If the forces acting on one end of the bar total 1000 lb. then the stress on the section *mn* equals 1000 lb.

**COMPRESSIVE STRESS OR COMPRESSION** is the internal force that resists the action of external forces tending to decrease the length of a body. Compression is developed in a bar when external forces $P$ act on it in the directions toward its ends. In Fig. 2, the tendency of the external forces $P$ is to shorten the bar by pushing any two parts, as $A$ and $B$, closer together. As long as equilibrium is maintained, the resultant of the compressive stresses acting on either part at section *mn* is equal and opposite in direction to the resultant of the external forces acting at the end of the part considered.

**SHEARING STRESS OR SHEAR** is the internal force acting along a plane between adjacent parts of a body, when two equal forces, parallel to the plane considered, act on each part in opposite directions. The shear resists the tendency of one part to slide over the other part. In Fig. 3, the projecting part of the cantilever beam $A$ is acted on by external vertical forces due to the weight of $A$ and any loads that it carries. The resultant $P$ of these forces is equal, parallel to, and opposite in general direction to the upward pressure or reaction $R$ which acts on the part of the beam embedded in the wall. There exists, therefore, a tendency of the parts to assume the relative positions shown in Fig. 3. An internal force, or shear, acting along the cross-section *mn* resists the tendency of $A$ to slide vertically downward. As long as equilibrium is maintained, if $P$ equals 1000 lb., $R$ also equals 1000 lb., and the shear acting on the section will be equal to 1000 lb.

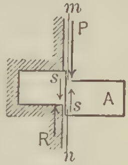

Fig. 3

**SIMPLE STRESS.**—Tension, compression, and shear are considered singly and in combination in engineering practice. When one of these is considered singly, the case is one of simple stress.

**NORMAL STRESS** is a stress that acts in a direction perpendicular to the section considered.

**AXIAL STRESS** is the stress existing in a straight homogeneous bar when the resultant of the applied loads coincides with the axis of the bar. The stress is distributed uniformly over any section normal to the axis of the bar.

**AXIAL TENSION** exists in a vertical bar supported only at its upper end. If the resultant of the applied load does not coincide with the axis of the bar, the stress is not uniform over the cross-section of the bar. A bar held in an inclined position, even though subjected to axial force applied at its ends, is not in simple axial tension, since the weight of the bar itself is an external force acting on it at an angle to its axis, and the stress is not uniform over a cross-section of the bar.

**AXIAL COMPRESSION** exists in a pier or prism of length not over 6 to 8 times the least side or diameter, when the resultant of the load supported acts through the centers of gravity of the end sections. When the length is greater than 6 or 8 times the least side or diameter, the pier is regarded as a column in which some bending has taken place. The stresses then are not uniformly distributed over a cross-section and the case is not one of simple axial compression.

**DEFORMATION** is the amount of the change in the shape of a body caused by the application of external forces. When the external forces cause tension, the deformation is the amount that the body is increased in length (elongation); when they cause compression, it is the amount that the body is decreased in length (shortening); when they cause shear, it is the amount that one part of the body slips over the adjacent part (detrusion). Deformations are measured by the same units of length used in measuring the linear dimensions of the body.

**Unit Deformation,** or deformation per unit of length, is the total amount of deformation divided by the original length of the body before the load causing the deformation was applied. Unit deformation is designated by $\epsilon$.

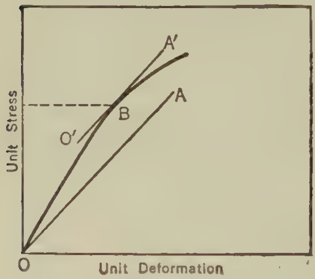

Fig. 4

Ultimate unit deformation is the unit deformation measured after the body has ruptured, expressed as a percentage of original length.

**HOOKE'S LAW** states that a body acted on by external forces will deform in proportion to the stress developed, so long as the unit stress does not exceed a certain value, which varies for the different materials. This value is the *proportional limit.*

**PROPORTIONAL LIMIT** is that unit stress at which unit deformation begins to increase at a faster rate than does unit stress; or it is the highest unit stress at which stress is proportional to deformation. It is determined by noting, on a stress-deformation diagram, the unit stress at which the curve departs from a straight line.

**ELASTIC LIMIT** is the maximum unit stress to which a material may be subjected and still be able to return to its original form upon the removal of the stress. When stressed beyond the elastic limit, a body will return to its original form only partially and thereby acquires a permanent deformation or "set."

The determination of the elastic limit logically involves the application and release of a series of increasing loads until a set is observed after the release of a load. This procedure is very slow; and, since for many metals, there seems to be no significant difference between the elastic limit so determined and the proportional limit, the latter often is accepted as equivalent to the elastic limit, and frequently is called the *proportional-elastic limit.* There is, however, no fundamental relation between the elastic limit and the proportional limit. In many of the older tests, particularly with timber, the proportional limit was incorrectly reported as the elastic limit.

**Johnson's Apparent Elastic Limit.**—In view of the difficulty of determining precisely for some materials the proportional limit, J. B. Johnson proposed as the "apparent elastic limit," the point on the stress-deformation diagram at which the rate of deformation is 50% greater than at the original. It is determined by drawing $OA$, Fig. 4, with a slope with respect to the vertical axis 50% greater than the straight-line part of the curve; the unit stress at which a parallel $O'A'$ to this line is tangent to the curve (point $B$, Fig. 4) is the apparent elastic limit.

**YIELD POINT** is the unit stress at which deformation first increases markedly without any increase in the applied load. The yield point is always above the proportional limit and true elastic limit.

Ductile metals have a well-defined yield point which is practically the same value as the proportional limit. For this reason, and because of the method of commercial testing, the yield point of ductile metals sometimes is reported as the elastic limit, or as the *commercial elastic limit.*

A.S.T.M. Standards, 1933, E8–33, specify that the term *yield point*, shall not be used in connection with material whose stress-deformation curve in the region of yield is smooth and of gradual curvature.

**YIELD STRENGTH** is the unit stress at which a material exhibits a specified limiting permanent set. It is a measure of the useful limit of materials, particularly of those whose stress-deformation curve in the region of yield is smooth and of gradual curvature.

**ULTIMATE STRENGTH,** (tensile or compressive strength), is the highest unit stress a material can sustain before rupturing.

**THE POINT OF RUPTURE,** or breaking strength, is the unit stress at which a material breaks or ruptures. It is observed in tests on steel to be slightly less than the ultimate strength, because of the large reduction in area before rupture.

**STRESS-DEFORMATION DIAGRAM.**—The relation between unit stress and unit deformation is shown by a stress-deformation diagram, in which ordinates to a curve represent unit stresses, and abscissas represent the resulting unit deformations for all values of unit stress to the point of rupture.

Fig. 5 shows a typical tension stress-deformation curve for medium steel. The proportional limit is at $a$, the yield point is at $b$, the point of ultimate stress is at $c$, and the point of rupture is at the end of the curve. The form of the curve obtained from a test will vary according to the material tested, and will be different for compression than for tension. For some materials, like cast iron, concrete, and frequently, timber, no part of the curve is a straight line.

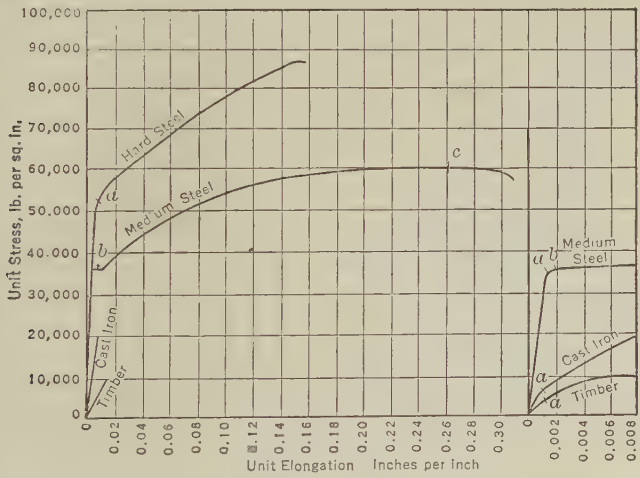

FIG. 5. Stress-Deformation Diagram.

**MODULUS OF ELASTICITY** in tension or compression is the constant which expresses the ratio of unit stress to unit deformation for all values of unit stress not exceeding the proportional limit of a material. The terms *coefficient of elasticity* and *Young's modulus* sometimes are used to express this ratio.

The deformation caused by any unit stress not exceeding the proportional limit may be computed if the modulus of elasticity is known. If a bar of length $l$ and cross-sectional area $A$ be acted on by an axial load $P$ to produce a total deformation $e$, the unit stress is $P/A$, and the unit deformation is $e/l$. Let $E$ = modulus of elasticity, $s$ = unit stress, and $\epsilon$ = unit deformation; then

$$E = \frac{\text{Unit stress}}{\text{Unit deformation}} = \frac{s}{\epsilon} = \frac{P/A}{e/l} = \frac{Pl}{Ae} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

or

$$e = \frac{Pl}{AE} = s\frac{l}{E} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

This formula can be used only for unit stresses not greater than the proportional limit, because for higher unit stresses the ratio $s/\epsilon$ is not constant. The modulus of elasticity $E$ is a measure of the stiffness of a body, or of its ability to resist deformation within the proportional limit of the material. The greater the modulus of elasticity the less will be the deformation for any unit stress not exceeding the proportional limit.

For any given material, the modulus of elasticity often is the same in tension or compression.

Average values for steel and wrought iron are respectively 30,000,000 lb. per sq. in. and 25,000,000 lb. per sq. in. See p. 1–05 and Sections 4 and 5 for values for other materials. Axial stresses always are implied when the unqualified term "modulus of elasticity" is used.

**Modulus of Elasticity in Shear.**—The expression for the modulus of elasticity in shear is similar to that for tension or compression; the deformation, however, is in a direction lying in the plane of shear of the body. The modulus of elasticity in shear also is called the *modulus of rigidity.*

Let $E_s$ = shearing modulus of elasticity, $s_s$ = unit shearing stress, $e_s$ = total lateral deformation or detrusion due to the shear, and $\epsilon_s$ = unit detrusion. Then

$$E_s = \frac{\text{Unit shearing stress}}{\text{Unit detrusion}} = \frac{s_s}{\epsilon_s} = \frac{P/A}{e_s/l} = \frac{Pl}{e_s A} \quad \text{[3]}$$

or

$$e_s = \frac{Pl}{AE_s} = \frac{s_s l}{E_s} \quad \text{[4]}$$

For metals, the shearing modulus of elasticity is approximately 0.4 times the tensile modulus of elasticity.

**POISSON'S RATIO** is the ratio of lateral unit deformation to linear unit deformation, within the elastic limit. It also is known as the *factor of lateral contraction.* See Fig. 6.

It has been established by experiment that the lateral unit deformation is proportional to the linear elongation or shortening. Let $\lambda$ = Poisson's ratio; $d$, $d_1$ = respectively, diameter of bar before and after elongation or shortening; $l$, $l_1$ = respectively, length of bar before and after elongation or shortening; $\epsilon$ = longitudinal unit elongation or shortening, as the case may be. Then

For tension.............. $l_1 = (1 + \epsilon)l$; $d_1 = (1 - \lambda\epsilon)d$ . . . . . . . . . [5]

For compression.......... $l_1 = (1 - \epsilon)l$; $d_1 = (1 + \lambda\epsilon)d$ . . . . . . . . . [6]

Average values of Poisson's ratio for the usual materials of construction are:

| Material.......... | Steel | Wrought Iron | Cast Iron | Brass | Concrete |
|---|---|---|---|---|---|
| Poisson's ratio..... | 0.333 | 0.333 | 0.25 | 0.333 | 0.10 |

**ELASTICITY** is the property of a material that enables it to deform and return to its original shape upon the removal of the load.

**DUCTILITY AND MALLEABILITY** often are used synonymously. Specifically, ductility is the ability of a material to sustain large permanent deformations in tension, as drawing into a wire; malleability is the ability to sustain large permanent deformations in compression, as beating or rolling into thin sheets.

**PLASTICITY** is that quality of a material that permits of its being molded.

A material is plastic if the smallest load produces a permanent deformation. A perfectly plastic material is non-elastic, and has no ultimate strength in the ordinary meaning of that term. Lead is a plastic material; a prism tested in compression will deform permanently under a small load, and will continue to deform as the load is increased, until it flattens to a thin sheet. Wrought iron and steel are plastic when stressed beyond the elastic limit in compression. When stressed beyond the elastic limit in tension, they are partly elastic and partly plastic, the degree of plasticity increasing as the ultimate strength is approached.

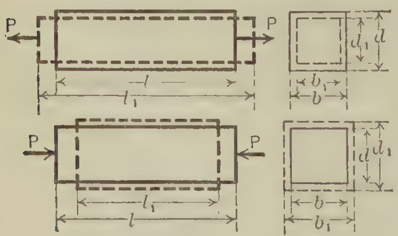

FIG. 6

**BRITTLENESS** is that property of a material that permits it to be only slightly deformed without rupture.

Brittleness is relative, no material being perfectly brittle, that is, capable of no deformation before rupture. Many materials are brittle to a greater or less degree, glass being one of the most brittle of materials. Brittle materials have relatively short stress-deformation curves. Of the common structural materials, cast iron, brick, and stone are brittle in comparison with steel.

**BRITTLENESS AND PLASTICITY** are opposite terms. Materials which have a high degree of plasticity have no brittleness, and they rupture with considerable reduction of area. The reduction of area at rupture may be considered the measure of the plasticity or brittleness of a material, a large reduction of area indicating a high degree of plasticity, and little or no reduction of area indicating a high degree of brittleness.

**TOUGHNESS** is the ability to withstand high unit stress together with great unit deformation, without complete fracture. The area $OAGH$, or $OJK$, under the curve of the stress-deformation diagram, Fig. 7, is a measure of the toughness of the material. The distinction between ductility and toughness is that ductility deals only with the ability to deform, whereas toughness considers both the ability to deform and the stress developed during deformation.

**STIFFNESS** is the ability to resist deformation under stress. The modulus of elasticity is the criterion of the stiffness of a material.

**HARDNESS** is the ability to resist very small indentations, abrasion, and plastic deformation. There is no single measure of hardness, as it is not a single property but is a combination of several properties.

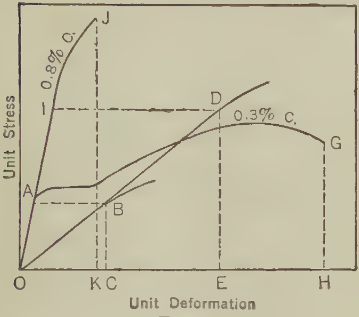

FIG. 7

**CREEP OR FLOW OF METALS** is a phase of plastic or inelastic action. Some solids, as asphalt or paraffin, flow appreciably at room temperatures under extremely small stresses; zinc, lead and tin show signs of creep at room temperature under moderate stresses. At sufficiently high temperatures, practically all metals creep under stresses which vary with temperature, the higher the temperature the lower being the stress at which creep takes place. The deformation due to creep continues to increase indefinitely and becomes of extreme importance in members subjected to high temperatures, as parts in turbines, boilers, superheaters, etc.

**Creep Limit** is the maximum unit stress under which unit distortion will not exceed a specified value during a given period of time at a specified temperature. A value much used in tests, and which has been suggested as a standard for comparing materials, is the maximum unit stress at which creep does not exceed 1% in 100,000 hours.

**TYPES OF FRACTURE.**—A bar of brittle material, as cast iron will rupture in a tension test, in a clean sharp fracture with very little reduction of cross-sectional area and very little elongation. See Fig. 8*a*. In a ductile material, as structural steel, the reduction of area and elongation are greater. See Fig. 8*b*. In compression, a prism of brittle material will break by shearing along oblique planes; the greater the brittleness of the material, the more nearly will these planes parallel the direction of the applied force. Figs. 9*a*, *b*, and *c*, arranged in order of brittleness, illustrate the type of fracture in prisms of brick, concrete, and timber. Fig. 9*d* represents the deformation of a prism of plastic material, as lead, which flattens out under load without failure.

## 2. WORK AND RESILIENCE

**EXTERNAL WORK.**—Let $P$ = axial load, lb., on a bar, producing an internal stress not exceeding the elastic limit; $s$ = unit stress produced by $P$, lb. per sq. in.; $A$ = cross-sectional area, sq. in.; $l$ = length of bar, in.; $e$ = deformation, in.; $E$ = modulus of elasticity; $W$ = external work performed on bar, in.-lb. = $^1/_2\,Pe$. Then

$$W = {}^1/_2\,As\,(sl/E) = {}^1/_2\,(s^2/E)\,Al \qquad . \; . \; . \; . \; . \; . \; . \qquad [7]$$

The factor $({}^1/_2\,s^2/E)$ is the work required per unit volume, the volume being $Al$; it is represented on the stress-deformation diagram by the area $ODE$ or area $OBC$, Fig. 7, in which $DE$ and $BC$ are ordinates representing the unit stresses considered.

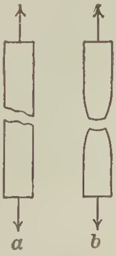

FIG. 8

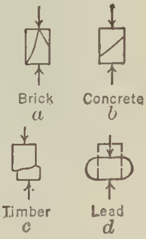

FIG. 9

**RESILIENCE** is the stress energy which may be recovered from a deformed body when the load causing the stress is removed. Within the proportional limit, the resilience is equal to the external work performed in deforming the bar, and may be determined by formula [7]. When $s$ is equal to the proportional limit, the factor $({}^1/_2\,s^2/E)$ is the *modulus of resilience, i.e.*, the measure of capacity of a unit volume of material to store stress energy up to the proportional limit. Average values of the modulus of resilience under tensile stress are given in Table 1.

The total resilience of a bar is the product of its volume and the *modulus of resilience*. The above formulas for work performed on a bar, and its resilience, do not apply if the unit stress is greater than the proportional limit.

**Work Required for Rupture.**—Since beyond the proportional limit the deformations are not proportional to the stresses, $^1/_2\,P$ does not express the mean value of the force

Table 1. Average Values of Tensile Modulus of Resilience and Toughness

| Material | Modulus of resilience, in.-lb. per cu. in. | Relative toughness (area under curve of stress-deformation diagram) |
|---|---|---|
| Gray cast iron | 1.2 | 70 |
| Malleable cast iron | 17.4 | 3,800 |
| Wrought iron | 11.6 | 11,000 |
| Low-carbon steel | 15.0 | 15,700 |
| Medium-carbon steel | 34.0 | 16,300 |
| High-carbon steel | 94.0 | 5,000 |
| Ni–Cr steel, hot-rolled | 94.0 | 44,000 |
| Vanadium steel, 0.98% C, 0.2% V, heat-treated | 260.0 | 22,000 |
| Duralumin, 17 ST | 45.0 | 10,000 |
| Rolled bronze | 57.0 | 15,500 |
| Rolled brass | 40.0 | 10,000 |
| Oak | 2.3* | 13* |

* Bending.

acting. Formula [7], therefore, does not express the work required for deformation after the proportional limit of the material has been passed, and cannot express the work required for rupture. The work per unit volume required to produce deformations beyond the proportional limit or for rupture may, however, be determined from the stress-deformation diagram, as it is measured by the area included between the axis of abscissas, and the stress-deformation curve up to the deformation in question, as *OAGH* or *OJK*, Fig. 7. This area, however, does not represent the resilience, since part of the work done on the bar is present in the form of heat and cannot be recovered.

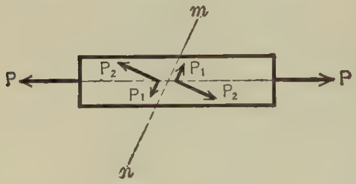

FIG. 10

## 3. COMBINED ELEMENTARY STRESSES

**SHEAR DUE TO AXIAL STRESS.**—An axial load on a bar causes normal stress on the cross-sections, and both normal and shearing stress on inclined sections. In Fig. 10, the component $P_1$ of axial stress $P$, acting parallel to the plane *mn* causes shear on section *mn*. Component $P_2$ acting perpendicular to *mn* causes normal stress on it. Shear will be zero if *mn* is at 0 or 90° to the axis. It will be maximum when *mn* is at 45°, and have the value $s_{s\,\max.} = {}^1/_2\,(P/A)$, $A$ being the area of the bar.

**COMBINED AXIAL STRESSES.**—Assuming tensile stresses to be positive and compression stresses negative, the normal stress on a section due to several applied axial loads is equal to the algebraic sum of the normal stresses caused by each load.

EXAMPLE.—If axial loads $P_1$, $P_2$, $P_3$, and $P_4$ cause unit stress on the cross-section of bar equal to $-20{,}000$, $+12{,}500$, $-6000$, and $+8000$ lb. per sq. in., respectively, the normal axial unit stress is $-5500$ lb. per sq. in. A negative sign indicates that the combined stress is compression.

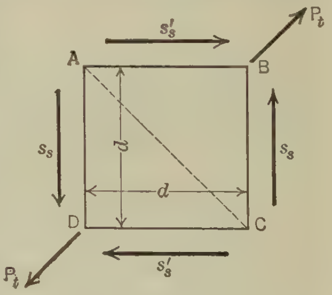

FIG. 11

**SHEAR DUE TO NORMAL STRESSES AT RIGHT ANGLES.**—When two forces act on a body in directions at right angles to each other, causing unit stresses $s_1$ and $s_2$ (tensile stress considered positive and compressive stress negative) on planes normal to the direction of the applied forces, the maximum unit shearing stress is $s_{s\,\max.} = {}^1/_2\,(s_1 - s_2)$. It acts along each of the two planes that make an angle of 45° with the planes normal to the direction of the applied forces.

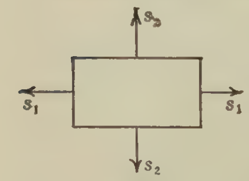

FIG. 12

**TENSILE OR COMPRESSIVE STRESS DUE TO SHEAR.**—In a body subjected to shear (e.g., a, bar in torsion) the balancing couples $s_s d$ and $s'_s d$ act on a particle of unit dimensions, Fig. 11. Since, for equilibrium, the two couples must be equal, $s_s = s'_s$. These stresses will result in a maximum tensile force, $P_t = \sqrt{2s_s^2}$, normal to the plane

$AC$ and perpendicular to the paper, and a maximum unit tensile stress normal to this plane, $s_t = \sqrt{2\,s_s^2}/\sqrt{2} = s_s$. Similarly, normal to the plane through $BD$ and perpendicular to the paper, there is a maximum unit compressive stress $s_c = s_s$. These stresses are particularly important in brittle materials, which when subjected to torsion, usually fail in tension with a helical fracture.

**DEFORMATION DUE TO NORMAL STRESSES AT RIGHT ANGLES.**—Let $s_1$ and $s_2$ be unit tensile stresses, acting on an element, at 90 deg. to each other, as shown in Fig. 12. Let $\lambda$ = Poisson's ratio, and $E$ = modulus of elasticity. The two stresses will produce unit elongations of $\epsilon_1 = s_1/E$ and $\epsilon_2 = s_2/E$, respectively, in the direction of the stresses. They will produce at the same time unit lateral contractions of $\lambda\epsilon_1$, and $\lambda\epsilon_2$, respectively, in directions at right angles to the stresses. The resultant unit elongation in the direction of $s_1$ will be $\epsilon_1 - \lambda\epsilon_2$; in the direction of $s_2$ it will be $\epsilon_2 - \lambda\epsilon_1$. If stress $s_2$ were compression and stress $s_1$ tension, the resultant elongation in the direction of $s_1$ would be $\epsilon_1 + \lambda\epsilon_2$; resultant contraction in the direction of $s_2$ would be $\epsilon_2 + \lambda\epsilon_1$.

Hence, the resultant deformation is reduced if both stresses are alike, and increased if the stresses are unlike. By experiment, $s = \epsilon E$, and for a uniaxial stress this accords with the definition that $s = P/A$; but if the resultant unit deformation due to biaxial loading is multiplied by $E$, the resulting unit stress in the direction of either applied load will not be the same as that obtained by $s = P/A$. The liability of the member to fail thus may be measured by the stress based on the external loads or by the deformation of the material. If both stresses are tension or compression, it is on the side of safety to neglect the effect of the lateral stresses and to use the stress based on the external loads, or $s = P/A$. If the lateral stresses are of opposite sign from the longitudinal stresses, and increase the longitudinal deformation, the effect of increased deformation may need to be considered. See Seely, Resistance of Materials, 2nd Ed. (John Wiley & Sons), pp. 234–239.

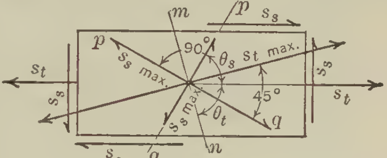

FIG. 13

**COMBINED TENSION OR COMPRESSION AND SHEAR.**—A particle subjected to tensile or compressive, as well as shearing stresses, will contain a plane $mn$, normal to which the unit tensile or compressive stress will be a maximum, and two planes, each at an angle of 45 deg. to $mn$, on which the shearing stress will be a maximum. In Fig. 13, the particle is subjected to a unit shearing stress of $s_s$, and a unit tensile stress of $s_t$, normal to plane $mn$. Then

$$s_{t\,\text{max.}} = (s_t/2) + \sqrt{s_s^2 + (s_t/2)^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [8]$$

The plane $mn$ is at an angle $\theta_t$ with the direction $s_t$, and $\tan 2\theta_t = -\,2s_s/s_t$. With the same combination of stresses, the maximum unit compressive or minimum unit tensile stress, at 90 deg. to $s_{t\,\text{max.}}$, is

$$s_{t\,\text{min.}} = s_{c\,\text{max.}} = (s_t/2) - \sqrt{s_s^2 + (s_t/2)^2} \quad . \quad . \quad . \quad . \quad . \quad . \qquad [9]$$

The maximum unit shearing stress under this combination of stresses is

$$s_{s\,\text{max.}} = \sqrt{s_s^2 + (s_t/2)^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [10]$$

and the angle $\theta_s$ of the two planes $pq$, 90 deg. apart, on which the stress acts, is given by $\tan 2\theta_s = s_t/(2s_s)$.

If the stresses are shear and compression, Fig. 14, maximum unit compressive stress is

$$s_{c\,\text{max.}} = (s_c/2) + \sqrt{s_s^2 + (s_c/2)^2} \quad . \quad . \qquad [11]$$

The angle $\theta_c$ of the plane to which the maximum unit compressive stress is normal is given by $\tan 2\theta_c = 2\,s_s/s_c$. Maximum unit shearing stress is

$$s_{s\,\text{max.}} = \sqrt{s_s^2 + (s_c/2)^2} \quad . \quad . \quad . \qquad [12]$$

and the angle $\theta_s$ of the plane on which it acts is given by $\tan 2\theta_s = -\,s_c/(2\,s_s)$. Maximum unit tensile stress for this combination of stresses is

$$s_{t\,\text{max.}} = s_{c\,\text{min.}} = (s_c/2) - \sqrt{s_s^2 + (s_c/2)^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [13]$$

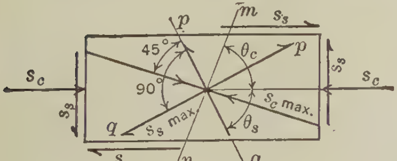

FIG. 14

**EFFECT OF NOTCHES AND HOLES IN MEMBERS UNDER SIMPLE STRESS.**—Formula $s = P/A$ assumes that the stress is uniformly distributed over the cross-section of the member. This is not true if there is any discontinuity in the cross-section, as an increase or decrease in area. In Fig. 15*a*, a plate of width $b$ having a small hole of radius $r$, is subjected to an axial pull $P$. The unit stress $s_1$, at any distance $x$ from the center line, is given by the formula (Timoshenko and Lessells, Applied Elasticity, p. 9),

$$s_1 = (s/2)\{2 + (r^2/x^2) + 3(r^4/x^4)\}. \qquad [14]$$

where $s$ = unit stress, based on the load $P$ and the area $A$, using the full width $b$. Where radius $r$ does not exceed $b/12$, stress $s_1 = 3s$.

For an elliptical hole of major axis $2a$ and minor axis $2b$, Fig. 15*b*, the maximum unit stress is $s_1 = s(1 + 2a/b)$. Notches of small radius on opposite sides of a tension specimen produce stresses approximately twice as great as the average stress; an example of this is the common briquette used in testing cement, in which the maximum tensile unit stress at the sides is 1.75 times average stress (Coker, *Proc.* Inter. Soc. Test. Mat., 1913).

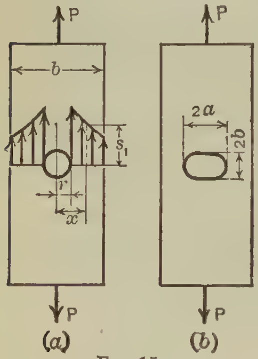

Fig. 15

**STRESS DUE TO TEMPERATURE.**—Let $l$ = length of a bar; $A$ = its sectional area; $c$ = coefficient of linear expansion for one degree, $t$ = rise or fall in temperature, deg.; $E$ = modulus of elasticity; $\lambda$ = change of length due to the rise or fall $t$; if the bar is free to expand or contract, $\lambda = ctl$.

If the bar is held so as to prevent its expansion or contraction, the stress produced by the change of temperature $= S = ActE$. The following are average values of the coefficients of linear expansion for a change in temperature of one degree Fahrenheit:

| | |
|---|---|
| For brick and stone | $c = 0.0000050$ |
| For cast iron | $c = 0.0000056$ |
| For wrought iron and steel | $c = 0.0000065$ |

The stress due to temperature should be added to or subtracted from the stress caused by other external forces, according as it acts to increase or to relieve the existing stress.

Example.—Compute the stress caused in a steel bar 1 in. square in area, by a change of temperature of 100° F. *Solution.*— $S = ActE = 1 \times 0.0000065 \times 100 \times 30{,}000{,}000 = 19{,}500$ lb. Suppose the bar is under tension of 19,500 lb. between rigid abutments before the change in temperature takes place, a cooling of 100° F. will double the tension, and a heating of 100° will reduce the tension to zero.

## 4. ALLOWABLE STRESS AND FACTOR OF SAFETY

**ALLOWABLE UNIT STRESS,** also called allowable working unit stress, allowable stress, or working stress, is the maximum unit stress to which it is considered safe to subject a member in service. The term allowable stress is preferable to working stress, since the latter often is used to indicate the actual stress in a material when in service. Allowable unit stresses for different materials for various conditions of service are specified by different authorities on the basis of test or experience. In general, for ductile materials, allowable stress is considerably less than the yield point.

**FACTOR OF SAFETY** is the ratio of ultimate strength of the material to allowable stress. The term was originated for determining allowable stress. The ultimate strength of a given material divided by an arbitrary factor of safety, dependent on material and the use to which it is to be put, gives the allowable stress. In present design practice, it is customary to use allowable stress as specified by recognized authorities or building codes rather than an arbitrary factor of safety. One reason for this is that the factor of safety is misleading, in that it implies a greater degree of safety than actually exists. For example, a factor of safety of 4 does not mean that a member can carry a load four times as great as that for which it was designed. It also should be clearly understood that, even though each part of a machine is designed with the same factor of safety, the machine as a whole does not have that factor of safety. When one part is stressed beyond the proportional limit, or particularly the yield point, the load or stress distribution may be completely changed throughout the entire machine or structure, and its ability to function thus may be changed, even though no part has ruptured.

While no definite rules can be given, if a factor of safety is to be used, the following circumstances should be taken into account in its selection:

1. When the ultimate strength of the material is known within narrow limits as for structural steel for which tests of samples have been made, when the load is entirely a steady one of a known amount and there is no reason to fear the deterioration of the metal by corrosion, the lowest factor that should be adopted is 3.

2. When the circumstances of 1 are modified by a portion of the load being variable, as in floors of warehouses, the factor should be not less than 4.

3. When the whole load, or nearly the whole, is apt to be alternately put on and taken off, as in suspension rods of floors of bridges, the factor should be 5 or 6.

4. When the stresses are reversed in direction from tension to compression, as in some bridge diagonals and parts of machines, the factor should be not less than 6.

5. When the piece is subjected to repeated shocks, the factor should be not less than 10.

6. When the piece is subjected to deterioration from corrosion the section should be sufficiently increased to allow for a definite amount of corrosion before the piece will be so far weakened by it as to require removal.

7. When the strength of the material or the amount of the load or both are uncertain, the factor should be increased by an allowance sufficient to cover the amount of the uncertainty.

8. When the strains are of a complex character and of uncertain amount, such as those in the crank-shaft of a reversing engine, a very high factor is necessary, possibly even as high as 40.

9. If the property loss caused by failure of the part may be large or if loss of life may result, as in a derrick hoisting materials over a crowded street, the factor should be large.

Table 2, from Merriman, Mechanics of Materials, gives average values of factors of safety for different conditions. The factors given by different authorities for some materials show wide differences.

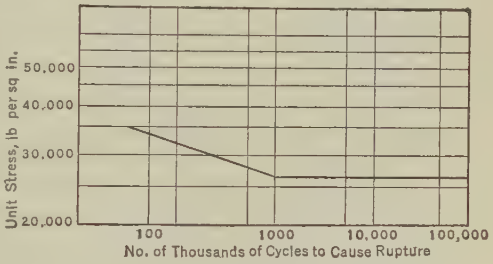

Fig. 16. Endurance Stress Curve

## 5. REPEATED STRESS

**PARTS SUBJECTED TO REPEATED STRESS.**—Where parts of a structure or machine are subjected to varying or repeated loads, the ordinary methods of computing stresses and of determining the strength of materials under static load conditions are not satisfactory. In general, in excess of one-half to ten million repetitions of loading are necessary before the parts come under this classification. Such parts are crank-shafts, shafts carrying rotating parts, as in motors and generators, turbine blades, valve parts, piston rods in steam engines, floor beams in elevated railroads, etc.

**CHARACTER OF FAILURE.**—Under repeated loadings, failures take place suddenly and without warning. The mechanism of *fatigue failure* under repeated stress appears to be that exceedingly small particles of metal are stressed beyond the elastic limit. As the stress is repeated, a tiny crack, sometimes called a *fatigue crack*, is formed and progresses until failure occurs. This action is referred to as a *progressive failure.* Therefore, where the load is repeated, anything which causes a concentration of stress is a source of weakness.

**DETERMINATION OF ENDURANCE LIMIT.**—The highest unit stress $s$ at which a material can be subjected to a very large number of repetitions of loading $N$ without evidence of failure, is the *endurance limit.* This usually is found for any given metal by constructing an $s$-$N$ diagram for that particular metal, from data obtained by tests. A piece of metal is so loaded as to produce a given stress, the load being repeated until failure occurs. Other specimens then are tested at different values of the unit stress, until enough points have been obtained to plot a curve. This curve will incline steeply at the higher stresses and gradually flatten out at the lower stresses until it approaches a horizontal line. The unit stress at which the curve becomes horizontal is the endurance limit. See Fig. 16. The most common test is that of reversed bending by the rotating beam method, since this specimen is the most easily made and the testing machines are the least costly. Tests also are made in direct tension, direct shear, direct compression, reversed torsion, and in various combination of these.

**Table 2.—Average Values of Factor of Safety**

| Material | Steady stress | Variable stress | Shocks | Material | Steady stress | Variable stress | Shocks |
|---|---|---|---|---|---|---|---|
| Cast iron.......... | 6 | 10 | 20 | Hard steel......... | 5 | 8 | 15 |
| Wrought iron...... | 4 | 6 | 10 | Timber............ | 8 | 10 | 15 |
| Structural steel..... | 4 | 6 | 10 | Brick and stone.... | 15 | 25 | 40 |

**VALUES OF ENDURANCE LIMIT.**—Moore and Kommers, and Moore and Jasper (*Bull.* 124 and 126, Engg. Exp. Sta., Univ. of Ill., 1921, 1923) reach a number of conclusions concerning wrought ferrous metals. A summary of the more important of these follows:

1. It is considered as well established that for wrought ferrous metals there is an endurance limit stress below which they will withstand an indefinite number of repetitions of stress.
2. In reversed bending, a fairly definite relation exists between the endurance limit, the ultimate tensile strength, and the Brinell hardness number. The endurance limit is about 50% of the ultimate tensile strength, with values ranging between 40% and 60%. The endurance limit is between 200 and 250 times the Brinell hardness number.
3. No relation exists between endurance limit and impact test or repeated-impact test results.
4. The endurance limit in reversed torsion (shear) varies from 48 to 60% of the endurance limit in reversed bending.
5. The endurance limit in reversed axial stress (tension to compression) varies from 56 to 70% of the endurance limit under reversed bending with nearly all the values 60% or above. The reason for this low value is that practically all axial loads are slightly eccentric and cause bending. Irwin (*Proc.* A.S.T.M., vol. xxv, Pt. II, p. 53, 1925), with a specially designed machine, found the reversed axial stress to be practically the same as that in reversed bending.

A report of Research Committee on Fatigue of Metals (*Proc.* A.S.T.M., vol. xxx, Pt. I, p. 259, 1930) gives a concise statement of the state of knowledge at that date, which corroborates the above conclusions with the following changes or additions:

1. No general relation exists between yield point or proportional limit and endurance limit.
2. For cast ferrous metals the endurance limit is about 40% of the tensile strength.
3. For non-ferrous metals no definite relation exists between endurance limit and tensile strength, the values varying between 18 and 50%.
4. For cast ferrous metals endurance limit is about 26% of the modulus of rupture.
5. For wrought ferrous metals the endurance limit is about 250 times the Brinell number for numbers under 400. For Brinell numbers exceeding 400, the relation does not hold. For cast ferrous metals and for non-ferrous metals sufficient data are not available to permit judging the degree of correlation.

Table 3 gives the endurance limit and other properties of a number of ferrous metals. Table 6 gives similar information for non-ferrous metals. The values of the endurance limit for heat-treated steels vary according to the treatment, ranging from 29,300 to 68,000 lb. per sq. in. for a 0.49% carbon steel, from 50,000 to 108,000 lb. per sq. in. for a

**Table 3.—Properties of Ferrous Metals Tested under Repeated Stress**
(*Bull.* Engg. Exp. Sta., Univ. of Ill., Nos. 124, 136, 142, 152, 156, 164, 165)

| | Yield Point, Tension, lb. per sq. in. | Ultimate Strength, Tension, lb. per sq. in. | Endurance Limit, Reversed Bending, lb. per sq. in. | Yield Point, Shear, lb. per sq. in. | Endurance Limit Reversed Torsion, lb. per sq. in. | Yield Point, Compression, lb. per sq. in. | Brinell Hardness Number |
|---|---|---|---|---|---|---|---|
| Steel | | | | | | | |
| 0.18% C hot-rolled | 40,300 | 61,500 | 28,000 | ........ | ........ | ........ | ........ |
| 0.37% C normalized | 34,900 | 71,900 | 33,000 | 22,500 | 16,000 | 38,100 | 132 |
| 0.37% C sorbitic | 87,300 | 102,600 | 57,000 | 60,200 | 32,500 | 84,500 | 209 |
| 0.52% C normalized | 47,600 | 98,000 | 42,000 | 34,600 | 22,000 | 51,000 | 193 |
| 0.52% C sorbitic | 84,300 | 111,400 | 55,000 | 52,200 | 31,500 | 87,400 | 227 |
| 0.93% C pearlitic | 33,400 | 84,100 | 30,500 | 22,500 | 16,300 | 29,700 | 162 |
| 0.93% C sorbitic | 67,600 | 115,000 | 56,000 | 42,000 | 29,000 | 72,700 | 227 |
| 1.20% C normalized | 60,700 | 116,900 | 50,000 | 39,700 | 24,500 | 57,900 | 224 |
| 1.20% C sorbitic | 130,100 | 179,900 | 92,000 | 80,600 | 48,000 | 111,500 | 369 |
| Cyclops metal | 62,000 | 115,900 | 55,000 | ........ | 37,000 | ........ | 239 |
| Stainless steel | 36,300 | 112,200 | 48,000 | ........ | 26,000 | ........ | 208 |
| Cold-drawn screw stock | 55,200 | 86,800 | 41,000 | ........ | ........ | ........ | ........ |
| Mn-steel, 0.94% Mn, 0.34% C, 0.037% Ni | 42,100 | 83,400 | 37,000 | ........ | ........ | ........ | 154 |
| Mn-steel, 0.94% Mn, 0.34% C, 0.037% Ni, sorbitic | 63,750 | 97,500 | 48,000 | ........ | ........ | ........ | 198 |
| 0.35% C, 1.17% Mn, cast | 39,000 | 80,800 | 32,000 | ........ | ........ | ........ | 179 |
| 0.25% C, 0.68% Mn, cast | 26,700 | 67,200 | 27,000 | ........ | ........ | ........ | 119 |
| Cast and Wrought Iron | | | | | | | |
| 0.02% C, Armco | 19,000 | 42,400 | 26,000 | 13,600 | 12,700 | 20,600 | 69 |
| Wrought iron, longitudinal section | 24,000 | 46,900 | 23,000 | ........ | ........ | ........ | 105 |
| Cast-iron pipe, 1/2 in. thick | ........ | 26,200 | 12,000 | ........ | ........ | 96,000 | 162 |
| Cast iron, 1 in. thick | ........ | 31,600 | 10,500 | ........ | ........ | 111,000 | 148 |
| Cast iron, 3 1/2 in. thick | ........ | 25,300 | 9,000 | ........ | ........ | 85,000 | 132 |

**Table 4.—Effect of Defects on Endurance of Rolled or Forged Ferrous Metals**

| Type of Defect | Percentage Reduction in Endurance Limit Based on a Standard Specimen | Type of Defect | Percentage Reduction in Endurance Limit Based on a Standard Specimen |
|---|---|---|---|
| Groove, 9.85-in. radius | 0 | Notch, 1/2-in. wide, square shoulders | 51 |
| Groove, 1-in. radius | 0 | Notch, 90° V | 60 |
| Groove, 1/4-in. radius | 8 | | |

**Table 5.—Endurance Ratios for Ferrous Metals under Different Stress Conditions**

| Character of Stress | Ratio, Ult. Tens. Str. ÷ Endurance Limit, % | | |
|---|---|---|---|
| | Wrought metals | Cast steel | Cast iron, gray |
| Reversed bending | 40–55 | 42 | 33–46 |
| Reversed torsion | 21–37 | ...... | ...... |
| Axial tension, zero to a maximum | ...... | ...... | 49 |
| Axial compression, zero to a maximum | ...... | ...... | 59 |
| Axial tension to axial compression | 24–40* | ...... | ...... |

* With precise loading devices this value has been found to be practically the same as in reversed bending, but such refinement in loading cannot be obtained in practice.

1.20% carbon steel, and from 54,000 to 120,000 lb. per sq. in. for a 3.50% nickel steel.

Table 5 summarizes the endurance ratios (ratio of ultimate tensile strength to endurance limit) for different types of stress. Moore and Kommers (Fatigue of Metals, McGraw-Hill, 1927, p. 147) state that the ratio of endurance limits in reversed torsion and reversed bending averages: Plain carbon steels, 0.55; alloy steels, 0.58; non-ferrous metals, 0.52.

**EFFECT OF SCRATCHES AND DEFECTS ON ENDURANCE LIMIT OF ROLLED OR FORGED FERROUS METALS.**—The beginning of rupture under repeated stress is supposed to be at a point of discontinuity in the material. The same principle applies to a discontinuity on the outer surface where, in bending or torsion, the stress is greatest. The presence of scratches, sharp corners, or grooves reduces the endurance limit an appreciable amount. Moore and Kommers (*Bull.* Engg. Exp. Sta., Univ. of Ill., No. 124) obtained the following reductions in the endurance limit, using steel of 0.02% C and 0.49% C, with results the same for each steel. The experiments were made on bars 0.40 in. diam., turned down to 0.275 in. diam. at bottom of the notch.

Surface finish has an appreciable effect on endurance limit. The standard finish used in the experiments of Moore and Kommers was obtained by polishing a smooth turned specimen with No. 0 and 00 emery cloth. Unpolished smooth-turned specimens had an endurance limit 8 to 12% lower than those that were polished; a rough-turned finish gave a slightly lower value than the smooth-turned finish. For cast-iron specimens having a 60° groove with a bottom radius of 5/64 in., the reduction in the endurance limit was only 8%. The effect of scratches, grooves, and external discontinuities is not so great in cast iron as in steel.

According to the theory of elasticity, a small hole, of diameter less than 1/6 the diameter of the specimen, should cause a stress at the edge of the hole three times the stress based on a uniform stress distribution, and therefore should reduce the endurance limit to 33% of that which would be obtained on a solid specimen. Tests on rotating beams of various steels with a small hole drilled through a diameter gave endurance limits from 50 to 65% of those for the solid beams, showing that theoretically high stresses do not produce corresponding failures, probably due to readjustments in the ductile metal.

**EFFECT OF RANGE OF STRESS.**—The endurance limit as determined by a rotating beam is the endurance limit for a complete reversal of stress from tension to compression, and *vice versa.* The endurance limit is less for a complete reversal than it is for those stress combinations where the stress goes only from a maximum to zero, or from a maximum to a minimum which is above zero. Howell (*Bull.* 136, Engg. Exp Sta., Univ. of Ill.), proposed the empirical relation $s = s_1(r + 3)/2$, where $s$ = endurance limit for the range of stress $r$; $s_1$ = endurance limit for completely reversed stress of the same character as $s$, *i.e.*, direct stress, bending, torsion, etc.; $r$ = ratio of the minimum stress to the maximum stress for a cycle of stress; for instance, if stress varies from 6000 lb. per sq. in. compression to 10,000 lb. per sq. in. tension, the ratio $r$ would be $-6000/+10{,}000 = 0.60$. Using the above notation, a formula was suggested by Moore, Kommers and Jasper (*Proc.* A.S.T.M., vol. xxii, Pt. II, 1922) where $s = 3s_1/(2 - r)$. This gives values for forged or rolled iron and steel which accord fairly well with tests, being on the safe side. For a complete digest of data, see Report of Research Committee on Fatigue of Metals, *Proc.* A.S.T.M., vol. xxx, Pt. I, p. 259, 1930.

### Table 6.—Properties of Non-ferrous Metals and Alloys under Repeated Stress

| Material | Proportional Limit, Tension, lb. per sq. in. | Ultimate Tensile Strength, lb. per sq. in. | Endurance Limit, Reversed Bending, lb. per sq. in. | Endurance Limit, Reversed Torsion, lb. per sq. in. | Brinell Hardness Number, 3000 kg. | Authority † |
|---|---|---|---|---|---|---|
| Aluminum | | | | | | |
| Rolled | 11,275 | 22,600 | 10,500 | ........ | 45* | 1 |
| Duralumin | | | | | | |
| As received (rolled, heated to 930° F. quenched in boiling water) | 25,030 | 50,960 | 14,000 | ........ | 100* | 2 |
| Tempered, as-received bars heated to 925° F., 30 min., quenched in boiling water 2 hr. | 18,550 | 51,170 | 12,000 | ........ | 100* | 2 |
| Annealed, as-received bars heated to 700° F., 20 min., cooled in furnace | 6,800 | 25,250 | 10,860 | ........ | 50* | 2 |
| Magnesium | | | | | | |
| Extruded | 1,224 | 32,490 | 7,800 | ........ | 41* | 1 |
| +4% Al, extruded as 3/4-in. round rods from 2 15/16-in. ingots at 350–400° F. | 8,125 | 35,160 | 12,000 | ........ | 52 | 6 |
| +4% Al, +0.25% Mn, extruded as above | 13,500 | 39,010 | 15,000 | ........ | 58 | 6 |
| +6.5% Al, extruded as above | 12,000 | 41,330 | 13,000 | ........ | 61 | 6 |
| +6.5% Al, +0.25% Mn, extruded as above | 15,250 | 44,380 | 15,000 | ........ | 65 | 6 |
| +10% Cu, extruded as above | 14,250 | 39,030 | 11,000 | ........ | 60 | 6 |
| Electron metal | | | | | | |
| Rolled, Mg +4% Zn | 6,260 | 36,500 | 17,000 | ........ | 64* | 2 |
| Nickel | | | | | | |
| Cold-rolled, annealed at 550° F. 60 min., cooled in furnace | 69,000 | 165,500 | 40,000 | 18,500 | ...... | 3 |
| Hot-rolled, annealed at 1400° F. 60 min., cooled in furnace | 25,100 | 76,200 | 31,500 | ........ | ...... | 3 |
| Hot-rolled to 5/8 in. round and drawn to 9/16 in. at 2000° F., drawn to 1/2 in. round at 1600° F., annealed at 1600° F., and cooled in furnace | 11,300 | 69,900 | 28,000 | ........ | 90 | 4 |
| Monel metal | | | | | | |
| As received, treatment unknown | 50,700 | 89,600 | 32,000 | ........ | 163 | 4 |
| Hot-rolled to 1 1/8 in. round | 49,600 | 89,800 | 32,000 | ........ | 169 | 4 |
| Constantan | | | | | | |
| Hot-rolled, tested as received | ........ | 70,500 | 34,500 | 14,500 | ...... | 3 |
| German silver | | | | | | |
| 11% Ni, 60% Cu, 29% Zn, cold-drawn, tested as received | 12,500 | 58,700 | 17,000 | ........ | ...... | 3 |
| 18% Ni, 65% Cu, 17% Zn, cold-drawn, tested as received | 20,300 | 62,400 | 22,000 | ........ | ...... | 3 |
| Copper | | | | | | |
| Cold-drawn, annealed bar cold-drawn to 1/2 in. round and left hard | 38,400 | 56,200 | 10,000 | ........ | 104 | 4 |
| Annealed, extruded to 1 in. round at 1380° F., cold-drawn to 7/8 in. and annealed at 1290° F. 30 min., cold-drawn to 3/4 in. and annealed at 1290° F. 30 min | 3,200 | 32,400 | 10,000 | ........ | 47 | 4 |
| Brass | | | | | | |
| 60–40, cold drawn, annealed bar cold-drawn to 1/2 in. round and left hard | 43,200 | 96,700 | 26,000 | ........ | 179 | 4 |
| 60–40 annealed, extruded to 1 in. round at 1290° F., and annealed at 1020° F. 30 min., drawn to 3/4 in. and annealed at 1020° F. | 15,600 | 54,200 | 22,000 | ........ | 72 | 4 |
| Naval, 61–38, rolled, tested as received | 33,400 | 68,200 | 21,000 | ........ | 13.5* | 1 |
| 70–30, cold-rolled | ........ | 73,200 | 17,500 | ........ | ...... | 5 |
| 70–30, cold-rolled, annealed 400° F. 90 min., cooled in furnace | 26,000 | 73,000 | 20,000 | ........ | ...... | 5 |
| 70–30, cold-rolled, annealed 1200° F. 60 min., cooled in furnace | ........ | 45,000 | 15,000 | ........ | ...... | 5 |
| 81–19, cold drawn, tested as received | 21,000 | 76,500 | 23,000 | ........ | ...... | 5 |
| 81–19, cold-drawn, annealed 450° F. 60 min., cooled in furnace | 27,700 | 80,500 | 26,000 | ........ | ...... | 5 |
| 81–19, cold-drawn, annealed 1000° F. 60 min., cooled in furnace | 7,500 | 44,000 | 17,500 | ........ | ...... | 5 |
| Muntz metal | | | | | | |
| Hot-rolled, quenched in iced brine and drawn at 840° F. | 20,000 | 65,600 | 18,000 | ........ | ...... | 3 |

**Table 6.—Properties of Non-ferrous Metals and Alloys under Repeated Stress**—*Continued*

| Material | Proportional Limit, Tension, lb. per sq. in. | Ultimate Tensile Strength, lb. per sq. in. | Endurance Limit, Reversed Bending, lb. per sq. in. | Endurance Limit, Reversed Torsion, lb. per sq. in. | Brinell Hardness Number 3000 kg. | Authority † |
|---|---|---|---|---|---|---|
| Bronze | | | | | | |
| 95–5, cold-rolled, tested as received | 23,000 | 55,800 | 22,500 | ........ | ...... | 5 |
| 95–5, cold-rolled, annealed 1200° F. 60 min., cooled in furnace | 11,500 | 48,100 | 22,500 | ........ | ...... | 5 |
| 89–11, cold-rolled, tested as received | 38,000 | 82,800 | 27,000 | ........ | ...... | 5 |
| 89–11, cold-rolled, annealed at 1100° F. 60 min., cooled in furnace | 24,500 | 67,600 | 27,000 | ........ | ...... | 5 |
| Mn–Bronze | | | | | | |
| Hot-rolled, annealed 1200° F. 60 min., cooled in furnace | ........ | 67,000 | 16,500 | ........ | ...... | 3 |
| 57% Cu, 41% Zn, cast | 13,000 | 70,000 | 15,000 | ........ | 93* | 2 |
| Al-Bronze | | | | | | |
| 90% Cu, 10% Al, extruded to 1 9/32 in. diam., drawn to 1 1/4 in. diam., quenched in water from 1650° F., annealed 1150° F. 30 min., cooled in furnace | 16,890 | 77,530 | 34,000 | ........ | 128 | 6 |
| 90% Cu, 10% Al, cast | 5,080 | 59,320 | 22,000 | ........ | 96 | 6 |
| 90° Cu, 10% Al, cast, quenched from 1650° F., annealed 1200° F. 30 min., cooled in furnace | 24,890 | 77,750 | 26,000 | ........ | 142 | 6 |

* 500 kg.

† Authorities: (1) R. R. Moore, *Proc.* A.S.T.M., vol. xxv, Pt. II, 1925; (2) R. R. Moore, *Proc.* A.S.T.M., vol. xxiii, Pt. II, 1923; (3) D. J. McAdam, *Trans.* A.S.S.T., vol. vii, 1925; (4) H. F. Moore and T. M. Jasper, *Bull.* No. 152, Engg. Exp. Sta., Univ. of Ill., 1925; (5) D. J. McAdam. *Trans.* A.S.S.T., vol. viii, 1925; (6) R. R. Moore, *Proc.* A.S.T.M., vol. xxiv, Pt. II, 1924.

**EFFECT OF CORROSION.**—If a part is subjected to repeated loads and at the same time to corrosive agencies, the endurance limit may be greatly reduced from its normal value as determined in air. McAdam at the U. S. Naval Engg. Exp. Sta., Annapolis (*Proc.*, A.S.T.M., vol. xxvi, Pt. II, p. 224, 1926, the first of many papers), has experimented on various steels with a number of corrosive agents. He found that the endurance limit may be reduced to one-third its normal value. Even so mild an agent as a stream of fresh water may reduce the endurance limit one-half, and salt water or water containing other salts in solution is very destructive. Practically all steels are affected, the corrosion-resistant ones less so than ordinary steels. Steam at 220 lb. per sq. in., and 700° F., without appreciable amounts of oxygen or liquid water did not greatly affect chromium-iron or nickel-steels, but when these steels were tested in air under a jet of steam, their strengths were reduced (Fuller, *Tech Pub.* No. 294, A.I.M. & M.E., and *Met. Prog.*, July, 1931, p. 79).

In *corrosion-fatigue*, the stress need not be high enough to have started fatigue cracks without corrosion. The protective film of oxide breaks down under alternating stress and allows the formation of corrosion pits which progress more rapidly under stress than without it. The pits cause a high concentration of stress at the bottom, and this cycle of the pit increasing the stress and the stress increasing the depth of the pit continues; when the stress has been raised beyond the endurance limit, a fatigue crack starts. It has not yet (1937) been possible to obtain "corrosion-fatigue" values which can be used directly in design.

The most effective method of protection is to keep from contact with corrosive agents, the parts subjected to repeated loads. Where this is not possible, a protective coating of some elastic material, such as varnish, cement, or enamel, will aid. Certain substances called inhibitors, such as sodium chromate, introduced into the corrosive stream or applied in some other way, have been found to help in restoring the protective film.

## 6. ENERGY OR DYNAMIC LOADS

**CLASSIFICATION OF LOADS.**—A **Static Load** is one that increases gradually from zero to its full intensity; its effect on a bar has been considered above.

A **Suddenly Applied Load** is one that is released at the instant of contact with the bar, and then acts at its full intensity from the beginning to the end of the deformation.

A **Moving Load,** is one that is moving when it is brought to bear on the bar. The effect of such a load which is denoted by the word *impact*, is much greater than that of a static or a suddenly applied load.

Suddenly applied and moving loads often are termed *dynamic loads* or *energy loads.*

**RELATION BETWEEN STORED ENERGY AND SIZE OF MEMBER.**—The external work (see p. 7–07) performed in deforming a bar within the proportional limit is $1/_2\, s^2\, Al/E$, where $s$ = unit stress; $A$ = cross-sectional area of bar; $l$ = length of bar; $E$ = modulus of elasticity. This also is the energy stored in the bar. Since the energy absorbed varies directly as the square of the unit stress, it follows that large amounts of energy will be absorbed by those portions where the stress is high. It is important that where high stresses occur they should be present over as great a length of the member as possible. If one part of the rod must be of small diameter, it is advisable that as great a length of the rod as possible be of same small diameter; otherwise the high concentration of energy in a small portion of the member may cause failure. See Seely, Resistance of Materials, 2nd Ed., p. 271.

**EFFECT OF SUDDEN LOADS.**—If a sudden load $P$ be applied to a bar, it will cause deformation $\epsilon l$, and the work done by the load will be $P\epsilon l$. Since the external work equals the internal work, $P\epsilon l = s^2Al/2E$, and since $\epsilon = s/E$, $P = sA/2$, or $s = 2P/A$. The unit stress, and also the unit deformation, is double that under an equal load applied gradually. However, the bar does not maintain equilibrium at the point of maximum stress and deformation, but, after a series of oscillations in which the surplus energy is dissipated in heat, it finally comes to rest with the same deformation and stress as that due to the equal static load.

**STRESS DUE TO LIVE LOADS.**—In structural design two loads are considered, the dead load or weight of the structure and the live load or superimposed loads to be carried. The stresses due to the dead load and to the live loads respectively are computed separately, regarding each as a static load. It is obvious that the stress due to the live load may be greatly increased, depending on the suddenness with which the load is applied. It has been shown above that the stress due to a suddenly applied load is double the stress caused by a static load. The term *coefficient of impact* is used extensively in structural engineering to denote the number by which the computed static stress is multiplied to obtain the value of the increased stress assumed to be caused by the suddenness of application of the live load. If $s$ = static unit stress computed from the live load, and $i$ = coefficient of impact, then the increase of unit stress due to sudden loading is $is$, and the total unit stress due to the live load is $s + is$. The value of $i$ has been determined by empirical methods and varies according to different conditions. In the building codes of most cities, specified floor loads for buildings include the impact allowance, and no increase is needed for live loads except for special cases of vibration or other unusual conditions. For railroad bridges, the value of $i$ depends upon the proportion of the length of the bridge which is loaded. No increase in the static stress is needed when the mass of the structure, as in monolithic concrete, is great. For machinery and for unusual conditions, such as elevator machinery and its supports, each structure should be considered by itself and the coefficient assumed accordingly. It should be noted that the meaning of the word impact used above differs somewhat from its strict theoretical meaning and as it is used in the next paragraph. The use of the terms impact and coefficient of impact in connection with live load stresses is, however, very general.

**AXIAL IMPACT ON BARS.**—A load $P$ dropped from a height $h$ on to the end of a vertical bar of cross-sectional area $A$, rigidly secured at the bottom end, produces in the bar a unit stress which increases from 0 up to $s'$, with a corresponding total deformation increasing from 0 up to $e_1$. The work done on the bar is $P\,(h + e_1)$ which, provided no energy is expended as heat or in giving velocity to the bar, is equal to the energy $1/_2\, s'\, Ae_1$ stored in the bar; that is,

$$P\,(h + e_1) = 1/_2\, s'\, Ae_1 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [15]$$

If $e$ = deformation produced by a static load $P$, then within the proportional limit

$$e_1/e = s'/(P/A) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [16]$$

Combining this with equation [15] gives

$$s' = s + s\sqrt{1 + (2\,h/e)} \quad . \; . \; . \quad [17]; \qquad e_1 = e + e\sqrt{1 + (2\,h/e)} \quad . \; . \quad [18]$$

A wrought-iron bar 1 in. square and 5 ft. long, under a static load of 5000 lb., will be shortened about 0.012 in., assuming no lateral flexure to occur; but, if a weight of 5000 lb. drops on its end from a height of 0.048 in., a stress of 20,000 lb. will be produced.

Formulas [17] and [18] give values of stress and deformation that are somewhat large, because part of the energy of the applied force is not effective in producing stress, but is expended in overcoming the inertia of the bar and in producing local stresses. For light bars, they give approximately correct results.

If the bar is horizontal and is struck at one end by a weight $P$, moving with a velocity $V$, the deformation produced is $e_1$. Then, as before, $\frac{1}{2} s' A e_1 = Ph$. In this case $h = V^2/2g$ = height from which $P$ would have to fall to acquire velocity $V$. $g$ = acceleration due to gravity = 32.16 ft. per sec. per sec. Combining with equation [16],

$$s' = s\sqrt{(2h/e)} \quad . \quad . \quad . \quad . \quad . \quad [19]; \qquad e_1 = e\sqrt{(2h/e)} \quad . \quad . \quad . \quad . \quad . \quad [20]$$

**IMPACT ON BEAMS.**—If a weight $P$ falls on a horizontal beam from a height $h$, producing a maximum deflection $y_1$ and a maximum unit stress $s'$ in the extreme fiber, the values of $s'$ and $y_1$ are given by

$$s' = s + s\sqrt{1 + 2h/y} \quad . \quad . \quad [21]; \qquad y_1 = y + y\sqrt{1 + 2h/y} \quad . \quad . \quad [22]$$

where $s$ = extreme fiber unit stress; $y$ = deflection due to $P$, considered as a static load. The value of $s$ may be obtained from the flexure formula (p. 7–19); that of $y$ from the proper formula for deflection under static load.

If a weight $P$ moving horizontally with a velocity $V$ strikes a beam, the ends of which are secured against horizontal movement, the maximum fiber unit stress and the maximum lateral deflection are given by

$$s' = s\sqrt{2h/y} \quad . \quad . \quad . \quad . \quad . \quad [23]; \qquad y_1 = y\sqrt{2h/y} \quad . \quad . \quad . \quad . \quad . \quad [24]$$

where $s$ and $y$ are as before, and $h$ is height through which $P$ would have to fall to acquire the velocity $V$. These formulas, like those for axial impact on bars, give results larger than those observed in tests, particularly if the weight of the beam is great. For further discussion, see Seely's Resistance of Materials, 2nd Ed., Chapter 11.

**RUPTURE FROM IMPACT.**—Rupture may be caused by impact provided the load has the requisite velocity. The above formulas, however, do not apply since they are valid only for stresses within the proportional limit. The only information available has been obtained through experiment. Hatt has determined from nearly 200 experiments that the work required for rupture from impact was about 30% greater than that required by static loads. From available test data, it does not seem that the ultimate elongation under impact (dynamic) loads is widely different from that under static loads. For the results of a large number of impact tests on cast iron, see *Proc.* A.S.T.M., vol. xxxiii, Pt. I, p. 87, 1933.

# BEAMS

## 1. THEORY OF FLEXURE

**TYPES OF BEAMS.**—A beam is a bar or structural member subjected to transverse loads that tend to bend it. Any structural member acts as a beam if bending is induced by external transverse forces.

**A Simple Beam** (Fig. 1*a*) is a horizontal member that rests on two supports at the ends of the beam. All parts between the supports have free movement in a vertical plane under the influence of vertical loads.

**A Fixed Beam, Constrained Beam, or Restrained Beam** (Fig. 1*b*) is one that is rigidly fixed at both ends or rigidly fixed at one end and simply supported at the other.

**A Continuous Beam** (Fig. 1*c*) is a member resting on more than two supports.

**A Cantilever Beam** (Fig. 1*d*) is a member with one end projecting beyond the point of support, free to move in a vertical plane under the influence of vertical loads placed between the free end and the support.

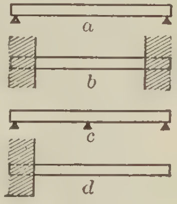

Fig. 1

**PHENOMENA OF FLEXURE.**—When a simple beam bends under its own weight, the fibers on the upper or concave side are shortened, and the stress acting on them is compression; the fibers on the under or convex side are lengthened and the stress acting on them is tension. In addition, shear exists along each cross-section, the intensity of which is greatest along the sections at the two supports and zero at the middle section.

When a cantilever beam bends under its own weight the fibers on the upper or convex side are lengthened under tensile stresses; the fibers on the under or concave side are shortened under compressive stresses; the shear is greatest along the section at the support, and zero at the free end.

**THE NEUTRAL SURFACE** is that horizontal section between the concave and convex surfaces of a loaded beam, where there is no change in the length of the fibers and no tensile or compressive stresses acting upon them.

**THE NEUTRAL AXIS** is the trace of the neutral surface on any cross-section of a beam. See Fig. 2.

**THE ELASTIC CURVE** of a beam is the curve formed by the intersection of the neutral surface with the side of the beam, it being assumed that the longitudinal stresses on the fibers are within the elastic limit.

**REACTIONS AT SUPPORTS.**—The reactions, or upward pressures at the points of support, are computed by applying the following conditions necessary for equilibrium of a system of vertical forces in the same plane: 1. The algebraic sum of all vertical forces must equal zero; *i.e.*, the sum of the reactions equals the sum of the downward loads. 2. The algebraic sum of the moments of all the vertical forces must equal zero. Condition 1 applies to cantilever beams, and to simple beams uniformly loaded, or with equal concentrated loads placed at equal distances from the center of the beam. In the cantilever beam, the reaction is the sum of all the vertical forces acting downward, comprising the weight of the beam and the superposed loads. In the simple beam each reaction is equal to one-half the total load, consisting of the weight of the beam and the superposed loads. Condition 2 applies to a simple beam not uniformly loaded. The reactions are computed separately, by determining the moment of the several loads about each support. The sum of the moments of the load around one support is equal to the moment of the reaction of the other support around the first support.

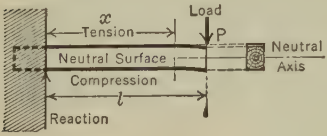

FIG. 2

EXAMPLE.—Determine reactions $R_1$, $R_2$ of a simple beam 20 ft. long, loaded as in Fig. 3. The weight of the beam may be taken as a concentrated load applied at the center of gravity of the beam. Then, moments around left-hand and right-hand supports are, respectively,

Left-hand, $(2000 \times 5) + (3000 \times 9) + (1200 \times 10) - (R_2 \times 20) = 0$; $R_2 = 2450$ lb.
Right-hand, $(R_1 \times 20) - (2000 \times 15) - (3000 \times 11) - (1200 \times 10) = 0$; $R_1 = 3750$ lb.
$R_1 + R_2 = 6200$ lb. = sum of all loads acting downward.

**CONDITIONS OF EQUILIBRIUM.**—The fundamental laws for the stresses at any cross-section of a beam in equilibrium are:

1. Sum of horizontal tensile stresses = sum of horizontal compressive stresses. 2. Resisting shear = vertical shear. 3. Resisting moment = bending moment.

**Vertical Shear.**—At any cross-section of a beam the resultant of the external vertical forces acting on one side of the section is equal and opposite to the resultant of the external vertical forces on the other side of the section. These forces tend to cause the beam to shear vertically along the section. The value of either resultant is known as the vertical shear at the section considered. It is computed by finding the algebraic sum of the vertical forces to the left of the section; that is, it is equal to the left reaction minus the sum of the vertical downward forces acting between the left support and the section.

**A Shear Diagram** is a graphic representation of the vertical shear at all cross-sections of the beam. Thus in the simple beam, Fig. 4, the ordinates to the line *AOB* represent to scale the intensity of the vertical shear at the corresponding sections of the beam. The vertical shear is greatest at the supports, where it is equal to the reactions, and it is zero at the center of the span. In the cantilever beam, Fig. 5, the vertical shear is greatest at the point of support where it is equal to the reaction, and it is zero at the free end. Fig. 6 shows graphically the vertical shear on all sections of a simple beam carrying two concentrated loads at equal distances from the supports, the weight of the beam being neglected.

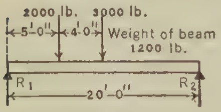

FIG. 3

**Resisting Shear.**—The tendency of a beam to shear vertically along any cross-section, due to the vertical shear, is opposed by an internal shearing stress at that cross-section known as the resisting shear; it is equal to the algebraic sum of the vertical components of all the internal stresses acting on the cross-section.

If $V$ = vertical shear, lb.; $V_r$ = resisting shear, lb.; $s_s$ = average unit shearing stress, lb. per sq. in.; and $A$ = area of the section, sq. in., then at any cross-section

$$V_r = V = s_sA;\ s_s = V/A \qquad [1]$$

The resisting shear is not uniformly distributed over the cross-section, but the intensity varies from zero at the extreme fiber to its maximum value at the neutral axis.

At any point in any cross-section the vertical unit shearing stress, lb. per sq. in., is

$$s_s = VA'c'/It \qquad [2]$$

where $V$ = total vertical shear, lb., for section considered; $A'$ = area, sq. in., of cross-section between a horizontal plane through point where shear is being found and the extreme fiber on the same side of the neutral axis; $c'$ = distance, in., from neutral axis to center of gravity of area $A'$; $I$ = moment of inertia of the section, in.$^4$; $t$ = width of section at plane of shear, in. Maximum value of the unit shearing stress, where $A$ = total area, sq. in., of cross-section of the beam, is

For a solid rectangular beam, $s_s = 3V/2A$ . . . . . . . . . . . . [3]

For a solid circular beam, $s_s = 4V/3A$ . . . . . . . . . . . . [4]

**Horizontal Shear.**—In a beam, at any cross-section where there is a vertical shearing force, there must be resultant unit shearing stresses acting on the vertical faces of particles which lie at that section. On a horizontal surface of such a particle, there is a unit shearing stress equal to the unit shearing stress on a vertical surface of the particle. Equation [2] therefore also gives the horizontal unit shearing stress at any point on the cross-section of a beam.

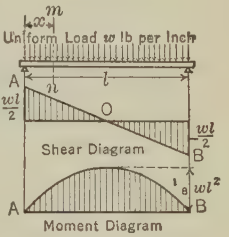

FIG. 4. Shear Diagram, Simple Beam

**Bending Moment,** at any cross-section of a beam, is the algebraic sum of the moments of the external forces acting on either side of the section. It is positive when it causes the beam to bend convex downward, hence causing compression in upper fibers and tension in lower fibers of beam. When the bending moment is determined from the forces that lie to the left of the section, it is positive if it acts in a clockwise direction; if determined from forces on the right side, it is positive if it acts in a counter-clockwise direction. If the moments of upward forces are given positive signs, and the moments of downward forces negative signs, the bending moment always will have the correct sign, whether determined from the right or left side. The bending moment should be determined for the side for which the calculation will be simplest.

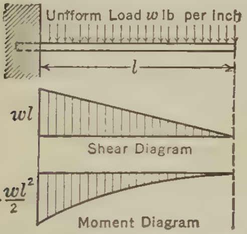

FIG. 5. Shear Diagram, Cantilever Beam

In Fig. 4 let $M$ be the bending moment, lb.-in., at a section $mn$ of a simple beam at a distance $x$, in. from the left support; $w$ = weight of beam per 1 in. of length; $l$ = length of the beam, in. Then the reactions are $\frac{1}{2}wl$, and $M = \frac{1}{2}wlx - \frac{1}{2}xwx$. For the sections at the supports, $x = 0$ or $l$ and $M = 0$. For the section at the center of the span $x = \frac{1}{2}l$ and $M = \frac{1}{8}wl^2 = \frac{1}{8}Wl$, where $W$ = total weight.

**A Moment Diagram** (Figs. 4, 5, and 6) shows the bending moment at all cross-sections of a beam. Ordinates to the curve represent to scale the moments at the corresponding cross-sections. The curve for a simple beam uniformly loaded is a parabola, showing $M = 0$ at the supports and $M = \frac{1}{8}wl^2 = \frac{1}{8}Wl$ at the center, $M$ being in lb.-in.

**The Dangerous Section** is the cross-section of a beam where the bending moment is greatest. In a cantilever beam it is at the point of support, regardless of the disposition of the loads. In a simple beam it is that section where the vertical shear changes from positive to negative, and may be located graphically by constructing a shear diagram, or numerically by taking the left reaction and subtracting the loads in order from the left until a point is reached where the sum of the loads subtracted equals the reaction. For a simple beam, uniformly loaded, the dangerous section is at the center of the span.

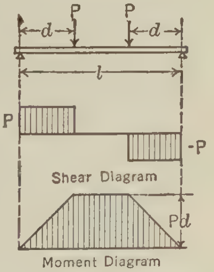

FIG. 6. Shear Diagram, Beam with Two Concentrated Loads

**Resisting Moment.**—The tendency to rotate about a point in any cross-section of a beam is due to the bending moment at that section; this tendency is resisted by the resisting moment which is the algebraic sum of the moments of all the horizontal stresses with reference to the same point.

**FORMULA FOR FLEXURE.**—Let $M$ = bending moment; $M_r$ = resisting moment of the horizontal fiber stresses; $s$ = unit stress (tensile or compressive) on any fiber, usually that one most remote from the neutral surface; $c$ = distance of that fiber from the neutral surface. Then

$M = M_r = sI/c$ . . . . [5]; $s = Mc/I$ . . . . . . . . . [6]

where $I$ = moment of inertia of the cross-section with respect to its neutral axis. If $s$ is in lb. per sq. in., then $M$ must be in lb.-in., $I$ in in.$^4$, and $c$ in in.

Formula [6] is the basis of the design and investigation of beams. It is true only when the maximum horizontal fiber stress $s$ does not exceed the proportional limit of the material.

### Table 1.—Elements of Sections

$A$ = area of section.

$I$ = moment of inertia about axis $I$–$I$.

$c$ = distance from axis $I$–$I$ to remotest point of section.

$I/c$ = section modulus.

$r$ = radius of gyration.

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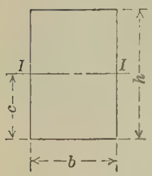
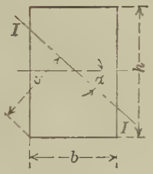
#### RECTANGLE
Axis through center



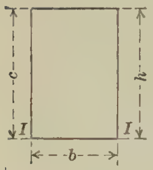
$$A = bh$$
$$c = h/2$$
$$I = bh^3/12$$
$$I/c = bh^2/6$$
$$r = h/\sqrt{12} = 0.289\,h$$

#### RECTANGLE
Axis any line through center of gravity



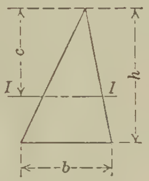
$$A = bh$$
$$c = (b\sin\alpha + h\cos\alpha)/2$$
$$I = bh(b^2\sin^2\alpha + h^2\cos^2\alpha)/12$$
$$I/c = \frac{bh(b^2\sin^2\alpha + h^2\cos^2\alpha)}{6(b\sin\alpha + h\cos\alpha)}$$
$$r = \sqrt{(b^2\sin^2\alpha + h^2\cos^2\alpha)/12}$$

#### RECTANGLE
Axis on base



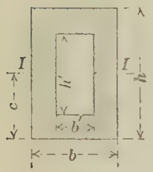
$$A = bh$$
$$c = h$$
$$I = bh^3/3$$
$$I/c = bh^2/3$$
$$r = h/\sqrt{3} = 0.577\,h$$

#### TRIANGLE
Axis through center of gravity



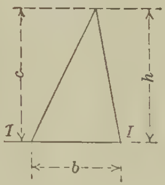
$$A = bh/2$$
$$c = \tfrac{2}{3}\,h$$
$$I = bh^3/36$$
$$I/c = bh^2/24$$
$$r = h/\sqrt{18} = 0.236\,h$$

#### HOLLOW RECTANGLE
Axis through center



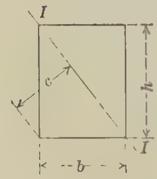
$$A = bh - b_1 h_1$$
$$c = h/2$$
$$I = (bh^3 - b_1 h_1^3)/12$$
$$I/c = (bh^3 - b_1 h_1^3)/6h$$
$$r = \sqrt{\frac{bh^3 - b_1 h_1^3}{12(bh - b_1 h_1)}}$$

#### TRIANGLE
Axis through base



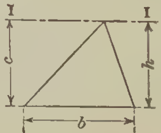
$$A = bh/2$$
$$c = h$$
$$I = bh^3/12$$
$$I/c = bh^2/12$$
$$r = h/\sqrt{6} = 0.408\,h$$

#### RECTANGLE
Axis on diagonal



$$A = bh$$
$$c = bh/\sqrt{b^2 + h^2}$$
$$I = b^3 h^3/6(b^2 + h^2)$$
$$I/c = b^2 h^2/6\sqrt{(b^2 + h^2)}$$
$$r = bh/\sqrt{6(b^2 + h^2)}$$

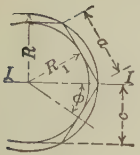
#### TRIANGLE
Axis through apex



$$A = bh/2$$
$$c = h$$
$$I = bh^3/4$$
$$I/c = bh^2/4$$
$$r = h/\sqrt{2} = 0.707\,h$$

Table 1.—Elements of Sections—(*Continued*)

### Equilateral Polygon



Axis through center, parallel to one side. $n$ = no. of sides

$A = nR_1^2 \tan \phi$

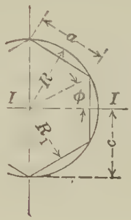
$c = a/2 \tan \phi = R_1$

$I = \{A(12\ R_1^2 + a^2)\}/48$

$I/c = \{A(12\ R_1^2 + a^2)\}/48\ R_1$

$r = \sqrt{(12\ R_1^2 + a^2)/48}$

### Equilateral Polygon



Axis through center, normal to side. $n$ = no. of sides.

$A = nR_1^2 \tan \phi$

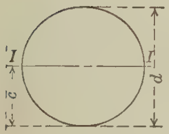
$c = a/(2 \sin \phi) = R$

$I = \{A(6R^2 - a^2)\}/24$

$I/c = \{A(6R^2 - a^2)\}/24\ R$

$r = \sqrt{(6R^2 - a^2)/24}$

### Circle



Axis through center

$A = \pi d^2/4 = 0.7854\ d^2$

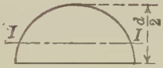
$c = d/2$

$I = \pi d^4/64 = 0.0491\ d^4$

$I/c = \pi d^3/32 = 0.0982\ d^3$

$r = d/4$

### Half Circle



Axis through center of gravity.

$A = \pi d^2/8 = 0.3927\ d^2$

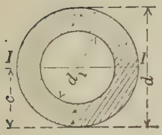
$c = \{d(3\pi - 4)\}/6\pi = 0.2878\ d.$

$I = \{d^4(9\pi^2 - 64)\}/1152\ \pi = 0.0068\ d^4$

$I/c = \dfrac{\{d^3(9\pi^2 - 64)\}}{\{192(3\pi - 4)\}} = 0.0238\ d^3$

$r = \{d\sqrt{(9\pi^2 - 64)}\}/12\pi = 0.1322\ d$

### Hollow Circle



Axis through center

$A = \pi(d^2 - d_1^2)/4 = 0.7854(d^2 - d_1^2)$

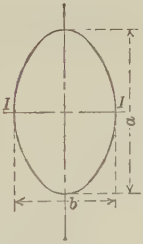
$c = d/2$

$I = \pi(d^4 - d_1^4)/64 = 0.0491(d^4 - d_1^4)$

$I/c = \pi(d^4 - d_1^4)/32\ d = 0.0982(d^4 - d_1^4)/d$

$r = \sqrt{(d^2 + d_1^2)}/4$

### Ellipse



Axis through center

$A = \pi ab/4 = 0.7854\ ab$

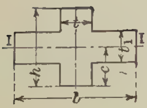
$c = a/2$

$I = \pi a^3\, b/64 = 0.0491\ a^3\, b$

$I/c = \pi a^2\, b/32 = 0.0982\ a^2\, b$

$r = a/4$

### Crossed Rectangles



Axis through center

$A = th + t_1(b - t)$

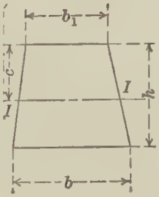
$c = h/2$

$I = \{th^3 + t_1^3(b - t)\}/12$

$I/c = \{th^3 + t_1^3(b - t)\}/6h$

$r = \sqrt{\dfrac{th^3 + t_1^3(b - t)}{12\,\{th + t_1(b - t)\}}}$

### Trapezoid



Axis through center of gravity.

$A = \{(b + b_1)h\}/2$

$c = \dfrac{\{(b_1 + 2b)h\}}{\{3(b + b_1)\}}$

$I = \dfrac{h^3(b^2 + 4\ bb_1 + b_1^2)}{36(b + b_1)}$

$I/c = \dfrac{h^2(b^2 + 4\ bb_1 + b_1^2)}{12(b_1 + 2b)}$

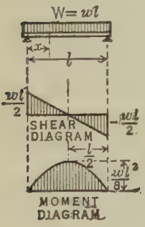
$r = \dfrac{h}{6(b + b_1)} \sqrt{2(b^2 + 4\ bb_1 + b_1^2}$

## Table 2.—Bending Moment, Vertical Shear, and Deflection of Beams of Uniform Cross-section under Various Conditions of Loading

$P$ = concentrated loads, lb.
$R_1, R_2$ = reactions, lb.
$w$ = uniform load per unit of length, lb. per in.
$W$ = total uniform load on beam, lb.
$l$ = length of beam, in.
$x$ = distance from support to any section, in.
$E$ = modulus of elasticity, lb. per sq. in.

$I$ = moment of inertia, in.$^4$
$V_x$ = vertical shear at any section, lb.
$V$ = maximum vertical shear, lb.
$M_x$ = bending moment at any section, lb.-in.
$M$ = maximum bending moment, lb.-in.
$y$ = maximum deflection, in.

### Simple Beam—Uniform Load



$$R_1 = R_2 = \frac{wl}{2}$$

$$V_x = \frac{wl}{2} - wx$$

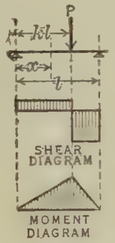
$$V = \pm\frac{wl}{2}\left(\text{when } \begin{cases} x = 0 \\ x = l \end{cases}\right)$$

$$M_x = \frac{wlx}{2} - \frac{wx^2}{2}$$

$$M = \frac{wl^2}{8}\left(\text{when } x = \frac{l}{2}\right)$$

$$y = \frac{5\,Wl^3}{384\,EI} \text{ (at center of span)}$$

### Simple Beam — Concentrated Load at Any Point

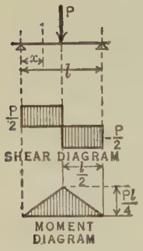


$R_1 = P(1 - k)$
$R_2 = Pk$
$V_x = R_1$ (when $x < kl$)
$= R_2$ (when $x > kl$)
$V = P(1 - k)$ (when $k < 0.5$)
$= -Pk$ (when $k > 0.5$)
$M_x = Px(1 - k)$ (when $x < kl$)
$= Pk(l - x)$ (when $x > kl$)
$M = Pkl(1 - k)$ (at point of load)

$$y = \frac{Pl^3}{3\,EI}(1 - k) \times (2/3\,k - 1/3\,k^2)^{3/2}$$

$$\left(\text{at } x = l\sqrt{2/3\,k - 1/3\,k^2}\right)$$

### Simple Beam—Concentrated Load at Center



$$R_1 = R_2 = \frac{P}{2}$$

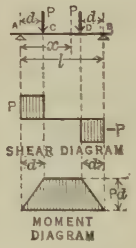
$$V_x = V = \pm\frac{P}{2}$$

$$M_x = \frac{Px}{2}$$

$$M = \frac{Pl}{4}\left(\text{when } x = \frac{l}{2}\right)$$

$$y = \frac{Pl^3}{48\,EI} \text{ (at center of span)}$$

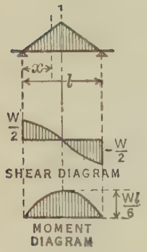
### Simple Beam—Two Equal Concentrated Loads at Equal Distances from Supports



$R_1 = R_2 = P$
$V_x = P$ for AC
$= 0$ for CD
$= -P$ for DB
$V = \pm P$
$M_x = Px$ for AC
$= Pd$ for CD
$= P(l - x)$ for DB
$M = Pd$

$$y = \frac{Pd}{24\,EI}(3\,l^2 - 4\,d^2) \text{ (at center of span)}$$

### Simple Beam—Load Increasing Uniformly from Supports to Center of Span



$$R_1 = R_2 = \frac{W}{2}$$

$$V_x = W\left(\frac{1}{2} - \frac{2x^2}{l^2}\right)\left(\text{when } x < \frac{l}{2}\right)$$

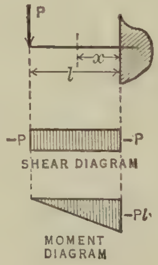
$$V = \pm\frac{W}{2} \text{ (at supports)}$$

$$M_x = Wx\left(\frac{1}{2} - \frac{2x^2}{3l^2}\right)$$

$$M = \frac{Wl}{6} \text{ (at center of span)}$$

$$y = \frac{Wl^3}{60\,EI} \text{ (at center of span)}$$

### Cantilever Beam — Load Concentrated at Free End



$$R = P$$

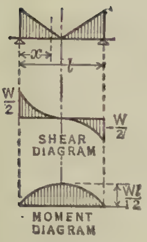
$$V_x = V = -P$$

$$M_x = -P(l - x)$$

$$M = -Pl \text{ (when } x = 0)$$

$$y = \frac{Pl^3}{3\,EI}$$

**Table 2.**—*Continued*



### Simple Beam — Load Increasing Uniformly from Center to Supports

$$R_1 = R_2 = \frac{W}{2}$$

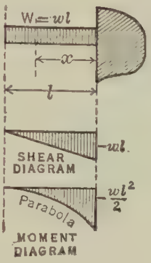
$$V_x = -W\left(\frac{2x}{l} - \frac{2x^2}{l^2} - \frac{1}{2}\right) \quad \left(\text{when } x < \frac{l}{2}\right)$$

$$V = \pm\frac{W}{2}$$

$$M_x = Wx\left(\frac{1}{2} - \frac{x}{l} + \frac{2}{3}\frac{x^2}{l^2}\right) \quad \left(\text{when } x < \frac{l}{2}\right)$$

$$M = \frac{Wl}{12} \text{ (at center of span)}$$

$$y = \frac{3}{320}\frac{Wl^3}{EI} \text{ (at center of span)}$$



### Cantilever Beam—Uniform Load

$$R = W = wl$$

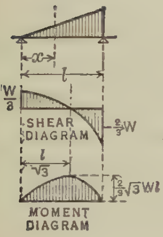
$$V_x = -w(l - x)$$

$$V = -wl \text{ (when } x = 0)$$

$$M_x = -w(l - x)\left(\frac{l - x}{2}\right)$$

$$M = -\frac{wl^2}{2} \text{ (when } x = 0)$$

$$y = \frac{Wl^3}{8EI}$$



### Simple Beam — Load Increasing Uniformly from One Support to the Other

$$R_1 = \frac{W}{3}; \quad R_2 = \frac{2}{3}W$$

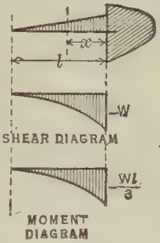
$$V_x = W\left(\frac{1}{3} - \frac{x^2}{l^2}\right)$$

$$V = -\frac{2}{3}W \text{ (when } x = l)$$

$$M_x = \frac{Wx}{3}\left(1 - \frac{x^2}{l^2}\right)$$

$$M = \frac{2}{9\sqrt{3}}Wl \quad \left(\text{when } x = \frac{l}{\sqrt{3}}\right)$$

$$y = \frac{0.01304}{EI}Wl^3$$



### Cantilever Beam—Load Increasing Uniformly from Free End to Support

$$R = W$$

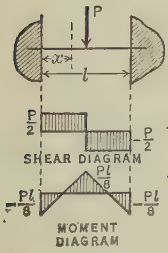
$$V_x = -W\frac{(l - x)^2}{l^2}$$

$$V = -W \text{ (when } x = 0)$$

$$M_x = -\frac{W}{3}\frac{(l - x)^3}{l^2}$$

$$M = -\frac{Wl}{3} \text{ (when } x = 0)$$

$$y = \frac{Wl^3}{15EI}$$



### Fixed Beam — Concentrated Load at Center of Span

$$R_1 = R_2 = \frac{P}{2}$$

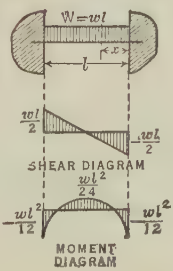
$$V_x = V = \pm\frac{P}{2}$$

$$M_x = P\left(\frac{x}{2} - \frac{l}{8}\right)$$

$$M_x = -\frac{Pl}{8}\left(\text{when } \begin{cases} x = 0 \\ x = l \end{cases}\right)$$

$$M = +\frac{Pl}{8} \text{ (at center of span)}$$

$$y = \frac{Wl^3}{192EI}$$



### Fixed Beam—Uniform Load

$$R_1 = R_2 = \frac{wl}{2} = \frac{W}{2}$$

$$V_x = \frac{wl}{2} - wx$$

$$V = \pm\frac{wl}{2} \text{ (at ends)}$$

$$M_x = -\frac{wl^2}{2}\left(\frac{1}{6} - \frac{x}{l} + \frac{x^2}{l^2}\right)$$

$$M = -{}^1/_{12}\, wl^2 \quad \left(\text{when } \begin{cases} x = 0 \\ x = l \end{cases}\right)$$

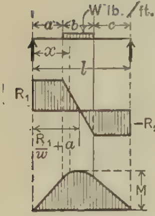
$$M = \frac{wl^2}{24}\left(\text{when } x = \frac{l}{2}\right)$$

$$y = \frac{Wl^3}{384EI}$$

*Table continued on following page*

Table 2.—*Concluded*

**Simple Beam—Distributed Load over Part of Beam**



$$R_1 = \frac{wb(2c+b)}{2l}$$

$$R_2 = \frac{wb(2a+b)}{2l}$$

$$V_x = \frac{wb(2c+b)}{2l} - w(x-a)$$

$$V = R_1 \text{ (when } a < c\text{)}$$

$$= R_2 \text{ (when } a > c\text{)}$$

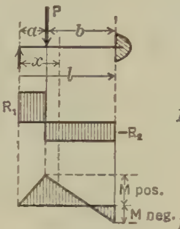
$$M_x = \frac{wbx(2c+b)}{2l} \text{ for AB}$$

$$= R_1x - \frac{w(x-a)^2}{2} \text{ for BC}$$

$$= R_2(l-x) \text{ for CD}$$

$$M = \frac{wb(2c+b)[4al+b(2c+b)]}{8l^2}$$

**Beam Supported at One End, Fixed at Other—Concentrated Load at Any Point**



$$R_1 = \frac{Pb^2(2l+a)}{2l^3}$$

$$R_2 = P - R_1$$

$$V_x = R_1 \text{ (when } x < a\text{)}$$

$$= R_2 \text{ (when } x > a\text{)}$$

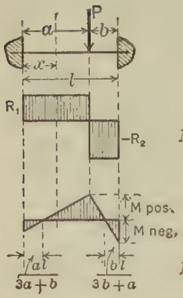
$$M_x = \frac{Pb^2\,x(2l+a)}{2l^3} \text{ (when } x < a\text{)}$$

$$= R_1x - P(x-a) \text{ (when } x > a\text{)}$$

$$M_{\text{positive}} = \frac{Pab^2(2l+a)}{2l^3} \text{ (when } x = a\text{)}$$

$$M_{\text{negative}} = -\frac{Pab(l+a)}{2l^2} \text{ (when } x = l\text{)}$$

**Fixed Beam—Concentrated Load at Any Point**



$$a > b$$

$$R_1 = Pb^2(l+2a)/l^3$$

$$R_2 = Pa^2(l+2b)/l^3$$

$$V_x = R_1 \text{ (when } x < a\text{)}$$

$$= R_2 \text{ (when } x > a\text{)}$$

$$V = R_2$$

$$M_x = R_1x - \frac{Pab^2}{l^2} \text{ (when } x < a\text{)}$$

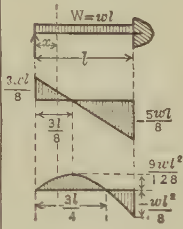
$$= R_2(l-x) - \frac{Pa^2b}{l^2} \text{ (when } x > a\text{)}$$

$$M_{\text{positive}} = \frac{2\,Pa^2b^2}{l^3}$$

$$M_{\text{negative}} = -\frac{Pa^2b}{l^2}$$

$$y = -\frac{2\,Pa^3b^2}{3\,EI(3a+b)^2}$$

**Beam Supported at One End, Fixed at Other—Distributed Load**



$$R_1 = \frac{3\,wl}{8}$$

$$R_2 = \frac{5\,wl}{8}$$

$$V_x = \frac{3\,wl}{8} - wx$$

$$V = \frac{3\,wl}{8} \text{ (at left support)}$$

$$= \frac{5\,wl}{8} \text{ (at right support)}$$

$$M_x = wx\left(\frac{3l}{8} - \frac{x}{2}\right)$$

$$M_{\text{positive}} = \frac{9\,wl^2}{128}$$

$$M_{\text{negative}} = -\frac{wl^2}{8}$$

$$y = -\frac{0.0054\,wl^4}{EI} \text{ (at 0.4215 } l \text{ from } R_1\text{)}$$

**MOMENT OF INERTIA** is the sum of the products of each elementary area of the cross-section multiplied by the square of the distance of that area from the assumed axis of rotation, or

$$I = \Sigma r^2\Delta A = \int r^2dA \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [7]$$

where $\Sigma$ is the sign of summation, $\Delta A$ = an elementary area of the section, and $r$ = distance of $\Delta A$ from the axis. The moment of inertia is greatest in those sections (such as I-beams) having much of the area concentrated at a distance from the axis. Unless otherwise stated the neutral axis is the axis of rotation considered. $I$ usually is expressed in in.$^4$ See Table 1 for values of moments of inertia of various sections.

**MODULUS OF RUPTURE** is the term applied to the value of $s$ as found by formula [6], when a beam is loaded to the point of rupture. Since formula [6] is true only for stresses within the proportional limit, the value $s$ of the rupture strength so found is incorrect. However, the equation is used, as a measure of the ultimate load-carrying capacity of a beam. The modulus of rupture does not show the actual stress in the extreme fiber of a beam; it is useful only as a basis of comparison. If the strength of a beam in tension differs from its strength in compression, the modulus of rupture is intermediate between the two.

**SECTION MODULUS** is the factor $I/c$ in flexure formula [5], and is expressed in in.$^3$ It is the measure of a capacity of a section to resist a bending moment. For values of $I/c$ for simple shapes, see Table 1. See pp. 17–50 to 17–72 for properties of standard steel structural sections, and p. 6–12 for properties of aluminum sections.

**ELASTIC DEFLECTION OF BEAMS.**—When a beam bends under load, all points of the elastic curve except those over the supports are deflected from their original positions. The radius of curvature $\rho$ of the elastic curve at any section is expressed as

$$\rho = EI/M \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

where $E$ = modulus of elasticity of the material, lb. per sq. in.; $I$ = moment of inertia, in.$^4$, of the cross-section with reference to its neutral axis; $M$ = bending moment, lb.-in., at the section considered. Where there is no bending moment, $\rho$ is infinity and the curve is a straight line; where $M$ is greatest, $\rho$ is smallest, and the curvature, therefore, is greatest.

If the elastic curve be referred to a system of co-ordinate axes in which $x$ represents horizontal distances, $y$ vertical distances, and $l$ distances along the curve, the value of $\rho$ is found, by the aid of the calculus, to be $d^3l/dx \cdot d^2y$. Substituting this value in the expression $\rho = EI/M$ and assuming that $dx$ and $dl$ are practically equal, there results differential equation [9] of the elastic curve which applies to all beams when the elastic limit of the material is not exceeded:

$$EI(d^2y/dx^2) = M \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

Equation [9] is used to determine the deflection of any point of the elastic curve, by regarding the point of support as the origin of the co-ordinate axis, taking $y$ as the vertical deflection at any point on the curve and $x$ as the horizontal distance from the support to the point considered. The values of $E$, $I$, and $M$ are substituted and the expression is integrated twice, giving proper values to the constants of integration, and the deflection $y$ is determined for any point. See Table 2.

For example, the cantilever beam shown in Fig. 2 has a length = $l$, inches, and carries a load $P$, pounds, at the free end. It is required to find the deflection of the elastic curve at a point distant $x$, inches, from the support, the weight of the beam being neglected.

The moment $M = -P(l - x)$; substituting in [9], the equation for the elastic curve becomes $EI(d^2y/dx^2) = -Pl + Px$. Integrating and determining the constant of integration by the condition that $dy/dx = 0$ when $x = 0$, there results $EI(dy/dx) = -Plx + {}^1/_2\,Px^2$. Integrating a second time and determining the constant by the condition that $x = 0$ when $y = 0$, there results $EIy = -{}^1/_2\,Plx^2 + {}^1/_6\,Px^3$, which is the equation of the elastic curve. When $x = l$, the value of $y$, or the deflection in inches at the free end, is found to be $-Pl^3/3\,EI$.

**DEFLECTION DUE TO SHEAR.**—The deflection of a beam as computed by the ordinary formulas is that due to flexural stresses only. In short beams the deflection due to vertical shear sometimes is appreciable and may need to be considered. Because of the non-uniform distribution of the shear over the cross-section of the beam, computing the deflection due to shear by exact methods is difficult. It may be approximated by $y_s = M/AE_s$, where $y_s$ = deflection, in., due to shear; $M$ = bending moment, lb.-in., at the section where the deflection is calculated; $E_s$ = modulus of elasticity in shear, lb. per sq. in.; $A$ = area of cross-section of beam, sq. in. (see Seely's Resistance of Materials, 2nd Ed., p. 136). Swain says (Strength of Materials, McGraw-Hill Book Co., p. 223) that for a rectangular section, the ratio of deflection due to shear, to deflection due to bending, will be less than 5% if the depth of the beam is less than ${}^1/_8$ of the length.

## 2. BEAMS OF UNIFORM CROSS-SECTION

**DESIGN PROCEDURE.**—In designing a beam the procedure is: 1. Compute reactions. 2. Determine position of the dangerous section and the bending moment at that section. 3. Divide the maximum bending moment (expressed in lb.-in.) by the allowable unit stress (expressed in lb. per sq. in.) to obtain the minimum value of the section modulus. 4. Select a beam section with a section modulus equal to or slightly greater than the section modulus required.

**WEB SHEAR.**—A beam designed in the above manner is safe against rupture of the extreme fibers due to bending in a vertical plane, and usually the cross-section will have sufficient area to sustain the shearing stresses with safety. For short beams carrying heavy loads, however, the vertical shear at the supports is large, and it may be necessary to increase the area of the section to keep the unit shearing stress within the limit allowed. For steel beams, the average unit shearing stress is computed by $s_s = V/A$, where $V$ = total vertical shear, lb.; $A$ = area of web, sq. in. For allowable average unit shearing stresses in cross-section of web of girders and rolled steel beams, see p. 17–47. For timber beams, see Horizontal Shear in Timber Beams, p. 7–26.

**MISCELLANEOUS CONSIDERATIONS.**—Other considerations which will influence the choice of section under certain conditions of loading are: 1. Maximum vertical deflection that may be permitted in beams coming in contact with plaster. 2. Danger of failure by sidewise bending in long beams, unbraced against lateral deflection. 3. Danger of failure by the buckling of the web of steel beams of short span carrying heavy loads. See p. 17–47. 4. Danger of failure by horizontal shear, particularly in wooden beams.

**VERTICAL DEFLECTION.**—If a beam is to support or come in contact with materials like plaster, which may be broken by excessive deflection, it is usual to select such a beam that the maximum deflection will not exceed (1/360 × span). It may be shown that for a simple beam, supported at the ends, with a total uniformly distributed load $W$, pounds, the deflection, inches, is

$$y = 30\ sL^2/Ed \quad \ldots\ldots\ldots\ldots [10]$$

where $s$ = allowable unit fiber stress, lb. per sq. in.; $L$ = span of beam, ft.; $E$ = modulus of elasticity, lb. per sq. in.; $d$ = depth of beam, in.

If the deflection of a steel beam is to be less than $^1/_{360}$ of the span, it may be shown from equation [10] that, for a maximum allowable fiber stress of 18,000 lb. per sq. in., the limit of span, in feet, is approximately 1.8 $d$, where $d$ = depth of the beam, in.

For the deflection due to the impact of a moving load falling on a beam, see p. 7-15.

**LATERAL DEFLECTION, STEEL BEAMS.**—See p. 17–48.

**BUCKLING OF THE WEB, STEEL BEAMS.**—See p. 17–47.

**HORIZONTAL SHEAR IN TIMBER BEAMS.**—See Horizontal Shear, p. 7–19. In beams of a homogeneous material which can withstand equally well shearing stresses in any direction, vertical and horizontal shearing stresses are equally important. In timber, however, shearing strength along the grain is much less than that perpendicular to the grain. Hence, the beams may fail owing to horizontal shear. Short wooden beams should be checked for horizontal shear in order that allowable unit shearing stress along the grain shall not be exceeded. See Example 1, below.

**RESTRAINED BEAMS.**—A beam is considered to be restrained if one or both ends are not free to rotate. This condition exists if a beam is built into a masonry wall at one or both ends, if it is riveted or otherwise fastened to a column, or if the ends projecting beyond the supports carry loads which tend to prevent tilting of the ends which would naturally occur as the beam deflects. The shears and moments given in Table 2 for fixed end conditions are seldom, if ever, attained, since the restraining elements themselves deform and reduce the magnitude of the restraint. This reduction of restraint decreases the negative moment at the support and increases the positive moment in the central portion of the span. The amount of restraint which exists is a matter which must be judged for each case in the light of the construction used, the rigidity of the connections, and the relative sizes of the connecting members.

**SAFE LOADS ON SIMPLE BEAMS.**—Formula [11] gives the safe loads on simple beams. This formula is obtained by substituting in the flexure formula [5], the value of $M$ for a simple beam, uniformly loaded, as given in Table 2. Let $W$ = total load, lb.; $s$ = extreme fiber unit stress, lb. per sq. in.; $S$ = section modulus, in.$^3$; $L$ = length of span, ft. Then

$$W = {}^2/_3\, s(S/L) \quad \ldots\ldots\ldots\ldots [11]$$

If $s$ is taken as a maximum allowable unit fiber stress, this equation gives the maximum allowable load on the beam. Most building codes permit a value of $s$ = 18,000 lb. per sq. in. for quiescent loads on steel. For this value of $s$, equation [11] becomes

$$W = 12{,}000\ S/L \quad \ldots\ldots\ldots\ldots [12]$$

See p. 17–50 for tables of safe loads on structural steel beams, based on equation [12].

If the load is concentrated at the center of the span, the safe load is one-half the value given by equation [12]. If the load is neither uniformly distributed nor concentrated at the center of the span, the maximum bending moment must be used. The foregoing equations are for beams laterally supported and are for flexure only. The other factors which influence the strength of the beam, as shearing, buckling, etc., also must be considered. See pp. 17–47 to 17–49.

**USE OF TABLES IN DESIGN.**—For the use of tables of safe loads, and of properties of rolled structural steel sections, see pages 17–50 to 17–72. Examples of the use of these tables in the design of steel beams are there given. The following is an example in the use of tables for the design of a wooden beam.

EXAMPLE 1.—Design a southern pine girder, of common structural grade, to carry a load of 9600 lb. uniformly distributed over a 16-ft. span in the interior of a building, the beam being a simple beam, freely supported at each end.

*Solution.*—From Table 2, the bending moment of a simple beam uniformly loaded is $M = wl^2/8$. Since $W = wl$ and $L = 12\ l$,

$$W = 9600 \times 16 \times 12/8 = 230{,}400 \text{ lb.-in.}$$

From Table 3, the allowable unit stress on yellow pine is 1200 lb. per sq. in., whence

$$I/c = 230{,}400/1200 = 192 \text{ in.}^3$$

From Table 1, the section modulus of a rectangular section is $bd^2/6$. Assume $b = 8$ in. Then $8\,d^2/6 = 192$, and $d = \sqrt{144} = 12.0$ in. A beam 8 × 12 in. is selected tentatively, and checked for shear.

Maximum shearing stress (horizontal and vertical) is at the neutral surface over the supports. Formula [3] for horizontal shear in a solid rectangular beam is $s_s = 3\,V/2A$; $V = 9600/2 = 4800$, and $A = 8 \times 12 = 96$, whence $s_s = (3 \times 4800)/(2 \times 96) = 75$ lb. per sq. in.

From Table 4, the safe horizontal unit shearing stress for common grade southern yellow pine is 88 lb. per sq. in. Since the actual horizontal unit shearing stress is less than 88 lb., the 8 × 12 in. beam will be satisfactory.

**STRENGTH OF CAST-IRON BEAMS.**—(C. H. Benjamin, *Machy.*, May, 1906.)—Numerous tests were made of beams of different sections including hollow rectangles and cylinders, I- and T-shapes, etc. All the sections were made approximately the same area, about 4.4 sq. in., and all were tested by transverse loading, with supports 18 in. apart. The results, when reduced by the ordinary formula for stress on the extreme fiber, showed an extraordinary variation, some of the values being as follows: Square bar, 23,300; Round bar, 25,000. Hollow round, 3.4 in. outside and 2.5 in. inside diam., 26,450, and 35,800. Hollow ellipse, 3 in. wide, 3.9 in. high, 0.9 in. thick, 36,000. I-beam, 4 in. high, web 0.44 in. thick, 17,700. The hollow cylindrical and elliptical sections are much stronger than the solid sections. This is due to the thinner metal, the greater surface of hard skin, and freedom from shrinkage strains. Professor Benjamin's conclusions from these tests are:

1. The commonly accepted formulas for the strength and stiffness of beams do not apply well to cored and ribbed sections of cast-iron.
2. Neither the strength nor the stiffness of a section increases in proportion to the increase in the section modulus or the moment of inertia.
3. The best way to determine these qualities for a cast-iron beam is by experiment with the particular section desired and not by reasoning from any other section.

## 3. BEAMS OF UNIFORM STRENGTH

**A BEAM OF UNIFORM STRENGTH** is one in which the dimensions are such that the maximum fiber stress $s$ is the same throughout the length of the beam. The form of the beam is determined by finding the areas of various cross-sections from the flexure formula $M = sI/c$, keeping $s$ constant and making $I/c$ vary with $M$. For a rectangular section of width $b$ and depth $d$, the section modulus $I/c = {}^1/_6\,bd^2$ and, therefore, $M = {}^1/_6\,sbd^2$. By making $bd^2$ vary with $M$, the dimensions of the various sections are obtained. Table 5 gives the dimensions $b$ and $d$, at any section, the maximum unit fiber stress $s$, and the maximum deflection $y$, of some rectangular beams of uniform strength. In this table, the bending moment has been assumed to be the controlling factor. On account of the vertical shear near the ends of the beams, the area of the sections must be increased over that given by an amount necessary to keep the unit shearing stress within the allowable unit shearing stress. The discussion of beams of uniform strength, though of considerable theoretical interest, is of little practical value since the cost of fabrication will offset any economy in the use of the material. A plate girder in a bridge or a building is an approximation in practice to a steel beam of uniform strength.

## 4. CURVED BEAMS

The derivation of the flexure formula, $s = Mc/I$, assumes that the beam is initially straight; therefore, any deviation from this condition introduces an error in the value of the stress. If the curvature is slight the error involved is not large, but in beams with a large amount of curvature, as hooks, chain links, frames of punch presses, etc., the error involved in the use of the ordinary flexure formula is considerable. The effect of the curvature is to increase the stress in the inside and to decrease it on the outside fibers of the beam and to shift the position of the neutral axis from the centroidal axis toward the concave or inner side.

The correct value for the unit fiber stress may be found by introducing a correction factor in the flexure formula, *viz.*, $s = KMc/I$; the factor $K$ depends on the shape of the beam and on the ratio $R/c$, where $R$ = distance, inches, from the centroidal axis of the section to the center of curvature of the central axis of the unstressed beam; and $c$ = distance, inches, of centroidal axis from the extreme fiber on the inner or concave side. Seely's Advanced Mechanics of Materials contains an analysis of curved beams and also Table 6 which gives values of $K$ for a number of shapes and ratios of $R/c$. For slightly different shapes or proportions $K$ may be found by interpolation.

### Table 3.—Allowable Unit Stresses for Timber in Bending *

Recommended by the Forest Products Laboratory, Forest Service, U. S. Dept. of Agriculture.† All values are in pounds per square inch.

| Species | Continuously Dry | | Occasionally Wet but Quickly Dried | | | | More or Less Continuously Damp or Wet | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | All Thicknesses | | 4 in. and Thinner | | 5 in. and Thicker | | 4 in. and Thinner | | 5 in. and Thicker | |
| | Select | Common | Select | Common | Select | Common | Select | Common | Select | Common |
| Ash, black | 1000 | 800 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| " commercial white | 1400 | 1120 | 1070 | 910 | 1200 | 960 | 890 | 760 | 1000 | 800 |
| Aspen and large tooth aspen | 800 | 640 | 580 | 490 | 650 | 520 | 440 | 370 | 500 | 400 |
| Basswood | 800 | 640 | 580 | 490 | 650 | 520 | 440 | 370 | 500 | 400 |
| Beech | 1500 | 1200 | 1150 | 980 | 1300 | 1040 | 890 | 760 | 1000 | 800 |
| Birch, paper | 900 | 720 | 670 | 570 | 750 | 600 | 530 | 450 | 600 | 480 |
| " yellow and sweet | 1500 | 1200 | 1150 | 980 | 1300 | 1040 | 890 | 760 | 1000 | 800 |
| Cedar, Alaska | 1100 | 880 | 890 | 760 | 1000 | 800 | 800 | 680 | 900 | 720 |
| " western red | 900 | 720 | 710 | 600 | 800 | 640 | 670 | 570 | 750 | 600 |
| " northern and southern white | 750 | 600 | 580 | 490 | 650 | 520 | 530 | 450 | 600 | 480 |
| " Port Orford | 1100 | 880 | 890 | 760 | 1000 | 800 | 800 | 680 | 900 | 720 |
| Chestnut | 950 | 760 | 760 | 650 | 850 | 680 | 620 | 530 | 700 | 560 |
| Cottonwood, eastern and black | 800 | 640 | 580 | 490 | 650 | 520 | 530 | 450 | 600 | 480 |
| Cypress, southern | 1300 | 1040 | 980 | 830 | 1100 | 880 | 800 | 680 | 900 | 720 |
| Douglas fir (western Wash. and Ore.) ‡ | 1600 | 1200 | 1233 | 983 | 1387 | 1040 | 948 | 756 | 1067 | 800 |
| Douglas fir (dense) ‡ | 1750 | 1400 | 1349 | 1147 | 1517 | 1213 | 1037 | 882 | 1167 | 933 |
| " (Rocky Mt.) | 1100 | 880 | 800 | 680 | 900 | 720 | 620 | 530 | 700 | 560 |
| Elm, rock | 1500 | 1200 | 1150 | 980 | 1300 | 1040 | 890 | 760 | 1000 | 800 |
| " slippery and American | 1100 | 880 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| Fir, balsam | 900 | 720 | 670 | 570 | 750 | 600 | 530 | 450 | 600 | 480 |
| " commercial white | 1100 | 880 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| Gum, red, black, and tupelo | 1100 | 880 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| Hemlock, eastern | 1100 | 880 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| " western | 1300 | 1040 | 980 | 830 | 1100 | 880 | 800 | 680 | 900 | 720 |
| Hickory (true and pecan) | 1900 | 1520 | 1330 | 1130 | 1500 | 1200 | 1070 | 910 | 1200 | 960 |
| Larch, western | 1200 | 960 | 980 | 830 | 1100 | 880 | 800 | 680 | 900 | 720 |
| Maple, sugar and black | 1500 | 1200 | 1150 | 980 | 1300 | 1040 | 890 | 760 | 1000 | 800 |
| " red and silver | 1000 | 800 | 800 | 680 | 900 | 720 | 620 | 530 | 700 | 560 |
| Oak, commercial red and white | 1400 | 1120 | 1070 | 910 | 1200 | 960 | 890 | 760 | 1000 | 800 |
| Pine, southern yellow ‡ | ...... | 1200 | ...... | 983 | ...... | 1040 | ...... | 756 | ...... | 800 |
| " southern yellow (dense) ‡ | 1750 | 1400 | 1349 | 1147 | 1517 | 1213 | 1037 | 882 | 1167 | 933 |
| " northern and western white, western yellow, sugar | 900 | 720 | 710 | 600 | 800 | 640 | 670 | 570 | 750 | 600 |
| " Norway | 1100 | 880 | 890 | 760 | 1000 | 800 | 710 | 600 | 800 | 640 |
| Poplar, yellow | 1000 | 800 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| Redwood | 1200 | 960 | 890 | 760 | 1000 | 800 | 710 | 600 | 800 | 640 |
| Spruce, red, white and Sitka | 1100 | 880 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| " Engelmann | 750 | 600 | 580 | 490 | 650 | 520 | 440 | 370 | 500 | 400 |
| Sycamore | 1100 | 880 | 800 | 680 | 900 | 720 | 710 | 600 | 800 | 640 |
| Tamarack (eastern) | 1200 | 960 | 980 | 830 | 1100 | 880 | 800 | 680 | 900 | 720 |

* Stress in tension. The working stresses recommended for fiber stress in bending may be safely used for tension parallel to grain.

† American lumber standards. Basic provisions for American lumber standards grades are published by the U. S. Dept. of Commerce, Simplified Practice Recommendation No. 16, Lumber, revised July 1, 1926; specifications for grades conforming to American lumber standards are published in the 1933 Standards A.S.T.M., and in *Amer. Ry. Eng. Assoc. Bull.*, vol. xxx, No. 314, Feb. 1929.

‡ Exact figures given. In order to preserve the exact numerical relations among working stresses for grades involving rate of growth and density requirements, the values for Douglas fir (western Wash. and Ore.) and for southern yellow pine have not been rounded off, as have the values for the other species.

### Table 4.—Allowable Unit Stresses for Timber in Compression and Shear

Recommended by the Forest Products Laboratory, U. S. Dept. of Agriculture.*
All values are in pounds per square inch.

| Species | Compression ⊥ to Grain, Select and Common Grades | | | Horizontal Shear ‖ | | Compression ‖ to Grain (Short Columns with Ratio of Length to Least Dimension of 10 or Less) | | | | | | Average Modulus of Elasticity ‡ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Continuously Dry | Occasionally Wet but Quickly Dried | More or Less Continuously Damp or Wet | Not Varied with Conditions of Exposure | | Continuously Dry | | Occasionally Wet, but Quickly Dried | | More or Less Continuously Damp or Wet | | Not Varied with Conditions of Exposure or Grade |
| | | | | Select | Common | Select | Common | Select | Common | Select | Common | |
| Ash, black | 300 | 200 | 150 | 90 | 72 | 650 | 520 | 550 | 440 | 500 | 400 | 1,100,000 |
| " commercial white | 500 | 375 | 300 | 125 | 100 | 1100 | 880 | 1000 | 800 | 900 | 720 | 1,500,000 |
| Aspen and largetooth aspen | 150 | 125 | 100 | 80 | 64 | 700 | 560 | 550 | 440 | 450 | 360 | 900,000 |
| Basswood | 150 | 125 | 100 | 80 | 64 | 700 | 560 | 550 | 440 | 450 | 360 | 900,000 |
| Beech | 500 | 375 | 300 | 125 | 100 | 1200 | 960 | 1100 | 880 | 900 | 720 | 1,600,000 |
| Birch, paper | 200 | 150 | 100 | 80 | 64 | 650 | 520 | 550 | 440 | 450 | 360 | 1,000,000 |
| " yellow and sweet | 500 | 375 | 300 | 125 | 100 | 1200 | 960 | 1100 | 880 | 900 | 720 | 1,600,000 |
| Cedar, Alaska | 250 | 200 | 150 | 90 | 72 | 800 | 640 | 750 | 600 | 650 | 520 | 1,200,000 |
| " western red | 200 | 150 | 125 | 80 | 64 | 700 | 560 | 700 | 560 | 650 | 520 | 1,000,000 |
| " northern and southern white | 175 | 140 | 100 | 70 | 56 | 550 | 440 | 500 | 400 | 450 | 360 | 800,000 |
| " Port Orford | 250 | 200 | 150 | 90 | 72 | 900 | 720 | 825 | 660 | 750 | 600 | 1,200,000 |
| Chestnut | 300 | 200 | 150 | 90 | 72 | 800 | 640 | 700 | 560 | 600 | 480 | 1,000,000 |
| Cottonwood, eastern and black | 150 | 125 | 100 | 80 | 64 | 700 | 560 | 550 | 440 | 450 | 360 | 900,000 |
| Cypress, southern | 350 | 250 | 225 | 100 | 80 | 1100 | 880 | 1000 | 800 | 800 | 640 | 1,200,000 |
| Douglas fir (western Wash. and Ore.) † | §<br>347 | §<br>240 | §<br>213 | 90 | 72 | 1173 | 880 | 1067 | 800 | 907 | 680 | 1,600,000 |
| Douglas fir (dense) † | 379 | 262 | 233 | 105 | 84 | 1283 | 1027 | 1167 | 933 | 992 | 793 | 1,600,000 |
| " " (Rocky Mt.) | 275 | 225 | 200 | 85 | 68 | 800 | 640 | 800 | 640 | 700 | 560 | 1,200,000 |
| Elm, rock | 500 | 375 | 300 | 125 | 100 | 1200 | 960 | 1100 | 880 | 900 | 720 | 1,300,000 |
| " slippery and Amer. | 250 | 175 | 125 | 100 | 80 | 800 | 640 | 750 | 600 | 650 | 520 | 1,200,000 |
| Fir, balsam | 150 | 125 | 100 | 70 | 56 | 700 | 560 | 600 | 480 | 500 | 400 | 1,000,000 |
| " commercial white | 300 | 225 | 200 | 70 | 56 | 700 | 560 | 700 | 560 | 600 | 480 | 1,100,000 |
| Gum, red, black, and tupelo | 300 | 200 | 150 | 100 | 80 | 800 | 640 | 750 | 600 | 650 | 520 | 1,200,000 |
| Hemlock, eastern | 300 | 225 | 200 | 70 | 56 | 700 | 560 | 700 | 560 | 600 | 480 | 1,100,000 |
| " western | 300 | 225 | 200 | 75 | 60 | 900 | 720 | 900 | 720 | 800 | 640 | 1,400,000 |
| Hickory (true and pecan) | 600 | 400 | 350 | 140 | 112 | 1500 | 1200 | 1200 | 960 | 1000 | 800 | 1,800,000 |
| Larch, western | 325 | 225 | 200 | 100 | 80 | 1100 | 880 | 1000 | 800 | 800 | 640 | 1,300,000 |
| Maple, sugar and black | 500 | 375 | 300 | 125 | 100 | 1200 | 960 | 1100 | 880 | 900 | 720 | 1,600,000 |
| " red and silver | 350 | 250 | 200 | 100 | 80 | 800 | 640 | 700 | 560 | 600 | 480 | 1,100,000 |
| Oak, commercial red and white | 500 | 375 | 300 | 125 | 100 | 1000 | 800 | 900 | 720 | 800 | 640 | 1,500,000 |
| Pine, southern yellow † | § | § | § | .... | 88 | ..... | 880 | ..... | 800 | .... | 680 | 1,600,000 |
| " southern yellow (dense) † | 379 | 262 | 233 | 128 | 103 | 1283 | 1027 | 1167 | 933 | 992 | 793 | 1,600,000 |
| " northern and western white, western yellow, sugar | 250 | 150 | 125 | 85 | 68 | 750 | 600 | 750 | 600 | 650 | 520 | 1,000,000 |
| " Norway | 300 | 175 | 150 | 85 | 68 | 800 | 640 | 800 | 640 | 700 | 560 | 1,200,000 |
| Poplar, yellow | 250 | 150 | 125 | 80 | 64 | 800 | 640 | 700 | 560 | 600 | 480 | 1,100,000 |
| Redwood | 250 | 150 | 125 | 70 | 56 | 1000 | 800 | 900 | 720 | 750 | 600 | 1,200,000 |
| Spruce, red, white and Sitka | 250 | 150 | 125 | 85 | 68 | 800 | 640 | 750 | 600 | 650 | 520 | 1,200,000 |
| " Engelmann | 175 | 140 | 100 | 70 | 56 | 600 | 480 | 550 | 440 | 450 | 360 | 800,000 |
| Sycamore | 300 | 200 | 150 | 80 | 64 | 800 | 640 | 750 | 600 | 650 | 520 | 1,200,000 |
| Tamarack (eastern) | 300 | 225 | 200 | 95 | 76 | 1000 | 800 | 900 | 720 | 800 | 640 | 1,300,000 |

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* See footnote (†), Table 3.

† See footnote (‡), Table 3.

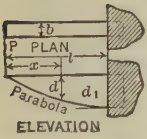
‡ The values for modulus of elasticity are average for species, and not safe working stresses. They may be used as given for computing average deflection of beams. To prevent sag in beams, values one-half those given should be used. For safe loads for long columns, values one-third those given should be used.

§ Values given are for the Select grade. Working stresses in compression, perpendicular to the grain, for common grades of Douglas fir (western Wash. and Ore.) and southern yellow pine are 325, 225, and 200, respectively, for continuously dry, occasionally wet but quickly dried, and more or less continuously damp or wet conditions.

‖ Joint details. The shearing stresses for joint details may be taken for any grade as 50% greater than the horizontal shear values for the Select grade.

**Table 5.—Rectangular Beams of Uniform Strength** *

$s$ = maximum fiber unit stress, pounds per square inch; $E$ = modulus of elasticity; $w$ = uniform load, pounds per inch; $d$ = depth of beam, inches; $b$ = width of beam, inches; $y$ = maximum deflection, inches. All other dimensions in inches.



**I. Cantilever Beam Loaded at Free End**

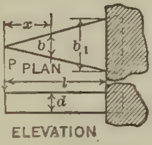
Width is constant. Depth varies.

$d = d_1\sqrt{x/l}$

$s = 6\,Pl/bd_1^2$

$y = 8\,Pl^3/Ebd_1^3$

Elevation is formed by a straight line and a parabola with its vertex at the loaded end.



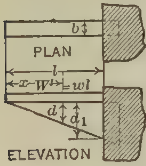
**II. Cantilever Beam Loaded at Free End**

Depth constant. Width varies.

$b = b_1x/l$

$s = 6Pl/b_1d^2$

$y = 6Pl^3/Eb_1d^3$



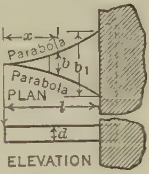
**III. Cantilever Beam Uniformly Loaded**

Width is constant. Depth varies.

$d = (x/l)d_1$

$s = 3\,wl^2/bd_1^2$

$y = 6wl^4/bEd_1^3$



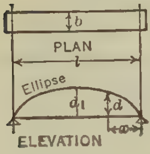
**IV. Cantilever Beam Uniformly Loaded**

Depth is constant. Width varies.

$b = b_1x^2/l^2$

$s = 3wl^2/b_1d^2$

$y = 3wl^4/b_1Ed^3$



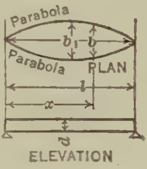
**V. Simple Beam Uniformly Loaded**

Width is constant. Depth varies.

$$d = \sqrt{\frac{4d_1^2(lx - x^2)}{l^2}}$$

$$s = \frac{3wl^2}{4bd_1^2}$$

Elevation is formed by a straight line and an ellipse.



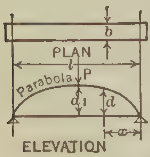
**VI. Simple Beam Uniformly Loaded**

Depth is constant. Width varies.

$$b = \frac{4b_1}{l^2}(lx - x^2)$$

$$s = \frac{3}{4}\frac{wl^2}{b_1d^2}$$

Plan is two parabolas, with vertices at center of span.



**VII. Simple Beam Loaded at Center of Span**

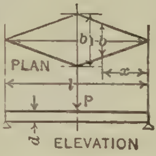
Width is constant. Depth varies.

$d = d_1\sqrt{2x/l}$

$$s = \frac{3}{2}\frac{Pl}{bd_1^2}$$

$$y = \frac{1}{2}\frac{Pl^3}{Ebd_1^3}$$

Elevation is a parabola with vertices at points of support.



**VIII. Simple Beam Loaded at Center of Span**

Depth is constant. Width varies.

$b = 2b_1x/l$

$$s = \frac{3}{2}\frac{Pl}{b_1d^2}$$

$$y = \frac{3}{8}\frac{Pl^3}{Eb_1d^3}$$

Plan is two triangles with vertices at points of support.

---

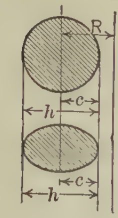
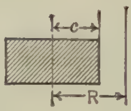
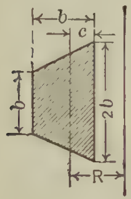
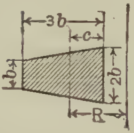
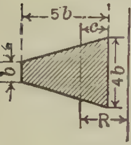
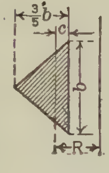
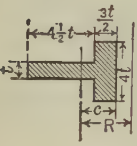
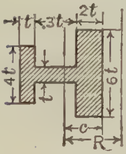
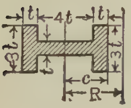
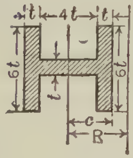
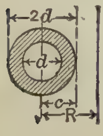
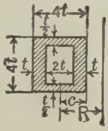
* The sections of the beams near the ends must be increased over the amounts shown to resist the vertical shear expressed by the formula $s = {}^3/_2V/A$.

## 5. CONTINUOUS BEAMS

As in simple beams, the expressions $M = sI/c$ and $s_s = V/A$ govern the design and investigation of beams resting on more than two supports. In the case of continuous beams, however, the reactions cannot be obtained in the manner described for simple beams. Instead, the bending moments at the various sections must be determined, and from these values the vertical shears at the sections and the reactions at the supports may be derived.

Consider the second span of length $l_2$, in., of the continuous beam, Fig. 7. Vertical shear $V_x$ at any section distant $x$, in. from the left support of the span is equal to the algebraic sum of all the vertical forces on one side of the section. Thus, if $V_2$ = vertical

**Table 6.—Values of Constant *K* for Curved Beams**

| Section | $\frac{R}{c}$ | Values of *K* | | $\frac{Y_0}{R}$ * | Section | $\frac{R}{c}$ | Values of *K* | | $\frac{Y_0}{R}$ * |
|---|---|---|---|---|---|---|---|---|---|
| | | Inside Fiber | Outside Fiber | | | | Inside Fiber | Outside Fiber | |
|  | 1.2 | 3.41 | 0.54 | 0.224 |  | 1.2 | 2.89 | 0.57 | 0.305 |
| | 1.4 | 2.40 | 0.60 | 0.151 | | 1.4 | 2.13 | 0.63 | 0.204 |
| | 1.6 | 1.96 | 0.65 | 0.108 | | 1.6 | 1.79 | 0.67 | 0.149 |
| | 1.8 | 1.75 | 0.68 | 0.084 | | 1.8 | 1.63 | 0.70 | 0.112 |
| | 2.0 | 1.62 | 0.71 | 0.069 | | 2.0 | 1.52 | 0.73 | 0.090 |
| | 3.0 | 1.33 | 0.79 | 0.030 | | 3.0 | 1.30 | 0.81 | 0.041 |
| | 4.0 | 1.23 | 0.84 | 0.016 | | 4.0 | 1.20 | 0.85 | 0.021 |
| | 6.0 | 1.14 | 0.89 | 0.0070 | | 6.0 | 1.12 | 0.90 | 0.0093 |
| | 8.0 | 1.10 | 0.91 | 0.0039 | | 8.0 | 1.09 | 0.92 | 0.0052 |
| | 10.0 | 1.08 | 0.93 | 0.0025 | | 10.0 | 1.07 | 0.94 | 0.0033 |
|  | 1.2 | 3.01 | 0.54 | 0.336 |  | 1.2 | 3.09 | 0.56 | 0.336 |
| | 1.4 | 2.18 | 0.60 | 0.229 | | 1.4 | 2.25 | 0.62 | 0.229 |
| | 1.6 | 1.87 | 0.65 | 0.168 | | 1.6 | 1.91 | 0.66 | 0.168 |
| | 1.8 | 1.69 | 0.68 | 0.128 | | 1.8 | 1.73 | 0.70 | 0.128 |
| | 2.0 | 1.58 | 0.71 | 0.102 | | 2.0 | 1.61 | 0.73 | 0.102 |
| | 3.0 | 1.33 | 0.80 | 0.046 | | 3.0 | 1.37 | 0.81 | 0.046 |
| | 4.0 | 1.23 | 0.84 | 0.024 | | 4.0 | 1.26 | 0.86 | 0.024 |
| | 6.0 | 1.13 | 0.88 | 0.011 | | 6.0 | 1.17 | 0.91 | 0.011 |
| | 8.0 | 1.10 | 0.91 | 0.0060 | | 8.0 | 1.13 | 0.94 | 0.0060 |
| | 10.0 | 1.08 | 0.93 | 0.0039 | | 10.0 | 1.11 | 0.95 | 0.0039 |
|  | 1.2 | 3.14 | 0.52 | 0.352 |  | 1.2 | 3.26 | 0.44 | 0.361 |
| | 1.4 | 2.29 | 0.54 | 0.243 | | 1.4 | 2.39 | 0.50 | 0.251 |
| | 1.6 | 1.93 | 0.62 | 0.179 | | 1.6 | 1.99 | 0.54 | 0.186 |
| | 1.8 | 1.74 | 0.65 | 0.138 | | 1.8 | 1.78 | 0.57 | 0.144 |
| | 2.0 | 1.61 | 0.68 | 0.110 | | 2.0 | 1.66 | 0.60 | 0.116 |
| | 3.0 | 1.34 | 0.76 | 0.050 | | 3.0 | 1.37 | 0.70 | 0.052 |
| | 4.0 | 1.24 | 0.82 | 0.028 | | 4.0 | 1.27 | 0.75 | 0.029 |
| | 6.0 | 1.15 | 0.87 | 0.012 | | 6.0 | 1.16 | 0.82 | 0.013 |
| | 8.0 | 1.12 | 0.91 | 0.0060 | | 8.0 | 1.12 | 0.86 | 0.0060 |
| | 10.0 | 1.11 | 0.93 | 0.0039 | | 10.0 | 1.09 | 0.88 | 0.0039 |
|  | 1.2 | 3.63 | 0.58 | 0.418 |  | 1.2 | 3.55 | 0.67 | 0.409 |
| | 1.4 | 2.54 | 0.63 | 0.299 | | 1.4 | 2.48 | 0.72 | 0.292 |
| | 1.6 | 2.14 | 0.67 | 0.229 | | 1.6 | 2.07 | 0.76 | 0.224 |
| | 1.8 | 1.89 | 0.70 | 0.183 | | 1.8 | 1.83 | 0.78 | 0.178 |
| | 2.0 | 1.73 | 0.72 | 0.149 | | 2.0 | 1.69 | 0.80 | 0.144 |
| | 3.0 | 1.41 | 0.79 | 0.069 | | 3.0 | 1.38 | 0.86 | 0.067 |
| | 4.0 | 1.29 | 0.83 | 0.040 | | 4.0 | 1.26 | 0.89 | 0.038 |
| | 6.0 | 1.18 | 0.88 | 0.018 | | 6.0 | 1.15 | 0.92 | 0.018 |
| | 8.0 | 1.13 | 0.91 | 0.010 | | 8.0 | 1.10 | 0.94 | 0.010 |
| | 10.0 | 1.10 | 0.92 | 0.0065 | | 10.0 | 1.08 | 0.95 | 0.0065 |
|  | 1.2 | 2.52 | 0.67 | 0.408 |  | 1.2 | 2.37 | 0.73 | 0.453 |
| | 1.4 | 1.90 | 0.71 | 0.285 | | 1.4 | 1.79 | 0.77 | 0.319 |
| | 1.6 | 1.63 | 0.75 | 0.208 | | 1.6 | 1.56 | 0.79 | 0.236 |
| | 1.8 | 1.50 | 0.77 | 0.160 | | 1.8 | 1.44 | 0.81 | 0.183 |
| | 2.0 | 1.41 | 0.79 | 0.127 | | 2.0 | 1.36 | 0.83 | 0.147 |
| | 3.0 | 1.23 | 0.86 | 0.058 | | 3.0 | 1.19 | 0.88 | 0.067 |
| | 4.0 | 1.16 | 0.89 | 0.030 | | 4.0 | 1.13 | 0.91 | 0.036 |
| | 6.0 | 1.10 | 0.92 | 0.013 | | 6.0 | 1.08 | 0.94 | 0.016 |
| | 8.0 | 1.07 | 0.94 | 0.0076 | | 8.0 | 1.06 | 0.95 | 0.0089 |
| | 10.0 | 1.05 | 0.95 | 0.0048 | | 10.0 | 1.05 | 0.96 | 0.0057 |
|  | 1.2 | 3.28 | 0.58 | 0.269 |  | 1.2 | 2.63 | 0.68 | 0.399 |
| | 1.4 | 2.31 | 0.64 | 0.182 | | 1.4 | 1.97 | 0.73 | 0.280 |
| | 1.6 | 1.89 | 0.68 | 0.134 | | 1.6 | 1.66 | 0.76 | 0.205 |
| | 1.8 | 1.70 | 0.71 | 0.104 | | 1.8 | 1.51 | 0.78 | 0.159 |
| | 2.0 | 1.57 | 0.73 | 0.083 | | 2.0 | 1.43 | 0.80 | 0.127 |
| | 3.0 | 1.31 | 0.81 | 0.038 | | 3.0 | 1.23 | 0.86 | 0.058 |
| | 4.0 | 1.21 | 0.85 | 0.020 | | 4.0 | 1.15 | 0.89 | 0.031 |
| | 6.0 | 1.13 | 0.90 | 0.0087 | | 6.0 | 1.09 | 0.92 | 0.014 |
| | 8.0 | 1.10 | 0.92 | 0.0049 | | 8.0 | 1.07 | 0.94 | 0.0076 |
| | 10.0 | 1.07 | 0.93 | 0.0031 | | 10.0 | 1.06 | 0.95 | 0.0048 |

* $Y_0$ is distance from centroidal axis to neutral axis, where beam is subjected to pure bending.

shear at a section to the right of, but infinitely close to, the left support, $w_2x$ = uniform load and $\Sigma P_2$ = sum of the concentrated loads along the distance $x$, applied at a distance $kl_2$ from the left support, $k$ being a fraction less than unity, then

$$V_x = V_2 - w_2 x - \Sigma P_2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \qquad [13]$$

At any section, distant $x$ from the left support, the bending moment is equal to the algebraic sum of the moments of all forces on one side of the section. If $M_2$ is the moment, lb.-in., at the support to the left,

$$M_x = M_2 + V_2 x - (w_2 x^2/2) - \Sigma P_2(x - kl_2) \quad . \; . \; . \qquad [14]$$

Assume that $x = l_2$. Then $M_x$ becomes the moment $M_3$ at the next support to the right, and the expression may be written

$$V_2 l_2 = M_3 - M_2 + (w_2 l_2^2/2) + \Sigma P_2(l_2 - kl_2) \; . \; . \; . \; . \qquad [15]$$

From formulas [13], [14], and [15] it is evident that the bending moment $M_x$ and the shear $V_x$ at any section between two consecutive supports may be determined if the bending moments $M_2$ and $M_3$ at those supports are known.

To determine bending moments at the supports an expression known as the *theorem of three moments is used.* This gives the relation between the moments at any three consecutive supports of a beam. For beams with the supports on the same level, and uniformly loaded over each span, the formula is

$$M_1 l_1 + 2\,M_2(l_1 + l_2) + M_3 l_2 = -\,{}^1\!/_4\, w_1 l_1^3 - {}^1\!/_4\, w_2 l_2^3 \quad . \; . \; . \qquad [16]$$

where $M_1$, $M_2$ and $M_3$ = moments of three consecutive supports; $l_1$ = length between first and second support; $l_2$ = length between second and third support; $w_1$ = uniform load per lineal unit over the first span; $w_2$ = uniform load per lineal unit over the second span. When both spans are of equal length and when the load on each span is the same, $l_1 = l_2$, $w_1 = w_2$, and formula [16] reduces to

$$M_1 + 4M_2 + M_3 = -\,{}^1\!/_2\, w\, l^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \qquad [17]$$

which applies to most cases in practice.

Formulas [16] and [17] are used as follows: For any continuous beam of $n$ spans there are $(n + 1)$ supports. Assuming the ends of the beam to be simply supported without any overhang, the moments at the end supports are zero, and there are, therefore, to be determined $(n - 1)$ moments at the other supports. This may be done by writing $(n - 1)$ equations of the form of [16] and [17] for each support. These equations will contain $(n - 1)$ unknown moments and their solution will give values of $M_1$, $M_2$, $M_3$, etc., expressed as coefficients of $wl^2$. The shear $V_1$ at any support may be determined by substituting values of $M_1$ and $M_2$ in formula [15], and the bending moment at any point in any span may be obtained by formula [14]. The shear at any point in any span may be determined from formula [13].

Fig. 8 gives values and diagrams for the reactions, shears, and moments at all sections of continuous beams uniformly loaded up to five spans. Note that the reaction at any support is equal to the sum of the shears to the right and to the left of that support.

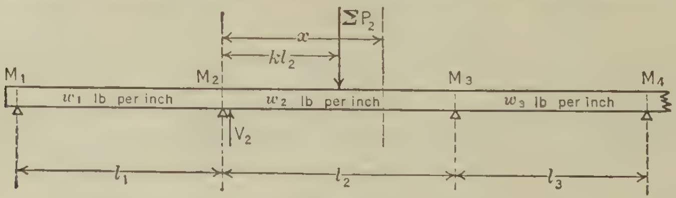


FIG. 7

## 6. AXIALLY END-LOADED BEAMS *

**DEFLECTION OF BEAM NEGLIGIBLE.**—When a beam is subjected to an axial end load as well as to a transverse bending load, the resultant unit stress developed at any point in the beam is the algebraic sum of the unit stresses produced by each of the loads acting independently of each other. If the beam of cross-sectional area $A$, sq. in., is considered so short that its deflection under load may be neglected, the unit stress produced at any cross-section of the beam by an axial load $P$, pounds, is $P/A$. The maximum unit stress (tensile or compressive), lb. per sq. in., due to the transverse or bending load is given by $s = M/(I/c)$ where $M$, lb.-in., is the maximum bending moment due to this load; $I$, in.$^4$, is the moment of inertia of the cross-section with respect to the neutral axis; $c$, in., is the distance from the neutral axis to the extreme fiber. If the axial load is compressive, the maximum unit stress is compressive, occurring at the top of the beam, and is

---

* Condensed from Seely, Resistance of Materials, 2nd Ed., Chapter VII.

$$s = (P/A) + (Mc/I) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [18]$$

The unit stress on the bottom fiber may be either tensile or compressive, according as $Mc/I$ is larger or smaller than $P/A$. If the longitudinal load $P$ were a tensile load, the maximum unit stress would occur on the bottom fiber and would be a tensile stress.

**DEFLECTION OF BEAM NOT NEGLIGIBLE.**—If the deflection of the beam is not negligible, the load $P$ cannot be considered an axial load with respect to any cross-sections except the end sections. The longitudinal load then has a moment arm equal to the deflection, and the stress due to this moment should be added algebraically to that caused by the other moments or loads. The stress due to deflection, $y$, inches (all other symbols as above), is $Pyc/I$. Thus, for a compressive longitudinal load, the maximum unit stress, lb. per sq. in., is at the top of the beam and is

$$s = (P/A) + (M + Py)(c/I) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [19]$$

Unit tensile stress, lb. per sq. in., at the bottom of the beam for the same load is

$$s = (-P/A) + (M + Py)(c/I) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [20]$$

Since the value of $y$ depends on the total bending moment $(M + Py)$, and the bending moment depends, in turn, on the value of $y$, equations [19] and [20] usually are solved by a method of approximation. The value of $y$ that would be caused by the cross-bending moment $M$, considered acting alone, is found first, and then is used in the expression, $M + Py$. This new value of the total bending moment is used to find a closer approximation to $y$. This operation may be repeated as many times as desired. When the deflection is small, as in comparatively short beams, the value of $y$ as found from the cross-bending moment above often is used without any further approximation.

Example.—Find the stress at the bottom and at the top of a timber beam 6 in. wide, 8 in. deep, and 12 ft. long, supported at each end, and carrying a distributed load of 400 lb. per ft., and also

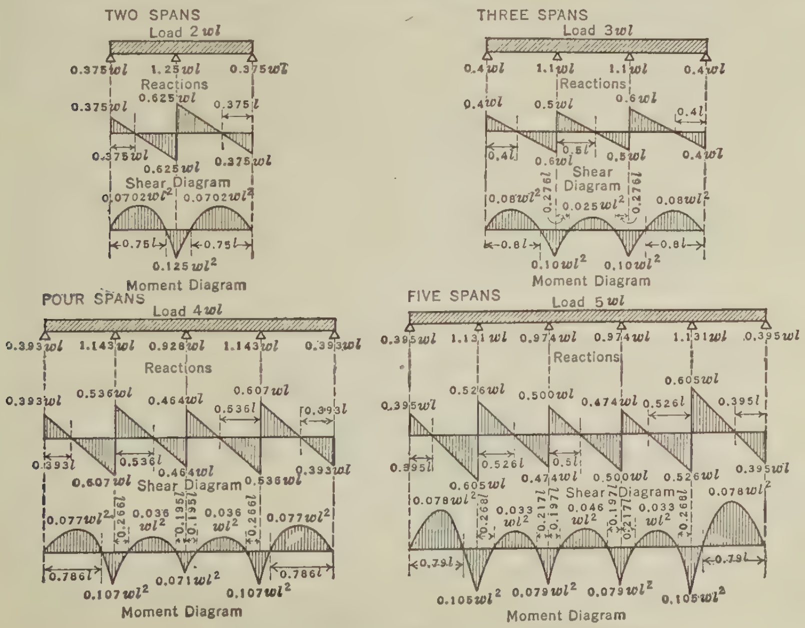


**Fig. 8. Shear and Moment Diagrams of Continuous Beams**

an axial compressive load of 10,000 lb. The modulus of elasticity of the timber may be taken as 1,600,000.

*Solution.*—Maximum unit stress will be at the center of the beam.

$$(P/A) = (10{,}000/48) = 208 \text{ lb. per sq. in.}$$

$$(Mc/I) = \{(400 \times 12 \times 144)/8\} \times \{6/(6 \times 8^2)\} = 1350 \text{ lb. per sq. in.}$$

See Table 2, p. 7–20.

$$y = \frac{5 \times 4800 \times 144^3 \times 12}{384 \times 6 \times 8^3 \times 1{,}600{,}000} = 0.456 \text{ in.}$$

$$Pyc/I = (10{,}000 \times 0.456 \times 6)/(6 \times 8^2) = 71 \text{ lb. per sq. in.}$$

Total unit compressive stress (top) = 208 + 1350 + 71 = 1629 lb. per sq. in.
Total unit tensile stress (bottom) = − 208 + 1350 + 71 = 1213 lb. per sq. in.
Here the stress due to deflection is small, as it usually will be in short deep beams.

# COLUMNS

## 1. DEFINITIONS

**A COLUMN OR STRUT** is a bar or structural member under axial compression, which has an unbraced length greater than about 8 or 10 times the least dimension of its cross-section. On account of its length, it is impossible to hold a column in a straight line under a load; a slight sidewise bending always occurs, causing flexural stresses in addition to the compressive stresses induced directly by the load. The lateral deflection will be in a direction perpendicular to that axis of the cross-section about which the moment of inertia is the least. Thus in Fig. 1*A* the column will bend in a direction perpendicular to *aa*; in Fig. 1*B* it will bend perpendicular to *aa* or *bb* and in Fig. 1*C* it is apt to bend in any direction.

**RADIUS OF GYRATION** of a section with respect to a given axis is equal to the square root of the quotient of the moment of inertia with respect to that axis, divided by the area of the section, that is

$$k = \sqrt{I/A}; \quad I/A = k^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

where $I$ is the moment of inertia and $A$ the sectional area. Unless otherwise mentioned, an axis through the center of gravity of the section is the axis considered. As in beams, the moment of inertia is an important factor in the ability of the column to resist bending, but for purposes of computation it is more convenient to use the radius of gyration.

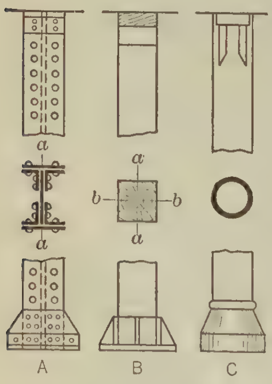


FIG. 1

**LENGTH OF A COLUMN** is the distance between points unsupported against lateral deflection.

**SLENDERNESS RATIO** is the length $l$ divided by the least radius of gyration $k$, both in inches. For steel, a *short column* is one in which $l/k < 20$ or 30, and its failure under load is due mainly to direct compression; in a *medium length column*, $l/k =$ about 30 to 175, and it will fail by a combination of direct compression and bending; in a *long column*, $l/k >$ about 175–200, and it will fail mainly by bending. For timber columns these ratios are about 0–30, 30–90, and above 90 respectively. The load which will cause a column to fail decreases as $l/k$ increases. The above ratios apply to round-end columns. If the ends are fixed (see below), the effective slenderness ratio is one-half of that for round-end columns, as the distance between the points of inflection is one-half the total length of the column. For flat ends it is intermediate between the two.

**CONDITION OF ENDS.**—The various conditions which may exist at the ends of columns usually are divided into four classes: 1. Columns with round ends; the bearing at either end has perfect freedom of motion, as there would be with a ball-and-socket joint at each end. 2. Columns with hinged ends; have perfect freedom of motion at the ends in one plane, as in compression members in bridge trusses where loads are transmitted through end pins. 3. Columns with flat ends; the bearing surface is normal to the axis of the column and of sufficient area to give at least partial fixity to the ends of the columns against lateral deflection. 4. Columns with fixed ends; the ends are rigidly secured, so that under any load the tangent to the elastic curve at the ends will be parallel to the axis in its original position.

Experiments prove that columns with fixed ends are stronger than columns with flat, hinged, or round ends, and that columns with round ends are weaker than any of the other types. Columns with hinged ends are equivalent to those with round ends in the plane in which they have free movement; columns with flat ends have a value intermediate between those with fixed ends and those with round ends. It often happens that columns have one end fixed and one end hinged, or some other combination. Their relative values may be taken as intermediate between those represented by the condition at either end. The extent to which strength is increased by fixing the ends depends on the length of column, fixed ends having a greater effect on long columns than on short ones.

## 2. COLUMN FORMULAS

There is no exact theoretical formula which gives the strength of a column of any length under an axial load. Formulas involving the use of empirical coefficients have been deduced, however, and they give results which are consistent with the results of tests of full-sized members.

**EULER'S FORMULA** assumes that the failure of a column is due solely to the stresses induced by sidewise bending. This assumption is not true for very short columns, which fail mainly by direct compression, nor is it true for columns of medium length. The failure in such cases is by a combination of direct compression and bending. For columns in which $l/k > 200$, Euler's formula is approximately correct and agrees closely with the results of tests.

Let $P$ = axial load, lb.; $l$ = length of column, in.; $I$ = least moment of inertia, in.$^4$; $k$ = least radius of gyration, in.; $E$ = modulus of elasticity; $y$ = lateral deflection, in., at any point along the column, that is caused by load $P$. If a column has round ends, so that the bending is not restrained, the equation of its elastic curve is

$$EI(d^2y/dx^2) = -Py, \qquad [2]$$

when the origin of the coordinate axes is at the top of the column, the positive direction of $x$ being taken downwards and the positive direction of $y$ in the direction of the deflection. Integrating the above expression twice and determining the constants of integration, there results

$$P = \Omega\pi^2 EI/l^2 \qquad [3]$$

which is Euler's formula for long columns. The factor $\Omega$ is a constant depending on the condition of the ends. For round ends $\Omega = 1$; for fixed ends $\Omega = 4$; for one end round and the other fixed $\Omega = 2.05$. $P$ is the load at which, if a slight deflection is produced, the column will not return to its original position; if $P$ is decreased the column will approach its original position but if $P$ is increased the deflection will increase until the column fails by bending.

For columns with value of $l/k$ less than about 150, Euler's formula gives results distinctly higher than those observed in tests. Euler's formula is now little used except for long members and as a basis for the analysis of the stresses in some types of structural and machine parts. It always gives an *ultimate* and never an allowable load.

**SECANT FORMULA.**—The deflection of the column is used in the derivation of the Euler formula, but if the load were truly axial it would be impossible to compute the deflection. If the column is assumed to have an initial eccentricity of load of $e$ in. (see Seely, Resistance of Materials, 2nd Ed., p. 212, for suggested values of $e$), the equation for the deflection $y$ becomes

$$y_{\max.} = e\{\sec (l/2)(\sqrt{P/EI}) - 1\} \qquad [4]$$

The maximum unit compressive stress becomes

$$s = (P/A)\{1 + (ec/k^2) \sec (l/2)\sqrt{P/EI}\} \qquad [5]$$

where $l$ = length of column, in.; $P$ = total load, lb.; $A$ = area, sq. in.; $I$ = moment of inertia, in.$^4$; $k$ = radius of gyration, in.; $c$ = distance from neutral axis to the most compressed fiber, in.; $E$ = modulus of elasticity; $I$ and $k$ both are taken with respect to the axis about which bending takes place. Because the formula contains the secant of the angle $(l/2)\sqrt{P/EI}$ it is sometimes called the *secant formula*. It has been suggested by the committee on Steel-Column Research (*Trans.*, A.S.C.E., 1933) that the best rational column formula can be constructed on the secant type, though of course it must contain experimental constants.

The secant formula can be used also for columns that are eccentrically loaded, if $e$ is taken as the actual eccentricity plus the assumed initial eccentricity.

**RANKINE'S FORMULA** is a modification of an earlier formula by Gordon, and sometimes is called Gordon's Formula. It applies to columns of slenderness ratios of 20

**Table 1.—Value of Coefficient $\phi$ in Rankine's Formula**

| Material | Both Ends Fixed | Fixed and Round | Both Ends Round |
|---|---|---|---|
| Timber | 1/3000 | 1.95/3000 | 4/3000 |
| Cast iron | 1/5000 | 1.95/5000 | 4/5000 |
| Wrought iron | 1/36,000 | 1.95/36,000 | 4/36,000 |
| Steel | 1/25,000 | 1.95/25,000 | 4/25,000 |

to 200, which are usually required in practice. Rankine's formula assumes that the maximum unit stress, $s$, lb. per sq. in., occurs in the fiber most remote from the axis on the concave side of the deflected column, and is equal to the sum of the direct unit compressive stress and the unit compressive stress due to bending alone; that is, $s = (P/A) + s_1$, where $P$ = load, lb.; $A$ = sectional area, sq. in.; $s_1$ = unit compressive stress due to bending, lb. per sq. in. The value of $s_1$ in this equation may be expressed in terms of $P$ by using the flexure formula, since the action is analogous to that of beams. Let $l$ = length of the column, in.; $I$ = least moment of inertia, in.$^4$; $k$ = corresponding radius of gyration, in.; $c$ = distance from axis to the remotest fiber on the concave side, in.; $y$ = lateral deflection of column, in. Substituting in the flexure formula, $s_1 = Mc/I$, and by replacing $M$ by its value $Py$ and $I$ by its value $Ak^2$, $s_1 = Pyc/(Ak^2)$. Hence maximum unit compressive stress on the remotest fiber on the concave side is equal to $P/A + Pyc/(Ak^2)$. By analogy with the theory of beams the deflection $y$ varies as $l^2/c$, so that by introducing an experimental coefficient $\phi$, the expression for maximum unit stress may be written

$$s = (P/A) + \phi(Pl^2/Ak^2) = (P/A)[1 + \phi(l/k)^2] \quad . \quad . \quad . \quad . \quad [6]$$

which is Rankine's formula for the investigation of columns. Variations in the value of $\phi$ have been determined by many experiments on columns of different materials and different conditions of ends. Values in Table 1, given by Merriman, have been used extensively in the past. Present practice (1937) tends toward the use of the same constant for all end conditions, as tests show that for columns of the proportions in common use, variations in material are more important than variations in end conditions.

The value to be taken for $s$ in Rankine's formula is the ultimate compressive stress of the material for rupture, and the allowable compressive unit stress for design. Rankine's formula may be used in the investigation of an existing column by comparing the computed value of $s$ with the proportional limit and with the ultimate strength of the material, thus determining the factor of safety. Similarly, the safe load $P$ for an existing column may be computed by assuming an allowable unit stress $s$.

**STRAIGHT-LINE FORMULA.**—Plotted results of actual tests on columns show that the relation between ultimate load and $l/k$ is fairly well represented by a straight line, for columns which fail by flexure of the whole column and not by local collapse. For a value of $l/k = 0$, average unit stress, $P/A$, on the section, is equal to the ultimate stress in compression for brittle materials and to the yield point in compression for ductile materials. From the point thus determined the straight line representing the average results of tests is drawn tangent to the curve of Euler's formula for the material. The equation of the straight line is

$$P/A = s - Cl/k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

where $s$ = ultimate unit stress in compression for a brittle material, or unit stress at the yield point in compression for a ductile material; $C$ = an experimental constant. Table 2 gives average values of $s$ and $C$ for structural steel, cast iron, and wood. The loads given by the straight-line formula using the constants in this table are *ultimate loads.* For design the straight-line formula usually is put in a form which gives allowable loads directly, constants $s$ and $C$ being divided by a factor of safety. The straight-line formula is not suitable for investigating a column, *i.e.*, determining values of $s$ due to given loads, as $(Cl/k)$ is not a function of $(P/A)$. It may be used to find the safe load for a given column under a given unit stress, or to design a column for a given load and unit stress.

**Table 2.—Ultimate Strength Constants for the Straight-line Column Formula**

| Material | $s_1$ lb. per sq. in. | Character of Ends | | | | | |
|---|---|---|---|---|---|---|---|
| | | Round | Fixed | 1 round 1 fixed | Round | Fixed | 1 round 1 fixed |
| | | $C$ | | | Limit of $(l/k)$ | | |
| Structural steel | 35,000 | 150 | 75 | 100 | 160 | 320 | 240 |
| Cast iron | 34,000* | 175 | 88 | 116 | 90 | 160 | 115 |
| Wood | 5,000* | 40 | 20 | 30 | 75 | 150 | 112 |

* This is less than the ultimate in compression for small specimens of cast iron or wood, but from tests of full-size columns it seems to be the value to be used for full-size castings or timbers which may contain defects.

## 3. WOODEN COLUMNS

**WOODEN COLUMN FORMULAS.**—One of the principal formulas is that formerly used by the A.R.E.A., $P/A = s_1(1 - l/60d)$, where $P/A$ = allowable unit load, lb. per sq. in.; $s_1$ = allowable unit stress in direct compression on short blocks, lb. per sq. in.; $l$ = length, in.; $d$ = least dimension, in. This formula is being replaced rapidly by formulas recommended by the A.S.T.M. and A.R.E.A. Committees of these societies, working with the U. S. Forest Products Laboratory, classified timber columns into three groups, as follows (A.S.T.M. Standards, 1933, D245–33):

**1. Short Columns.**—The ratio of unsupported length to least dimension does not exceed 10. For these columns, the allowable unit stress should not be greater than the values given in Table 4, under compression parallel to the grain.

**2. Intermediate-length Columns.**—Where the ratio of unsupported length to least dimension is greater than 10, formula [8], of the fourth power parabolic type, shall be used to determine allowable unit stress, until this allowable unit stress is equal to $^2/_3$ of the allowable unit stress for short columns.

$$(P/A) = s_1\{1 - {}^1/_3\,(l/Kd)^4\} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

where $P$ = total load, lb.; $A$ = area, sq. in.; $s_1$ = allowable unit compressive stress parallel to grain, lb. per sq. in.; see Table 4. $l$ = unsupported length, in.; $d$ = least dimension, in.; $K = l/d$ at the point of tangency of the parabolic and Euler curves, at which $P/A = (2/3)\,s_1$. The value of $K$ for any species and grade is $(\pi/2\sqrt{E/6\,s_1})$, where $E$ = modulus of elasticity.

**3. Long Columns.**—Where $P/A$ as computed by formula [8] is less than $^2/_3\,s_1$, formula [9] of the Euler type, which includes a factor of safety of 3, shall be used:

$$P/A = {}^1/_{36}\{\pi^2\,E/(l/d)^2\} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

Timber columns should be limited to a ratio of $l/d$ equal to 50. No higher loads are allowed for square-ended columns. The strength of round columns may be considered the same as that of square columns of the same cross-sectional area.

**USE OF TIMBER COLUMN FORMULAS.**—The values of $E$ (modulus of elasticity) and $s_1$ (compression parallel to grain) to be used in the above formulas are given in Table 4. Table 3 gives the computed values of $K$ for some common types of timbers. These may be substituted directly in formula [8] for intermediate-length columns, or may be used in conjunction with Table 4, which gives the strength of columns of intermediate length, expressed as a percentage of strength ($s_1$) of short columns. In the tables, the term "continuously dry" refers to interior construction where there is no excessive dampness or humidity; "occasionally wet but quickly dry" refers to bridges, trestles, bleachers and grandstands; "usually wet" refers to timber in contact with the earth or exposed to waves or tide-water.

**Table 3.—Values of $K$ for Columns of Intermediate Length**

A.S.T.M. Standards, 1933, D245–33

| Species | Continuously dry | | Occasionally wet | | Usually wet | |
|---|---|---|---|---|---|---|
| | Select | Common | Select | Common | Select | Common |
| Cedar, western red | 24.2 | 27.1 | 24.2 | 27.1 | 25.1 | 28.1 |
| " Port Orford | 23.4 | 26.2 | 24.6 | 27.4 | 25.6 | 28.7 |
| Douglas fir, coast region | 23.7 | 27.3 | 24.9 | 28.6 | 27.0 | 31.1 |
| " " dense | 22.6 | 25.3 | 23.8 | 26.5 | 25.8 | 28.8 |
| " " Rocky Mountain region | 24.8 | 27.8 | 24.8 | 27.8 | 26.5 | 29.7 |
| Hemlock, west coast | 25.3 | 28.3 | 25.3 | 28.3 | 26.8 | 30.0 |
| Larch, western | 22.0 | 24.6 | 23.1 | 25.8 | 25.8 | 28.8 |
| Oak, red and white | 24.8 | 27.8 | 26.1 | 29.3 | 27.7 | 31.1 |
| Pine, southern | ........ | 27.3 | ........ | 28.6 | ........ | 31.1 |
| " dense | 22.6 | 25.3 | 23.8 | 26.5 | 25.8 | 28.8 |
| Redwood | 22.2 | 24.8 | 23.4 | 26.1 | 25.6 | 28.6 |
| Spruce, red, white, Sitka | 24.8 | 27.8 | 25.6 | 28.7 | 27.5 | 30.8 |

## 4. STEEL COLUMNS

**TYPES.**—Two general types of steel columns are in use: 1. Rolled shapes; 2. Built-up sections. The rolled shapes are easily fabricated, accessible for painting, neat in appearance where they are not covered, and convenient in making connections. A disadvantage is the probability that thick sections are of lower-strength material than thin sections because of the difficulty of adequately rolling the thick material. For the effect of thickness of material on yield point, see *Trans.* A.S.C.E., vol. xcviii, p. 1377, 1933.

**Table 4.—Strength of Columns of Intermediate Length, Expressed as a Percentage of Strength of Short Columns**

A.S.T.M. Standards, 1933, D245–33

Values for expression $\{1 - \frac{1}{3}(l/Kd)^4\}$ in formula [8]

| $K$ | Ratio of Length to Least Dimension in Rectangular Timbers, $l/d$ | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 |
| 22 | 97 | 96 | 95 | 93 | 91 | 88 | 85 | 81 | 77 | 72 | 67 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 23 | 98 | 97 | 95 | 94 | 92 | 90 | 87 | 84 | 81 | 77 | 72 | 67 | ... | ... | ... | ... | ... | ... | ... | ... |
| 24 | 98 | 97 | 96 | 95 | 93 | 92 | 89 | 87 | 84 | 80 | 76 | 72 | 67 | ... | ... | ... | ... | ... | ... | ... |
| 25 | 98 | 98 | 97 | 96 | 94 | 93 | 91 | 89 | 86 | 83 | 80 | 76 | 72 | 67 | ... | ... | ... | ... | ... | ... |
| 26 | 99 | 98 | 97 | 96 | 95 | 93 | 92 | 91 | 89 | 86 | 83 | 80 | 76 | 72 | 67 | ... | ... | ... | ... | ... |
| 27 | 99 | 98 | 98 | 97 | 96 | 95 | 93 | 92 | 90 | 88 | 85 | 82 | 79 | 74 | 71 | 67 | ... | ... | ... | ... |
| 28 | 99 | 98 | 98 | 97 | 96 | 95 | 94 | 93 | 91 | 89 | 87 | 85 | 82 | 79 | 75 | 71 | 67 | ... | ... | ... |
| 29 | 99 | 99 | 98 | 98 | 97 | 96 | 95 | 94 | 92 | 91 | 89 | 87 | 84 | 82 | 79 | 75 | 71 | 67 | ... | ... |
| 30 | 99 | 99 | 98 | 98 | 97 | 97 | 96 | 95 | 94 | 92 | 90 | 88 | 86 | 84 | 81 | 78 | 75 | 71 | 67 | ... |
| 31 | 99 | 99 | 99 | 98 | 98 | 97 | 96 | 95 | 94 | 93 | 92 | 90 | 88 | 86 | 84 | 81 | 78 | 75 | 71 | 67 |

NOTE: This table can also be used for columns not rectangular, the $l/d$ being equivalent to $0.289\ l/k$, where $k$ is the least radius of gyration of the section.

**GENERAL PRINCIPLES IN DESIGN.**—The design of steel columns is always a cut-and-try method, as no law governs the relation between area and radius of gyration of the section. A column of given area is selected, and the amount of load that it will carry is computed by the proper formula. If the allowable load so computed is less than that to be carried, a larger column is selected and the load for it computed, the process being repeated until a proper section is found.

A few general principles should guide in proportioning columns. The radius of gyration should be approximately the same in the two directions at right angles to each other; the slenderness ratio of the separate parts of the column should not be greater than that of the column as a whole; the different parts should be adequately connected in order that the column may function as a single unit; the material should be distributed as far as possible from the center line in order to increase the radius of gyration. See pp. 17–74 to 17–78 for tables of safe loads on rolled steel columns and data on their selection.

**STEEL COLUMN FORMULAS.**—A variety of steel column formulas are in use, differing mostly in the value of unit stress allowed with various values of $l/k$. See Eshbach, Handbook of Engineering Fundamentals, p. 5–42, forming Vol. 1, of this series, for a discussion of the several formulas.

**TESTS ON STEEL COLUMNS.**—Following the collapse of the Quebec Bridge in 1907 as a result of a column failure, the A.S.C.E., the A.R.E.A., and the U. S. Bureau of Standards co-operated in tests of full-size steel columns. The results of these tests are reported in *Trans.* A.S.C.E., vol. lxxxiii. pp. 1583–1688, 1919–20, and are summarized in Eshbach, Handbook of Engineering Fundamentals, pp. 5–43, 5–44. The tests showed that, for columns of the proportions commonly used, the effect of variation in the steel, kinks, initial stresses, and similar defects in the column was more important than the effect of length. They also showed that the thin metal gave definitely higher strength, per unit area, than the thicker metal of the same type of section.

## 5. ECCENTRIC AND TRANSVERSE LOADS ON COMPRESSION MEMBERS

**ECCENTRIC LOADS ON SHORT COMPRESSION MEMBERS.**—Where a direct push acting on a member does not pass through the centroid but at a distance $e$, inches, from it, both direct and bending stresses are produced. For short compression members in which column action may be neglected, the direct unit stress is $P/A$, where $P$ = total load, lb., and $A$ = area of cross-section, sq. in. The bending unit stress is $Mc/I$, where $M = Pe$ = bending moment, lb.-in.; $c$ = distance, in., from the centroid to the fiber in which the stress is desired; $I$ = moment of inertia, in.$^4$. The total unit stress at any point in the section is $s = P/A + Pec/I$, or $s = (P/A)(1 + ec/k^2)$, since $I = Ak^2$, where $k$ = radius of gyration, in.

**ECCENTRIC LOADS ON COLUMNS.**—Various column formulas must be modified when the loads are not balanced, that is, when the resultant of the loads is not in line with the axis of the column. If $P$ = load, lb., applied at a distance $e$ in. from the axis, bending moment $M = Pe$. Maximum unit stress $s$, lb. per sq. in., due to this bending moment alone, is $s = Mc/I = Pec/Ak^2$, where $c$ = distance, in., from the axis to the most remote fiber on the concave side; $A$ = sectional area, sq. in.; $k$ = radius of gyration in the direction of the bending, in. This unit stress must be added to the unit stress that would be induced if the resultant load were applied in line with the axis of the column. The modified formulas, expressed in allowable load per unit of area, are:

Modified Rankine's formula for eccentric loads (see p. 7–35):

$$P/A = s/\{1 + \phi(l/k)^2 + (ec/k^2)\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

Modified straight-line formula for eccentric loads (see p. 7–36):

$$P/A = \{s - (Cl/k)\}/\{1 + (ec/k^2)\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

The secant formula, p. 7–35, also can be used for columns that are eccentrically loaded if $e$ is taken as the actual eccentricity plus the assumed initial eccentricity.

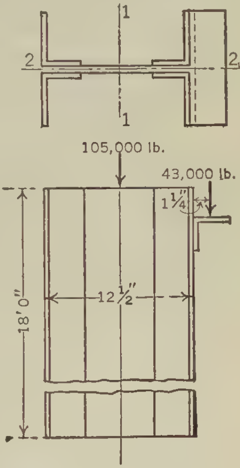


FIG. 2

EXAMPLE.—Design a steel column 18 ft. long to carry an axial load of 105,000 lb. and an additional load of 43,000 lb. which comes from a crane runway, and which is 1 1/4 in. beyond the outside edge of the column.

The formula used will be the A.I.S.C. formula,

$$P/A = 18{,}000/\{1 + (l^2/18{,}000k^2)\},$$

The shape selected will be a plate and four angles, Fig. 2.

Using 10,000 lb. per sq. in. as a preliminary estimate for the first selection, one plate 12 × 5/16 in. and four angles 5 × 3 1/2 × 3/8 in. will have an area of 15.95 sq. in. The safe load for this column will now be found. The moment of inertia $I_{1-1}$ is 412 in.$^4$, the radius of gyration $k_{1-1}$ is 5.08 in., and the radius of gyration $k_{2-2}$ is 2.08 in.

Considering the safe load with respect to axis 2–2 about which neither load is eccentric,

$$P = \frac{15.95 \times 18{,}000}{1 + \dfrac{(18 \times 12)^2}{18{,}000 \times (2.08)^2}} = 179{,}400 \text{ lb.}$$

Consider now the safe load about axis 1–1, with respect to which the load of 105,000 lb. is axial, and the load of 43,000 lb. has an eccentricity of 7 1/2 in. The load of 43,000 lb. will produce a stress due to its eccentricity, and this stress will reduce by that amount the unit load that the column can carry. The safe load will therefore be

$$P = 15.95\left(\frac{18{,}000}{1 + \dfrac{(18 \times 12)^2}{18{,}000 \times (5.08)^2}} - \frac{43{,}000 \times 7.5 \times 6.25}{412}\right) = 183{,}000 \text{ lb.}$$

The area exceeds that needed by about 3 sq. in.; the next smaller size is one plate 12 × 5/16 in. and four angles 5 × 3 1/2 × 5/16 in. The area is 13.99 sq. in., moment of inertia, $I_{1-1}$ = 356 in.$^4$, $k_{1-1}$ = 5.04 in., and $k_{2-2}$ = 2.03 in. With respect to the axis 2–2

$$P = \frac{13.99 \times 18{,}000}{1 + \dfrac{(18 \times 12)^2}{18{,}000 \times (2.03)^2}} = 154{,}600 \text{ lb.}$$

with respect to axis 1–1,

$$P = 13.99\left(\frac{18{,}000}{1 + \dfrac{(18 \times 12)^2}{18{,}000 \times (5.04)^2}} - \frac{43{,}000 \times 7.5 \times 6.25}{356}\right) = 149{,}300 \text{ lb.}$$

As the load to be carried is 148,000 lb. this column is satisfactory.

**COLUMN SUBJECTED TO TRANSVERSE OR CROSS-BENDING LOADS.**—(See Seely, Resistance of Materials, 1st Ed., p. 247.) A compression member that is subjected to cross-bending loads may be considered to be (1) a beam subjected to end thrust as discussed on p. 7–32 or (2) a column subjected to cross-bending loads, depending on the relative magnitude of the end thrust and cross-bending loads, and on the dimensions of the member. The various column formulas may be modified so as to include the effect of cross-bending loads. In this form they are:

Modified Rankine's Formula for transverse loads:

$$s = (P/A)\{1 + \phi(l^2/k^2) + (Pyc/Ak^2) + (Mc/Ak^2) \quad . \quad . \quad . \quad . \quad . \quad [12]$$

Modified straight-line formula for transverse loads:

$$P/A = \{s - (Cl/k)\} - (Pyc/Ak^2) - (Mc/Ak^2) \quad . \quad . \quad . \quad . \quad . \quad [13]$$

Modified secant formula for transverse loads:

$$s = (P/A)\{1 + (e + y)(c/k^2) \sec (l/2k)(\sqrt{P/AE}) + (Mc/Ak^2) \quad . \quad . \quad [14]$$

In these formulas, $s$ = maximum unit stress on concave side, lb. per sq. in.; $P$ = axial end load, lb.; $A$ = cross-sectional area, sq. in.; $M$ = moment due to cross-bending load, lb.-in.; $y$ = deflection due to cross-bending load, in.; $k$ = radius of gyration, in.; $l$ = length of column, in.; $e$ = assumed initial eccentricity, in.; $c$ = distance, in., from axis to the most remote fiber on the concave side; $C$ = experimental constant (see Table 2). The modified straight-line formula, like the straight-line formula, cannot be used to solve for $s$.

## 6. CAST-IRON COLUMNS

Hollow cast-iron columns are used to a limited extent in buildings of moderate height, because of cheapness, and because of the fact that they save space, as compared to wood columns. They have the disadvantage of uncertain strength, due to hidden defects, variation in wall thickness resulting from shifting of the core, and internal stresses from uneven cooling of the casting. Because of these variables, it is difficult to devise a satisfactory formula for design. Their use generally is governed by municipal building laws. The New York City building law, 1929, gives the formula $P/A = 9000 - 40(l/k)$, where $P$ = safe load, lb.; $A$ = area of section, sq. in.; $l$ = length, in.; $k$ = least radius of gyration. Most building laws specify a minimum metal thickness of $^1/_2$ to $^3/_4$ in.; minimum diameter, 6 in.; maximum length, 20 × diameter, or 70 × least radius of gyration. If it is necessary to use a wall thicker than 2 in., a steel column will be more economical.

Tables of safe loads allowed by the New York City Building Code on round and square hollow cast-iron columns will be found in Eshbach, Handbook of Engineering Fundamentals, pp. 5–46, 5–47, forming Vol. 1 of this series.

# SHAFTS

## 1. DEFINITIONS

**TORSIONAL STRESS.**—A bar is under torsional stress when it is held fast at one end and a force acts at the other end to twist the bar. In a round bar, Fig. 1, with a constant force acting, the straight line $ab$ becomes the helix $ad$, and a radial line in the cross-section, $ob$, moves to the position $od$. The angle $bad$ remains constant while the angle $bod$ increases with the length of the bar. Each cross-section of the bar tends to shear off the one adjacent to it, and in any cross-section the shearing stress at any point is normal to a radial line drawn through the point. Within the shearing proportional limit, a radial line of the cross-section remains straight after the twisting force has been applied, and the unit shearing stress at any point is proportional to its distance from the axis.

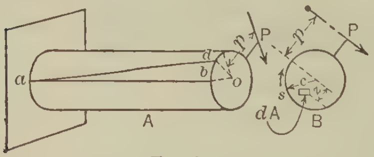


Fig. 1

**TWISTING MOMENT,** $T$, is equal to the product of the resultant, $P$, of the twisting forces, multiplied by its distance from the axis, $p$.

**RESISTING MOMENT,** $T_r$, in torsion, is equal to the sum of the moments of the unit shearing stresses acting along a cross-section with respect to the axis of the bar. If $dA$ is an elementary area of the section at a distance of $z$ units from the axis of a circular shaft (Fig. 1B), and $c$ is the distance from the axis to the outside of the cross-section where the unit shearing stress is $s$, then the unit shearing stress acting on $dA$ is $(sz/c)dA$, its moment with respect to the axis is $(sz^2/c)dA$, and the sum of all the moments of the unit shearing stresses on the cross-section is $\int(sz^2/c)dA$. In this expression the factor $\int z^2dA$ is the polar moment of inertia of the section with respect to the axis. Denoting this by $J$, the resisting moment may be written $sJ/c$.

**THE POLAR MOMENT OF INERTIA** of a surface about an axis through its center of gravity and perpendicular to the surface is the sum of the products obtained by multiplying each elementary area by the square of its distance from the center of gravity of its surface; it is equal to the sum of the moments of inertia taken with respect to two axes in the plane of the surface at right angles to each other passing through the center of gravity. It is represented by $J$, inches$^4$. For the cross-section of a round shaft,

$$J = (1/32)(\pi d^4), \quad \text{or} \quad 1/2\,\pi r^4 \qquad [1]$$

for a hollow shaft

$$J = (1/32)[\pi(d^4 - d_1^4)] \qquad [2]$$

where $d$ is the outside and $d_1$ the inside diameter, in inches, or

$$J = 1/2[\pi(r^4 - r_1^4)] \qquad [3]$$

where $r$ is the outside and $r_1$ the inside radius, in inches.

**THE POLAR RADIUS OF GYRATION,** $k_p$, sometimes is used in formulas; it is defined as the radius of a circumference along which the entire area of a surface might be concentrated and have the same polar moment of inertia as the distributed area. For a solid circular section,

$$k^2_p = (1/8)d^2 \qquad [4]$$

for a hollow circular section,

$$k^2_p = (1/8)(d^2 - d_1^2) \qquad [5]$$

## 2. DETERMINATION OF TORSIONAL STRESSES

**TORSION FORMULA FOR ROUND SHAFTS.**—The conditions of equilibrium require that the twisting moment, $T$, be opposed by an equal resisting moment, $T_r$, so that for the values of the maximum unit shearing stress, $s_s$, within the proportional limit, the torsion formula for round shafts becomes

$$T_r = T = s_s\,J/c \qquad [6]$$

If $s_s$ is in lb. per sq. in., then $T_r$ and $T$ must be in lb.-in., $J$ is in in.$^4$, and $c$ is in in. For solid round shafts having a diameter, $d$, in.,

$$J = (1/32)\,\pi d^4, \quad \text{and} \quad c = 1/2\,d \qquad [7]$$

and

$$T = (1/16)\,\pi d^3\,s_s, \quad \text{or} \quad s_s = 16\,T/\pi d^3 \qquad [8]$$

For hollow round shafts

$$J = \pi(d^4 - d_1^4)/32, \quad c = (1/2)\,d \qquad [9]$$

and the formula becomes

$$T = s_s\,\pi(d^4 - d_1^4)/16\,d \quad \text{or} \quad s_s = 16\,Td/\pi(d^4 - d_1^4) \qquad [10]$$

The torsion formula applies only to solid circular shafts or hollow circular shafts, and then only when the load is applied in a plane perpendicular to the axis of the shaft and when the shearing proportional limit of the material is not exceeded.

**SHEARING STRESS IN TERMS OF HORSEPOWER.**—If the shaft is to be used for the transmission of power, the value of $T$, lb.-in., in the above formulas becomes $63{,}030\,H/n$, where $H$ = horsepower to be transmitted; $n$ = r.p.m. The maximum unit shearing stress, lb. per sq. in., then, is:

$$\text{For solid round shafts:} \quad s_s = 321{,}000\,H/nd^3 \qquad [11]$$

$$\text{For hollow round shafts:} \quad s_s = 321{,}000\,Hd/n(d^4 - d_1^4) \qquad [12]$$

If $s_s$ is taken as the allowable unit shearing stress, the diameter, $d$, inches, necessary to transmit a given horsepower at a given shaft speed can then be determined. These formulas give the stress due to torsion only, and allowance must be made for any other loads, as the weight of shaft and pulleys, and tension in belts.

**ANGLE OF TWIST.**—When the unit shearing stress $s_s$ does not exceed the proportional limit, the angle *bod* (Fig. 1) for a solid round shaft may be computed from the formula

$$\theta = Tl/E_s\,J \qquad [13]$$

where $\theta$ = angle, radians; $l$ = length of shaft, in.; $E_s$ = shearing modulus of elasticity of the material; $T$ = twisting moment, lb.-in. Values of $E_s$ for different materials are: steel, 12,000,000; wrought iron, 10,000,000; and cast iron, 6,000,000.

When the angle of twist on a section begins to increase in a greater ratio than the twisting moment, it may be assumed that the shearing stress on the outside of the section has reached the proportional limit. The shearing stress at this point may be determined by substituting the twisting moment at this instant in the torsion formula.

**TORSION OF NON-CIRCULAR CROSS-SECTIONS.**—The analysis of shearing stress distribution along non-circular cross-sections of bars under torsion is complex. By drawing two lines at right angles through the center of gravity of a section before twisting, and observing the angular distortion after twisting, it has been found from many experiments that in non-circular sections the shearing unit stresses are not proportional to their distances from the axis. Thus in a rectangular bar there is no shearing stress at the corners

### Table 1.—Approximate Formulas for Maximum Shearing Stress and Angle of Twist in Members Subjected to Torsion

(From Seely's Advanced Mechanics of Materials)

$s_s$ = shearing unit stress, lb. per sq. in.; $T$ = twisting moment, lb.-in.; $\theta$ = angle of twist, radians, in length, $l$, in.; $J$ = polar moment of inertia, in.$^4$; $E_S$ = shearing modulus of elasticity; $A$ = area of cross-section, sq. in.; $y$ = distance of most remote edge from center of bar, in. Tests of brittle metal are those made by Bach, *Elastizität u. Festigkeit*, 1924, except for those marked *, which are by Kommers, *Am. Mach.*, vol. xl, p. 941, 1914.

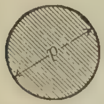
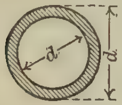
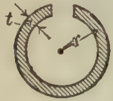
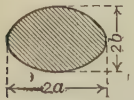
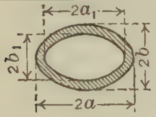
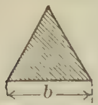
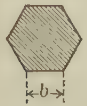
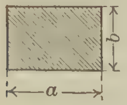
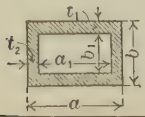
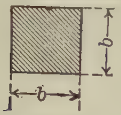
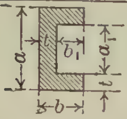
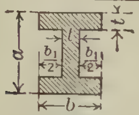
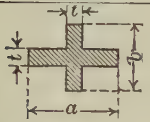
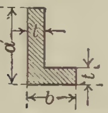
| Shape | Maximum Unit Stress | | Angle of Twist |
|---|---|---|---|
| | Mathematical Analysis | Tests of Brittle Material | |
|  | $s_s = \frac{16T}{\pi d^3}$ | | $\theta = \frac{Tl}{E_s J}$ |
|  | $s_s = \frac{16Td}{\pi(d^4 - d_1^4)}$ | | $\theta = \frac{32Tl}{\pi(d^4 - d_1^4)E_s}$ |
|  | $s_s = \frac{3T}{2\pi r t^2}$ | | $\theta = \frac{3Tl}{2\pi r t^3 E_s}$ |
|  | $s_s = \frac{2T}{\pi a b^2}$ | | $\theta = \frac{T(a^2 + b^2)l}{\pi a^3 b^3 E_s}$ |
|  | $s_s = \frac{T}{2\pi a_1 b_1^2 k(1 + k)}$<br>$k = \frac{a - a_1}{a_1} \doteq \frac{b - b_1}{b_1}$ | | $\theta = \frac{T(a^2 + b^2)l}{4\pi a_1^3 b_1^3 k E_s}$ |
|  | $s_s = \frac{20T}{b^3}$ | | $\theta = \frac{46.2Tl}{b^4 E_s}$ |
|  | $s_s = \frac{1.09T}{b^3}$ | | $\theta = \frac{0.967Tl}{b^4 E_s}$ |
|  | $s_s = \frac{(3a + 1.8b)T}{a^2 b^2}$ | $s_s = \frac{9T}{2ab^2}$ | $\theta = \frac{4\pi^2 TlJ}{A^4 E_s}$<br>Error is small where $a > 2b$ |
|  | $s_s = \frac{T}{2t(a - i)(b - t_1)}$ | $s_s = \frac{9bT}{2(b^3 a - b_1^3 a_1)}$ | $\theta = \frac{Tl(at + bt_1 - t^2 - t_1^2)}{2tt_1(a - t)^2(b - t_1)^2 E_s}$ |

**Table 1.**—*Continued*

| Shape | Maximum Unit Stress | | Angle of Twist |
|---|---|---|---|
| | Mathematical Analysis | Tests of Brittle Material | |
|  | $s_s = \frac{4.8T}{b^3}$ | | $\theta = \frac{7.11Tl}{b^4E_s}$ |
|  | | $s_s = \frac{9T}{2t^2(a + 2b_1)}$<br>$s_s = \frac{9T*}{2t^2a}$ | $\theta = \frac{36.2TlJ*}{A^4E_s}$ |
|  | | $s_s = \frac{9T}{2t^2(a + 2b_1)}$ | |
|  | | $s_s = \frac{9T}{2t^2(a + b - t)}$ | |
|  | | $s_s = \frac{9T}{2t^2(a + b - t)}$ | |
| Any compact section without re-entrant angles. | $s_s = \frac{4\pi^2JT}{A^4y}$ | | $\theta = \frac{kJTl}{A^4E_s}$<br>$k = 4\pi^2$ for ellipse.<br>$= 40$ to $42$ for rectangles. |

of the sections and the stress at the middle of the wide side is greater than at the middle of the narrow side. In an elliptical bar the shearing stress is greater along the flat side than at the round side.

It has been found by tests (Bach, *Elastizität u. Festigkeit*; and Young, *Bull.* 4, School of Engg. Research, Univ. of Toronto) as well as by mathematical analysis that the torsional resistance of a section, made up of a number of rectangular parts, is approximately equal to the sum of the resistances of the separate parts. It is on this basis that nearly all the formulas for non-circular sections have been developed. For example, the torsional resistance of an I–beam is approximately equal to the sum of the torsional resistances of the web and the outstanding flanges. In an I–beam in torsion the maximum shearing stress will occur at the middle of the side of the web, except where the flanges are thicker than the web, and then the maximum stress will be at the mid-point of the width of the flange. Re-entrant angles, as those in I-beams and channels, are always a source of weakness in members subjected to torsion. Table 1 gives approximate values of the maximum unit shearing stress $s_s$ and the angle of twist $\theta$ induced by twisting bars of various cross-sections, it being assumed that $s_s$ is not greater than the proportional limit and that the modulus of elasticity, $E_s$, remains a constant.

**ULTIMATE STRENGTH IN TORSION.**—In a torsion failure, the outer fibers of a section are the first to shear and the rupture extends toward the axis as the twisting is continued. The torsion formula for round shafts has no theoretical basis after the shearing stresses on the outer fibers exceed the proportional limit, as the stresses along the section then are no longer proportional to their distances from the axis. It is convenient, however, to compare the torsional strength of various materials by using the formula to compute values of $s_s$ at which rupture takes place. These computed values of the maximum stress sustained before rupture are somewhat higher for iron and steel than the ultimate strength of the materials in direct shear. Computed values of the ultimate strength in torsion are found by experiment to be as follows: Cast iron, 30,000 lb. per sq. in.; wrought iron, 55,000 lb. per sq. in.; medium steel, 65,000 lb. per sq. in.; timber, 2000 lb. per sq. in. These computed values of twisting strength may be used in the torsion formula to determine the probable twisting moment that will cause rupture of a given round bar or to determine the size of a bar that will be ruptured by a given twisting moment. In design, large factors of safety should be taken, especially when the stress is reversed as in reversing engines and when the torsional stress is combined with other stresses as in shafting.

**EFFECT OF KEYWAY IN SHAFT.**—The sharp re-entrant angles in keyways produce high local stresses at these points. Based on studies made by hydrodynamical and soap-film methods, the local stress in the corner has been determined as equal to the stress in a similar shaft without a keyway, multiplied by a constant, $K$, which depends on the radius of the corner. The constants are, approximately:

| Radius of corner, in........ | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 |
|---|---|---|---|---|---|---|---|
| $K$...................... | 5.4 | 3.4 | 2.7 | 2.3 | 2.1 | 2.0 | 1.9 |

Under steady loading these high stresses probably are not very important, as a ductile material allows an adjustment to take place, but if the shaft is subjected to repeated loads a crack may start at the corner.

Experiments by Moore (*Bull.* 42, Engg. Exp. Sta., Univ. of Ill., 1909) on shafts from $1\,^1/_4$ to $2\,^1/_4$ in. diameter showed that keyways reduced the elastic strength and the stiffness but did not reduce the ultimate strength. If $E'$ = ratio of strength at the proportional limit of a shaft with a keyway to strength at the proportional limit of a shaft without a keyway, $N$ = ratio of the angle of twist of a shaft with a keyway to that of a shaft without a keyway, $w$ = width of keyway divided by diameter of shaft, and $h$ = depth of keyway divided by diameter of shaft, the ratios obtained were: $E' = 1.0 - 0.2\,w - 1.1\,h$; $N = 1.0 + 0.4\,w + 0.7\,h$, for ranges of $w$ from 0.25 to 0.50, and of $h$ from 0.125 to 0.1875.

# CYLINDERS, PLATES AND ROLLERS

## 1. CYLINDERS

**THIN CYLINDERS UNDER INTERNAL PRESSURE.**—A cylinder is regarded as thin when the thickness of the wall is small compared with the diameter. It is assumed that in such cases the tensile stress across a longitudinal section is uniformly distributed over the thickness of the wall. If $p$ = internal pressure, lb. per sq. in.; $l$ = length of cylinder, in.; $t$ = thickness of wall, in.; $d$ = diameter of cylinder, in.; and $s$ = tensile stress, lb. per sq. in.,

$$pdl = 2\,stl \quad \text{or} \quad s = pd/2t \qquad [1]$$

For resistance to rupture in a circumferential seam, due to pressure on the ends of the cylinder, *i.e.*, the tensile stress across a transverse section,

$$p\pi d^2/4 = s\,t\pi d \quad \text{or} \quad s = pd/4t \qquad [2]$$

Comparing formulas [1] and [2], it is seen that the resistance to rupture around the circumference is twice the resistance to longitudinal rupture; for this reason cylindrical boilers are commonly single riveted in the circumferential seams and double or triple riveted in the longitudinal seams.

In using the above formulas to design cylindrical shells or piping, thickness $t$ must be increased to compensate for rivet holes in the joints. Water pipes, particularly those of cast iron, require a high factor of safety to provide security against shocks caused by water ram or rough handling before they are laid. Steam pipes also require a high factor of safety to resist shocks due to condensation and expansion of steam.

Formula [2] applies also to the stresses in the walls of a thin hollow sphere, hemisphere, or dome. This analysis does not apply where holes are cut in the cylinder for rivets or for other purposes. Where holes are cut, the tensile stresses must be found by the method used in riveted joints. See p. 9–49.

**Thickness of a Domed Head of a Boiler.**—If $s$ = safe unit tensile stress, $d$ = diameter of shell, in., and $t$ = thickness of unit shell, $t = pd \div 2s$; but the thickness of a hemispherical head of the same diameter is $t = pd \div 4s$. Hence, if we make the *radius* of curvature of a domed head equal to the diameter of the boiler, we shall have $t = 2\,pd/4s = pd/2s$ or the thickness of such a domed head will be equal to the thickness of the shell.

**THIN CYLINDERS UNDER EXTERNAL PRESSURE.**—Prof. Reid T. Stewart (*Trans.* A.S.M.E., xxvii, 730) made two series of tests on Bessemer steel lap-welded tubes 3 to 10 in. diam. One series was made on tubes 8 5/8 in. outside diam. with the different commercial thicknesses of wall, and in lengths of 2 1/2, 5, 10, 15 and 20 ft. between transverse joints tending to hold the tube in a circular form. A second series was made on single lengths of 20 ft. Seven sizes, from 3 to 10 in. outside diam., in all the commercial thicknesses obtainable, were tested. The principal conclusions drawn from the research were:

1. The length of tube, between transverse joints tending to hold it in circular form, has no practical influence on the collapsing pressure of a commercial lap-welded tube if this length is not less than about 6 diameters of tube.

2. The formulas based on this research, for the collapsing pressures of lap-welded Bessemer steel tubes, for all lengths greater than 6 diameters, are:

$$P = 1000\{1 - \sqrt{1 - 1600(t^2/d^2)}\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

$$P = 86{,}670(t/d) - 1386, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where $P$ = collapsing pressure, lb. per sq. in.; $d$ = outside diameter of tube, in.; $t$ = thickness of wall, in.

Formula [3] is for values of $P$ less than 581 lb., or for values of $t/d$ less than 0.023, while formula [4] is for values greater than these. When applying these formulas to practice, a suitable factor of safety must be used.

3. The apparent fiber stress under which the different tubes failed varied from about 7000 lb. for the relatively thinnest to 35,000 lb. per sq. in. for the relatively thickest walls. Since the average yield point of the material was 37,000 and the tensile strength 58,000 lb. per sq. in., it would appear that the strength of a tube subjected to a fluid collapsing pressure is not dependent alone on either the elastic limit or ultimate strength of the material constituting it. The element of greatest weakness in a tube is its departure from roundness, even when this departure is relatively small.

Table 1 is a condensed statement of the principal results of the tests. Table 2 has been calculated from Stewart's formulas.

Tests by Prof. A. P. Carmen (*Bull.* No. 5, Engg. Experiment Station, Univ. of Ill., 1906), showed that with ratios of $t/d$ of less than 0.025, for cold-drawn seamless steel tubes,

$$P = 50{,}200{,}000(t/d)^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

and for seamless brass tubes,

$$P = 25{,}150{,}000(t/d)^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

notation being as before.

**Table 1.—Collapsing Pressure of Lap-welded Steel Tubes**

(Stewart's Experiments)

| Outside Diameters, 8 5/8 in. Length of Pipe, 20 ft. | | | | | | | |
|---|---|---|---|---|---|---|---|
| Thickness, in. | Length, ft. | Bursting Pressure, lb. per sq. in. | Average | Outside Diam., in. | Thickness, in. | Bursting Pressure, lb. per sq. in. | Average |
| 0.176 | 2.21 | 815–1085 | 977 | 3 | 0.112 | 1550–2175 | 1860 |
| 0.180 | 4.70 | 525–705 | 792 | 3 | 0.143 | 2575–3350 | 2962 |
| 0.181 | 10.08 | 455–650 | 565 | 3 | 0.188 | 3700–4200 | 4095 |
| 0.184 | 14.71 | 425–610 | 548 | 4 | 0.119 | 860–1030 | 964 |
| 0.185 | 19.72 | 450–625 | 536 | 4 | 0.175 | 2050–2540 | 2280 |
| 0.212 | 2.21 | 1240–1353 | 1314 | 4 | 0.212 | 3075–3375 | 3170 |
| 0.212 | 4.70 | 805–975 | 907 | 4 | 0.327 | 5425–5625 | 5560 |
| 0.217 | 10.50 | 700–960 | 841 | 6 | 0.130 | 450–640 | 524 |
| 0.219 | 12.79 | 750–1115 | 905 | 6 | 0.167 | 715–1110 | 928 |
| 0.268 | 2.14 | 1475–2200 | 1872 | 6 | 0.222 | 1200–2075 | 1797 |
| 0.274 | 4.64 | 1345–2030 | 1684 | 6 | 0.266 | 1750–2890 | 2441 |
| 0.272 | 9.64 | 1150–1908 | 1583 | 7 | 0.160 | 515–675 | 592 |
| 0.273 | 14.64 | 1250–1725 | 1485 | 7 | 0.242 | 1525–1850 | 1680 |
| 0.268 | 19.64 | 1250–1520 | 1419 | 7 | 0.279 | 1835–2445 | 2147 |
| 0.311 | 2.16 | 2290–2490 | 2397 | 8.64 | 0.185 | 450–625 | 536 |
| 0.306 | 4.64 | 1795–2325 | 2073 | 8.66 | 0.268 | 1250–1520 | 1419 |
| 0.306 | 9.64 | 1585–2055 | 1807 | 8.67 | 0.354 | 1830–2180 | 2028 |
| 0.309 | 14.64 | 1520–2025 | 1781 | 10 | 0.165 | 210–240 | 225 |
| 0.302 | 19.75 | 1575–1960 | 1762 | 10 | 0.194 | 305–425 | 383 |
| ......... | ......... | ......... | ......... | 10 | 0.316 | 1275–1385 | 1319 |

**Table 2.—Collapsing Pressure of Lap-welded Tubes**

Calculated by Stewart's Formulas

| Thickness, in. | Outside Diameter | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 in. | 2 1/2 in. | 3 in. | 4 in. | 5 in. | 6 in. | 7 in. | 8 in. | 9 in. | 10 in. | 11 in. |
| | Collapsing Pressure, lb. per sq. in. | | | | | | | | | | |
| 0.10 | 2947 | 2081 | 1503 | 781 | ....... | ....... | ....... | ....... | ....... | ....... | ..... |
| 0.12 | 3814 | 2774 | 2081 | 1214 | 694 | 400 | ....... | ....... | ....... | ....... | ...... |
| 0.14 | 4671 | 3468 | 2659 | 1647 | 1041 | 636 | 400 | 286 | 217 | ....... | ...... |
| 0.16 | 5548 | 4161 | 3236 | 2081 | 1387 | 925 | 595 | 400 | 297 | 232 | 187 |
| 0.18 | 6414 | 4854 | 3814 | 3514 | 1734 | 1214 | 843 | 564 | 400 | 306 | 244 |
| 0.20 | 7281 | 5548 | 4392 | 2947 | 2081 | 1503 | 1090 | 781 | 542 | 400 | 314 |
| 0.22 | 8148 | 6241 | 4970 | 3381 | 2427 | 1792 | 1338 | 997 | 733 | 525 | 400 |
| 0.24 | 9014 | 6934 | 5548 | 3814 | 2774 | 2081 | 1586 | 1214 | 935 | 694 | 512 |
| 0.26 | 9881 | 7628 | 6125 | 4248 | 3121 | 2370 | 1833 | 1431 | 1118 | 867 | 633 |
| 0.28 | ...... | 8321 | 6703 | 4601 | 3468 | 2669 | 2081 | 1647 | 1310 | 1041 | 820 |
| 0.30 | ...... | 9014 | 7281 | 5114 | 3814 | 2947 | 2328 | 1864 | 1503 | 1214 | 978 |
| 0.32 | ...... | 9708 | 7859 | 5548 | 4161 | 3236 | 2576 | 2081 | 1696 | 1387 | 1135 |
| 0.34 | ...... | ....... | 8437 | 5981 | 4508 | 3525 | 2824 | 2297 | 1888 | 1561 | 1293 |
| 0.36 | ...... | ....... | 9014 | 6414 | 4854 | 3814 | 3071 | 2514 | 2081 | 1734 | 1450 |
| 0.38 | ...... | ....... | 9592 | 6848 | 5201 | 4103 | 3319 | 2731 | 2273 | 1907 | 1603 |
| 0.40 | ...... | ....... | ....... | 7281 | 5548 | 4392 | 3567 | 2947 | 2466 | 2081 | 1766 |
| 0.42 | ...... | ....... | ....... | 7714 | 5894 | 4681 | 3814 | 3164 | 2659 | 2254 | 1923 |
| 0.44 | ...... | ....... | ....... | 8148 | 6241 | 4970 | 4062 | 3381 | 2851 | 2427 | 2081 |
| 0.46 | ...... | ....... | ....... | 8581 | 6588 | 5259 | 4309 | 3598 | 3044 | 2601 | 2238 |
| 0.48 | ...... | ....... | ....... | 9014 | 6934 | 5548 | 4557 | 3814 | 3236 | 2774 | 2396 |
| 0.50 | ...... | ....... | ....... | 9448 | 7281 | 5887 | 4805 | 4031 | 3429 | 2947 | 2554 |

For an extended discussion see Strength of Thin Cylindrical Shells under External Pressure by Saunders and Windenburg, and The Collapsing Strength of Steel Tubes by Jasper and Sullivan (*Trans.* A.S.M.E., Applied Mechanics, September-December, 1931, pp. 207–245).

**COLLAPSING PRESSURE OF COPPER TUBES.**—An investigation conducted by M. G. Steele for Revere Copper & Brass, Inc., is reported in *Jour.*, Am. Soc. Nav. Engrs., Nov., 1939. This investigation showed that tubes properly designed to resist substantial internal pressures usually will withstand any external pressure to which they might be inadvertently subjected. The formulas developed for collapse of steel and brass tubes do not appear to be applicable to copper tubes. The investigation consisted of collapsing tubes of various diameters and gages in different alloys, with an accompanying correlation of the physical properties of the tube material. The results of the investigation have been translated into the curves Figs. 1 and 2, which may be used as design data for copper tubes subject to external collapsing pressure. The curves apply to three types of tubes, as follows:

Type K, generally used for underground services, plumbing purposes and severe water conditions; also for gas, steam and oil lines, and other industrial uses. It is supplied in hard or soft temper.

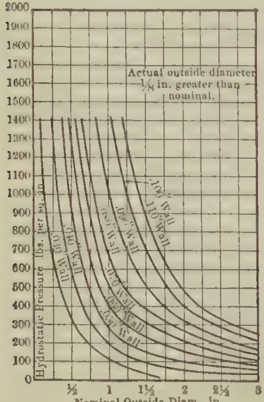


Fig. 1. Collapsing Pressure of Soft Copper Tubing.

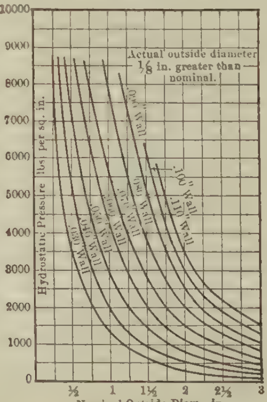


Fig. 2. Collapsing Pressure of Hard Copper Tubing.

Type L, used for general plumbing purposes and normal water conditions; also for gas, steam and oil lines and other industrial uses. Supplied in hard or soft temper.

Type M, used for general plumbing purposes with solder fittings only. Suitable for normal water conditions; also for gas, steam and oil lines and other industrial uses. Supplied usually in hard temper only.

In using the curves of Figs. 1 and 2, a suitable safety factor should be allowed to cover shock loads, water-hammer, etc. The usual factor of safety is 3 but this should be varied according to conditions and to engineering judgment. The values for hard tubes are considered more useful than those for soft tubes, hard tubes being used more often where collapse is a possibility. Soft tubes are not much used in mechanical applications except where bending is requisite. They usually are used to conduct water, gas or other fluids and the external pressures encountered are seldom great enough to cause collapse.

**THICK CYLINDERS.**—Timoshenko and Lessells (Applied Elasticity) and Swain (Strength of Materials) give the following formulas for the stresses in thick hollow cylinders under internal and external pressure. Under internal pressure only, the maximum tensile stress $s_t$ in a tangential direction on the inner surface is

$$s_t = p\{(a^2 + b^2)/(b^2 - a^2)\} \quad \ldots\ldots\ldots \quad [7]$$

where $p$ = internal pressure, lb. per sq. in.; $a$ = inside radius and $b$ = outside radius, in. Under internal pressure maximum compressive stress in a radial direction is on the inside and equals $p$ lb. per sq. in.

When the cylinder is subjected to an external pressure $p_e$ the maximum compressive unit stress will be at the inner surface and is

$$s_c = 2\,p_e\,b^2/(b^2 - a^2) \quad \ldots\ldots\ldots \quad [8]$$

In *shrinkage fits* a hollow cylinder or collar is forced over a cylinder having an outer diameter slightly greater than the inside diameter of the outer cylinder. This will cause compressive stresses in the inner cylinder and tensile stresses in the outer cylinder. Timoshenko and Lessells (Applied Elasticity) give the equation for the pressure $p$, lb. per sq. in. between the cylinders as

$$p = \frac{E\delta}{b}\,\frac{(b^2 - a^2)\,(c^2 - b^2)}{2b^2(c^2 - a^2)} \quad \ldots\ldots\ldots \quad [9]$$

where $E$ = modulus of elasticity, $a$ = inner radius of inner cylinder, $b$ = outer radius of inner cylinder, $c$ = outer radius of outer cylinder, $\delta$ = amount by which the outer radius of the inner cylinder exceeds the inner radius of the outer cylinder. With the pressures known, external pressure on the inner cylinder and internal pressure on the outer cylinder, the stresses can be computed by means of equations [7] and [8].

A comparison of the various theories of failure of thick cylinders is given in Seely's Advanced Mechanics of Materials from which the following is abstracted:

For brittle materials, the *maximum stress theory* is recommended. It gives:

For internal pressure only, $t = r_1[\{\sqrt{(s_w + p)/(s_w - p)}\} - 1]$ . . . . . . . [10]

For external pressure only, $t = r_1[\{\sqrt{s_w/(2p_e + s_w)}\} - 1]$ . . . . . . . . [11]

For ductile materials the *maximum strain theory* is recommended. It is used by the U. S. Government for the design of guns. For internal pressure only this theory gives

$$t = r_1\left[\left\{\sqrt{\frac{E\epsilon_w + (1 - m)p}{E\,\epsilon_w - (1 + m)p}}\right\} - 1\right] \quad \ldots\ldots \quad [12]$$

and for external pressure only

$$t = r_1\left[\left\{\sqrt{\frac{E\,\epsilon_w}{E\,\epsilon_w - 2p_e}}\right\} - 1\right] \quad \ldots\ldots\ldots \quad [13]$$

where $t$ = thickness, in.; $s_w$ = allowable working unit stress, lb. per sq. in.; $p$ = internal pressure, and $p_e$ = external pressure, lb. per sq. in.; $r_1$ = inner radius of cylinder, in.; $E$ = modulus of elasticity; $m$ = Poisson's ratio; $\epsilon_w$ = allowable working unit deformation of the material.

**CORRUGATED FURNACES.**—The Board of Trade, Lloyd's Register, British Corporation and the Bureau Veritas have adopted the following uniform formula (see Kempe's Engineer's Yearbook, 1932) for allowable pressures on Fox type corrugated furnaces:

$$WP = 480(t - 1)/D \quad \ldots\ldots\ldots \quad [14]$$

where $D$ = external diameter, in., measured at the bottom of the corrugations; $t$ = thickness of furnace plate in 32nds of an inch, measured at the bottom of the corrugations or camber; $WP$ = allowable working pressure, lb. per sq. in. Thickness of plate is limited to $^{13}/_{16}$ in.

**HOLLOW COPPER BALLS.**—Hollow copper balls are used as floats in boilers or tanks, to control feed and discharge valves, and regulate the water-level. *The Locomotive* (Oct., 1891) gives two empirical rules for determining the thickness of a copper ball, which is to work under an external pressure, as follows:

1. Thickness = (diameter, in. × pressure, lb., per sq. in.)/16,000.
2. Thickness = (diameter × $\sqrt{\text{pressure}}$)/1240.

These rules give the same result for a pressure of 166 lb. per sq. in. only.

## 2. PLATES

**CIRCULAR FLAT PLATES.**—Table 3 gives equations for deflection and stresses in circular plates based mainly on the theoretical analyses developed by Grashof. (See Morley's Strength of Materials; Seely's Advanced Mechanics of Materials; Timoshenko and Lessells' Applied Elasticity.) Experience shows that the strength of flat plates is greater than is indicated by these equations.

**ELLIPTICAL FLAT PLATES.**—An approximate formula for the maximum stress $s$ in elliptical plates, simply supported at the edge, having a major axis $2a$, a minor axis $2b$, a thickness $t$, and carrying a distributed load of $w$ lb. per sq. in., is

$$s = \{(3a - 2b)/a\} \times (wb^2/t^2) \quad . \; . \; . \; . \; . \; . \; . \; . \quad [15]$$

**SQUARE FLAT PLATES.**—Where the load is $w$ lb. per sq. in. and the plate is supported on the four edges, each $b$ in. long, the maximum moment and maximum stress will be along a diagonal. The moment $M$ per unit length of diagonal will be $wb^2/24$, and, where $t$ = thickness, in., the unit stress will be

$$s = wb^2/4t^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [16]$$

With edges fixed and load distributed the maximum stress at the center of each edge is, approximately, $s = 0.20\, wb^2/t^2$. Nichols found in experiments with steel plates

$$s = 0.141\, wb^2/t^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [17]$$

and Bach found

$$s = 0.19\, wb^2/t^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [18]$$

**RECTANGULAR FLAT PLATES.**—For a distributed load $w$ lb. per sq. in., with supports along the four sides, and assuming a uniform distribution of moment along a

**Table 3.—Maximum Stresses and Deflection for Circular Flat Plates**

$m$ = Poisson's ratio; $P$ = concentrated load, lb.; $w$ = distributed load, lb. per sq. in.; $r$ = radius of plate, in.; $t$ = thickness of plate, in.; $E$ = modulus of elasticity.

| Type of Load and Support | Maximum Stress at Edge | Maximum Stress at Center | Maximum Deflection at Center, $m = 1/3$ |
|---|---|---|---|
| Uniform load, edge simply supported | | $\frac{3}{8}\frac{wr^2}{t^2}(3 + m)$ | $\frac{2}{3}\frac{wr^4}{Et^3}$ |
| Uniform load, edge fixed | Radial direction<br>$\frac{3}{4}\frac{wr^2}{t^2}$<br>Tangential direction<br>$\frac{3}{4}\frac{wmr^2}{t^2}$ | $\frac{3}{8}\frac{wr^2}{t^2}(1 + m)$ | $\frac{1}{6}\frac{wr^4}{Et^3}$ |
| Concentrated load at center, edge supported | | Infinite | $\frac{5}{3}\frac{Pr^2}{\pi Et^3}$ |
| Concentrated load at center, edge fixed | Radial direction<br>$\frac{3P}{2\pi t^2}$<br>Tangential direction<br>$\frac{3}{2}\frac{Pm}{\pi t^2}$ | Infinite | $\frac{2}{3}\frac{Pr^2}{\pi Et^3}$ |
| Uniform load, supported on central area of radius $r_0$ | | $\frac{3}{2}\frac{wr^2}{t^2}\left[(1 + m)\log_e\frac{r}{r_0} + \frac{1}{4}(1 - m)\left(1 - \frac{r_0^2}{r^2}\right)\right]$ | $\frac{3}{16}(1 - m)(7 + 3m)\frac{wr^4}{Et^3}$ |
| Load on central area of radius $r_0$, edge simply supported | | $\frac{3(1 + m)P}{2\pi t^2}\left[\frac{1}{m + 1} + \log_e\frac{r}{r_0} - \left(\frac{1 - m}{1 + m}\right)\frac{r_0^2}{4r^2}\right]$ | $\frac{5}{3}\frac{Pr^2}{\pi Et^3}$ |
| Load on central area of radius $r_0$, edge fixed | | $\frac{3(1 + m)P}{2\pi t^2}\left(\log_e\frac{r}{r_0} + \frac{r_0^2}{4r^2}\right)$<br>$r$ must be $> 1.7\, r_0$ | $\frac{2}{3}\frac{Pr^2}{\pi Et^3}$ |

**Table 4.—Values of Constants $D$, $K_1$ and $K_2$ for Rectangular Plates with Supported Edges and Uniformly Distributed Loads**

$a$ = long side; $b$ = short side; Poisson's ratio = 0.3

| $a/b$ | $D$ | $K_1$ | $K_2$ | $a/b$ | $D$ | $K_1$ | $K_2$ |
|---|---|---|---|---|---|---|---|
| 1.0 | 0.0443 | 0.0479 | 0.0479 | 1.8 | 0.1017 | 0.0948 | 0.0479 |
| 1.1 | .0530 | .0553 | .0494 | 1.9 | .1064 | .0985 | .0471 |
| 1.2 | .0616 | .0626 | .0501 | 2.0 | .1106 | .1017 | .0464 |
| 1.3 | .0697 | .0693 | .0504 | 3.0 | .1336 | .1189 | .0404 |
| 1.4 | .0770 | .0753 | .0506 | 4.0 | .1400 | .1235 | .0384 |
| 1.5 | .0843 | .0812 | .0500 | 5.0 | .1416 | .1246 | .0375 |
| 1.6 | .0906 | .0862 | .0493 | ∞ | .1422 | .1250 | .0375 |
| 1.7 | .0964 | .0908 | .0486 | | | | |

diagonal, Seely gives (Advanced Mechanics of Materials, p. 141) the stress as

$$s = a^2 b^2 w / \{2t^2(a^2 + b^2)\} \qquad [19]$$

where $a$ is the length of the long side and $b$ the length of the short side.

For rectangular plates fixed along the four sides, Bach found experimentally for ductile materials that the coefficient $K$ in the moment equation $M = Kwb^2$ was such that the moment along the fixed edge and along the center was about the same, and that the values of $K$ (see Seely, Advanced Mechanics of Materials, p. 144) for different ratios of the length of the sides $a$ and $b$ were about as follows:

| $b/a$ = | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| $K$ = | 0.63 | 0.62 | 0.56 | 0.48 | 0.40 | 0.32 |

Timoshenko and Lessells (Applied Elasticity) give an analysis which shows that the maximum moment at the center of the plate, in strips of unit width through the center of the plate, parallel to the short and long sides respectively, when the sides are simply supported, and the load is $w$ lb. per sq. in., uniformly distributed over the plate, is given by the equations

$$M_{1\,\max.} = K_1 wb^2 \qquad [20a]$$

$$M_{2\,\max.} = K_2 wb^2 \qquad [20b]$$

The approximate deflection $y$ is given by

$$y_{\max.} = Dwb^4/Et^3 \qquad [21]$$

where $t$ = thickness, in.; $E$ = modulus of elasticity; $K_1$, $K_2$, and $D$ are constants given in Table 4.

For rectangular plates with the edges fixed, the maximum deflection is given by the same equation as for the plates with supported edges, but with constant $D$ as follows, and Poisson's ratio being taken as 0.3:

| $a/b$ | 1.00 | 1.25 | 1.50 | 1.75 | 2.00 | ∞ |
|---|---|---|---|---|---|---|
| $D$ | 0.0138 | 0.0199 | 0.0240 | 0.0264 | 0.0277 | 0.0284 |
| $K$ | 0.0513 | 0.0665 | 0.0757 | 0.0817 | 0.0829 | 0.0833 |

The maximum bending moment in a strip of unit width is at the middle of the longer

**Table 5.—Formulas for Bending Moments in Rectangular Flat Slabs**

Values are per unit width. Poisson's ratio = 0, $a$ = longer side, $b$ = shorter side, $\alpha = b/a$

| | | Moments in Span $b$ | | Moments in Span $a$ | |
|---|---|---|---|---|---|
| | | At center of edge | At center of slab | At center of edge | At center of slab |
| Rectangular Slabs | Four edges simply supported | 0 | $\frac{1/8\ wb^2}{1 + 2\alpha^3}$ | 0 | $\frac{wb^2}{48}(1 + \alpha^2)$ |
| | Span $b$ fixed; span $a$ simply supported | $\frac{1/12\ wb^2}{1 + 0.2\ \alpha^4}$ | $\frac{1/24\ wb^2}{1 + 0.4\ \alpha^4}$ | 0 | $\frac{wb^2}{80}(1 + 0.3\ \alpha^2)$ |
| | Span $a$ fixed; span $b$ simply supported | 0 | $\frac{1/8\ wb^2}{1 + 0.8\ \alpha^2 + 6\alpha^4}$ | $\frac{1/8\ wb^2}{1 + 0.8\ \alpha^4}$ | $0.015\,wb^2\left(\frac{1 + 3\alpha^2}{1 + \alpha^4}\right)$ |
| | All edges fixed | $\frac{1/12\ wb^2}{1 + \alpha^4}$ | $\frac{1/8\ wb^2}{3 + 4\alpha^4}$ | $1/24\ wb^2$ | $0.009\,wb^2(1 + 2\alpha^2 - \alpha^4)$ |
| Elliptical slab with fixed edges; diameters $a$ and $b$; $b/a = \alpha$ | | $\frac{1/12\ wb^2}{1 + 2/3\ \alpha^2 + \alpha^4}$ | $\frac{1/24\ wb^2}{1 + 2/3\ \alpha^2 + \alpha^4}$ | $\frac{1/12\ wb^2\ \alpha^2}{1 + 2/3\ \alpha^2 + \alpha^4}$ | $\frac{1/24\ wb^2\ \alpha^2}{1 + 2/3\ \alpha^2 + \alpha^4}$ |

fixed side and is equal to $M_{\max} = Kwb^2$, $K$ being as given above, and $a$ and $b$ being the lengths of the long and short sides respectively.

Professor Westergaard in a study of reinforced-concrete slabs supported on columns has computed the moments in medium-thick rectangular and elliptical plates (*Proc. A.C.I.*, 1921). These are plates in which there is neither an appreciable absorption of energy by the vertical shearing stresses nor by the stretching of the middle plane of the plate. By means of a number of simplifying assumptions the values for the moments as given in Table 5 were found.

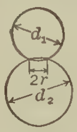
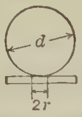
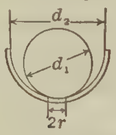
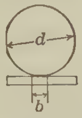
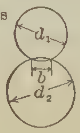
## 3. STRESSES IN SOLIDS UNDER PRESSURE

**CYLINDRICAL AND SPHERICAL ROLLERS.**—Approximate equations for areas of contact and pressures have been worked out for a number of cases (Timoshenko and Lessells' Applied Elasticity). These are given in Table 6.

**HERTZ EQUATIONS.**—A general mathematical solution of the problem of the stresses produced by the pressure of one solid upon another, of which cylindrical and spherical rollers are a special case, was made by H. Hertz (English translation in Miscellaneous Papers by H. Hertz). The Hertz equations are very complex and deal only with surface stresses. Thomas and Hoersch (*Bull.* 212, Engg. Exp. Sta., University of Illinois, 1930), as a part of an investigation of railroad rails and the development of transverse fissures in them, extended the Hertz equations to a study of the shearing stresses within the solid. They developed methods, too long to be given here, of computing the magnitude

**Table 6.—Areas of Contact and Pressures with Two Surfaces in Contact**

Poisson's ratio = 0.3; $P$ = load, lb.; $P_1$ = load per in. of length, lb.; $E$ = modulus of elasticity

| Character of Surfaces | Maximum Pressure, $s$, at Center of Contact, lb. per sq. in. | Radius, $r$, or Width, $b$, of Contact Area, in. |
|---|---|---|
| Two spheres  | $s = 0.616\sqrt[3]{PE^2\left(\frac{d_1+d_2}{d_1\,d_2}\right)^2}$ | $r = 0.881\sqrt[3]{\frac{P}{E}\left(\frac{d_1\,d_2}{d_1+d_2}\right)}$ |
| Sphere and plane  | $s = 0.616\sqrt[3]{\frac{PE^2}{d^2}}$ | $r = 0.881\sqrt[3]{\frac{Pd}{E}}$ |
| Sphere and hollow sphere  | $s = 0.616\sqrt[3]{PE^2\left(\frac{d_2-d_1}{d_1\,d_2}\right)^2}$ | $r = 0.881\sqrt[3]{\frac{P}{E}\left(\frac{d_1\,d_2}{d_2-d_1}\right)}$ |
| Cylinder and plane  | $s = 0.591\sqrt{\frac{P_1\,E}{d}}$ | $b = 2.15\sqrt{\frac{P_1\,d}{E}}$ |
| Two cylinders  | $s = 0.591\sqrt{P_1\,E\left(\frac{d_1+d_2}{d_1\,d_2}\right)}$ | $b = 2.15\sqrt{\frac{P_1}{E}\left(\frac{d_1\,d_2}{d_1+d_2}\right)}$ |

and location of the maximum shearing stresses produced in solids under pressure, such as in railroad rails under a wheel load. They have obtained experimental verification of the correctness of their equations by a "strain-etch" method, but point out that the mathematical equations derived do not hold except for thick material and where the ratio of the length of contact to the width of contact is large.

In a cylinder of radius $R$, in., under a load $P'$, lb. per in. of length, on a plane, assuming Poisson's ratio as $1/4$ the maximum shearing unit stress $s_s = 677\sqrt{P'/R}$, the width of contact equals $0.000564\sqrt{P'R}$, the depth to the point of maximum shear is $0.000222\sqrt{P'R}$, and the maximum intensity of pressure is $2256\sqrt{P'/R}$. In the case of crossed cylinders, the empirical equation is

$$s_s = 11{,}750\,P^{1/3}/(R_1/R_2)^{0.271}R_2^{2/3} \qquad [22]$$

where $R_1$ = radius of larger cylinder; $R_2$ = radius of smaller cylinder; $P$ = total pressure, lb., between the two solids; the ratio $R_1/R_2$ is only true within the limits of 1 and 8. Table 7 gives a number of values computed by the methods of Thomas and Hoersch.

Stribeck made extensive experiments on ball bearings in 1898–1900, and his reports are translated in *Trans.* A.S.M.E., vol. xxix, p. 421.

**Table 7.—Shearing Stresses in Solids under Pressure**

| Case | Radius, $R_1$, in. | Length of Cylinder, in. | Radius, $R_2$, in. | Load, lb. | Area in Contact, sq. in.* | Maximum Shearing Stress, lb. per sq. in. | Depth below Surface of Point of Maximum Shearing Stress, in. |
|---|---|---|---|---|---|---|---|
| Crossed cylinders.... | 16.5 | ....... | 14 | 25,000 | 0.215 | 56,500 | 0.121 |
| Crossed cylinders.... | 40 | ....... | 14 | 60,000 | 0.502 | 59,800 | 0.179 |
| Cylinder on plane... | 16.5 | 2 | Plane | 25,000 | 0.512 | 18,600 | 0.100 |
| Cylinder on plane... | 40 | 2 | Plane | 60,000 | 1.236 | 18,600 | 0.243 |
| Cylinder on plane... | 30 | ....... | Plane | 39,800 lb. per in. length | ........ | 24,700 | .......... |
| Sole plates Galveston Causeway Bascule Bridge.......... | 309 | ....... | Plane | 112,400 lb. per in. length | Width 3.45 in. | 12,900 | 1.355 |

* Computed by approximate equation

Where rollers are moving, as at the end of a truss or girder or with a rolling lift bridge, under relatively high pressures, the plate on which the roller rests is the critical part. The metal of the plate tends to flow longitudinally under the backward and forward motion. Wilson (*Bull.* 191, Engg. Exp. Sta., University of Illinois, 1929) reports on experiments with rollers of the size used in rolling lift bridges, between 116 in. and 476 in. radius, and concludes that the bearing capacity of a plate depends upon its thickness, the tensile strength of the material, and the diameter of the roller. He proposes the formula

$$P = (12{,}000 + 80\,D)\{p - 13{,}000)/23{,}000\} \qquad [23]$$

where $P$ = safe working load, lb. per in. width of plate; $D$ = diameter of roller, in.; $p$ = yield point strength of the material in tension, lb. per sq. in. The thickness of plate, in., should not be less than $(1.0 - 0.004\,D)$ with $D$ not less than 120 in. Steel bridges usually have one end on rollers to allow for expansion. The usual allowable load for expansion rollers is $P = 600\,D$; $P$ = load, lb. per lineal inch of roller, and $D$ = diameter of roller, in.

# TESTING OF MATERIALS

## 1. EQUIPMENT

**TESTING MACHINES.**—Machines for testing material are of varying construction and capacity, for testing specimens in tension, in compression, in torsion and for other specific purposes. Usually the mechanism for applying the load to the specimen is independent of the weighing apparatus. By this arrangement the accuracy of the weighing is not affected by the deformation of the specimen which takes place as the load is applied. In machines for testing bars in tension or compression, the ends of the test-piece are fastened in the two heads of the machine One of the heads is moved by screws, driven by a train of gears, causing a gradually increasing stress in the test-piece. Some machines are built so that the head is moved by hydraulic pressure, but the revolving screw construction is more common in the United States, although the use of hydraulic pressure machines is increasing rapidly (1937).

**THE COMPOUND LEVER MACHINE** derives its name from the weighing mechanism. The load may be applied by revolving screws or by hydraulic pressure. In either case, the tension or compression acting on the test-piece is transformed into a downward force bearing on a platform resting on levers equipped with hardened steel knife edges and arranged like those in a platform scale. The force acting on the platform is transmitted through the levers to a graduated scale beam which is balanced by a poise to give the load on the test-piece.

**The Screw-power Compound Lever Testing Machine (Olsen or Riehlé Machines)** is the type in most common use in the U. S. Power is supplied by a belt drive or by a direct-connected motor, and is transmitted through a series of gears to vertical screws which operate the crosshead of the machine. The force applied by the belt, multiplied many times by the gearing and the screws, is applied to the test specimen by the crosshead. Tension specimens are placed between the crosshead and the upper head of the testing machine. Compression specimens or cross-bending specimens are placed between the crosshead and the weighing table. As the crosshead moves downward, the specimen transmits downward pressure to the weighing table, which is supported on compound levers fitted with knife-edge bearings. These compound levers transmit the force, reduced, to a simple lever, which, in turn, transmits it still further reduced, to the weighing beam. As the load is applied the beam is kept in balance by moving the poise. The position of the latter, with the beam in balance, indicates the load on the specimen. These machines are built by Tinius Olsen Co., Philadelphia, and Riehlé Bros. Testing Machine Div. of American Machine & Metals, Inc., New York.

**THE DIRECT-ACTING HYDRAULIC MACHINE** has a head moved by hydraulic pressure, the intensity of which is read on a pressure gage. The load on the test-piece is determined by multiplying the pressure reading by a constant which depends on the area of the cylinder. Due to variations in the plunger friction, these machines are not adapted for accurate testing of small specimens, but are used to advantage in tests requiring big loads to rupture the specimen, and where slight variations in accuracy are permitted.

**The Amsler Machine** is a machine of the hydraulic type built in Schaffhausen, Switzerland, which has considerable use in the U. S. A plunger moving in a cylinder near the top of the machine raises and lowers a rigid framework with respect to a fixed tension head at the bottom and a fixed compression head at the top of the machine. Oil is pumped into the cylinder to move a plunger around to fit the cylinder. This renders packing unnecessary and eliminates a main source of error in the hydraulic machine. A plunger in a branch line from the main cylinder to a smaller one actuates a pendulum weight by means of a projecting arm. This displacement of the pendulum rotates a pointer on a dial from which the load is read directly. The Amsler machine can be handled by a single observer.

**The Emery-Tatnall Machine** is a hydraulic machine in which the load is applied through a plunger moving in a cylinder under oil pressure. The distinctive feature is the load-weighing device, called a weighing capsule. This is not connected to the load-applying mechanism, and is unaffected by friction between cylinder and plunger. One of the testing heads rests on a thin flat diaphragm over a shallow recess filled with oil. A slight movement of the diaphragm decreases the volume of the recess and creates pressure on the oil, which is recorded on a gage calibrated to read the load. One precaution is necessary: the weighing capsule must be kept filled with oil to avoid recorded loads being seriously in error.

**VERIFICATION OF TENSION-COMPRESSION TESTING MACHINES.**—A.S.T.M. Tentative Standard, E4–34T, *Proc.* A.S.T.M. vol. xxxiv, Pt. 1, p. 1233, 1934, lists four methods of verifying testing machines, in the order of their accuracy, *viz.*:

**1. Standard Weights** are placed on the weighing mechanism of the testing machine. This is the most accurate method but has the following limitations: (*a*) only a limited range of load can be covered; (*b*) non-portability of standard weights; (*c*) not applicable to horizontal testing machines.

**2. Proving Levers.**—These are simple levers whose fulcrums bear on the weighing platform; knife-edges on the short arms of the levers bear on the crosshead of the machine. At the ends of the long arms of the levers, standard weights are hung so that, from the ratio of the arms, the force acting on the weighing table may be computed. The range of load possible with this method is greater than that of the standard weights method, but it is not great enough for very large testing machines.

**3. Elastic Calibration Device** which consists of an elastic member, which previously has been calibrated by noting the deformation at a series of loads. For a primary standard, the loads should be obtained by dead weights or proving levers. When verifying a testing machine, load is applied to the device by the machine to be tested and the deformation noted. The difference between the load reading on the machine under test and the actual load, as determined for the observed deformation from the calibration curve of the device, represents the error in the machine being verified.

**4. Comparison Method.**—A series of test specimens (" companion specimens ") cut from the same piece of metal are necessary. Half of the specimens are tested in a machine

**Table 1.—Large Testing Machines in United States**

(All are of the hydraulic type)

| Place | Date Installed | Capacity, lb. | Type | Length of Maximum Specimen, ft.* |
|---|---|---|---|---|
| Phoenix Iron Co., Phoenixville, Pa...... | 1886 | 2,800,000 Tension<br>2,800,000 Compression | Hor. | 50<br>55 |
| American Bridge Co., Ambridge, Pa..... | 1905 | 4,000,000 Tension | Hor. | 42 |
| U. S. Steel Co., McKeesport, Pa........ | .... | 1,200,000 Tension<br>800,000 Compression | Hor. | 40<br>32 |
| U. S. Bureau of Stds., Washington, D. C. | 1912 | 10,000,000 Compression | Vert. | 65 |
| U. S. Bureau of Stds., Washington, D. C. | 1913 | 1,150,000 Tension<br>2,300,000 Compression | Hor. | 33 |
| Roebling Wire Works, Trenton, N. J..... | 1928 | 2,500,000 Tension<br>2,500,000 Compression | Vert. | 25 |
| U. S. Reclamation Service, Denver, Colo. | 1928 | 4,000,000 Compression | Vert. | 6¼ |
| Univ. of Illinois, Urbana, Ill............. | 1929 | 3,000,000 Tension<br>3,000,000 Compression | Vert. | 37 |
| Univ. of Calif., Berkeley, Calif.......... | 1931 | 3,000,000 Tension<br>4,000,000 Compression | Vert. | 33½ |

* The maximum length of tension specimen will depend on the amount of elongation which must be provided for before failure occurs.

which has been verified by one of the above methods, and the other half in the machine under test. The difference in load reading for the same deformation, between the two machines, is the error in the machine being tested. This method should be used only when none of the other methods above may be applied.

**FLEXURE TESTS.**—The regular tension-compression testing machines are used for flexure tests by arranging the specimen so that the end supports bear on the weighing table, as shown in Fig. 1.

**TORSION TESTING MACHINES.**—In Prof. Thurston's machine, invented in 1872, the specimen is held horizontally with its ends secured in two spindles. One of these spindles is rotated by power. The other spindle is connected to a pendulum carrying a heavy weight which resists the turning force as it deviates from its vertical position, thus producing a torque whose magnitude is shown by a gage or dial.

Power torsion machines are built so that the two ends of the specimen are held in rotating chucks. The power is applied at any desired speed by gearing connected to one of the chucks. The chuck at the other end is prevented from rotating by a system of levers connected to a scale beam which is balanced by a poise. The position of the poise indicates the twisting moment in in.-lb.

**IMPACT TESTS.**—The essential parts of an impact testing machine are: a moving weight which strikes the specimen with sufficient kinetic energy to cause fracture; an anvil or block on which the specimen is supported; a device for measuring the kinetic energy of the moving weight after it has broken the specimen.

**Drop-weight Machines.**—The impact is obtained by a hammer of known weight falling from a predetermined height on the specimen. In the Freemont machine the loss of kinetic energy in the hammer due to impact is obtained by having the weight, after breaking the specimen, strike a platen placed on calibrated springs, and measuring the downward movement of the platen against the resistance of the springs. Another method is used in the Hatt-Turner machine, in which a brass stylus is attached to the falling hammer; during impact the stylus is forced by a spring against a sheet of metallized paper mounted on a drum which rotates about its vertical axis, thus causing a curve to be plotted. The speed of the drum is determined from a second curve plotted on the metallized paper by a stylus fastened to one prong of a tuning fork. From the two curves the changes in velocity of the hammer as well as the stress-deformation relations may be calculated.

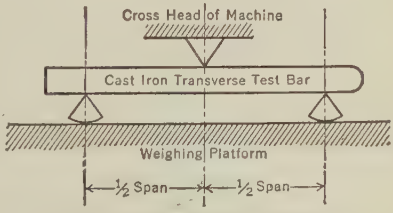


Fig. 1. Arrangement for Flexure Test

**The Charpy Pendulum Machine** is used for testing small notched specimens which are placed horizontally, with the ends supported by the anvil. The hammer strikes the specimen midway between the supports opposite the notch, and after breaking the specimen continues its swing upward. The energy used in breaking the specimen is the product of the weight of the pendulum and the difference between its height of fall and height of rise.

**The Izod Machine** is used for testing a notched specimen supported at its lower end in a vertical position as a cantilever; the pendulum strikes the specimen at a definite distance above the notch. See Energy or Dynamic Loads, p. 7–15.

**REPEATED STRESS TESTING MACHINE.**—The following types of machines are available for producing reversed bending: The Upton-Lewis for flat specimens; the short-specimen rotating beam; the long-specimen rotating beam; and the cantilever rotating beam. Only the long-specimen machine (sometimes known as the Sondricker or Farmer type) is here described. It consists of two rings or collars, rotating in a cage of ball bearings, for supporting the specimen which is 0.40 in. diameter and 13 in. long. The specimen is rotated by a motor; the number of revolutions is recorded by a counter attached to one end of the specimen. Suspended weights load the beam at two points. These act through ball bearings to produce a uniform bending moment over the central portion of the specimen. The number of cycles of completely reversed stress equals the number of revolutions. The speeds commonly used are from 1000 to 2000 r.p.m.

**EXTENSOMETERS.**—Several types of extensometers are used to measure elongation within the elastic limit during a test. In general, they comprise clamps which grip the test-piece at the gage marks and a micrometer or Ames dial to measure the distance between the clamps, which increases as the load is applied. With some extensometers, elongations thus may be measured to the nearest 0.0001 inch.

## 2. STANDARD TEST-PIECES

The results obtained in a test on material are affected by the method of testing and the size and shape of the specimen. Standards have been adopted by the A.S.T.M., the A.S.M.E. and various associations of manufacturers, to eliminate variations in results due to these causes. The standards of the A.S.T.M. are summarized below.

**SPECIMENS FOR TENSION AND BEND TESTS.**—(Extracts from Specifications A7–36, A9–36, A10–34, A113–33 for Structural Steel for Bridges, Buildings, Locomotives, etc., A.S.T.M. Standards, 1936). *a.* Test specimens shall be prepared for testing from the material in its rolled or forged condition, except that the specimens for annealed material shall be prepared from the material as annealed for use, or from a short length of a full section similarly treated; the specimen for rivet bars which have been cold drawn shall be normalized before testing. *b.* Test specimens shall be taken longitudinally and, except as noted in *c*, *d* and *e*, shall be of the full thickness or section of the material as rolled. For plates, shapes and flats the specimens may be machined to the form and dimensions of Fig. 2 or with both edges parallel. *c.* Tension test specimens for material over 1 1/2 in. thickness or diameter, except pins and rollers, may be machined to a thickness or diameter of at least 3/4 in. for a length of at least 9 in., or they may conform to the dimensions shown in Fig. 3. *d.* Bend test specimens for material over 1 1/2 in. thickness or diameter, except pins and rollers, may be machined to a thickness or diameter of at least 3/4 in. or to 1 × 1/2 in. in section. *e.* Tension test specimens for pins and rollers shall be taken so that the axis is 1 in. from the surface and shall conform to the dimensions shown in Fig. 3, while bend test specimens shall be 1 × 1/2 in. in section. *f.* The machined sides of rectangular bend specimens may have the corners rounded to a radius not over 1/16 in.

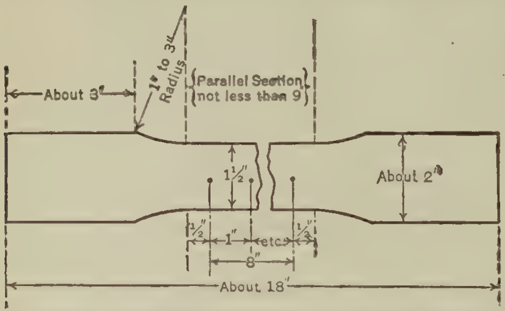


FIG. 2. Tension Test Specimen, 8-in. Gage Length

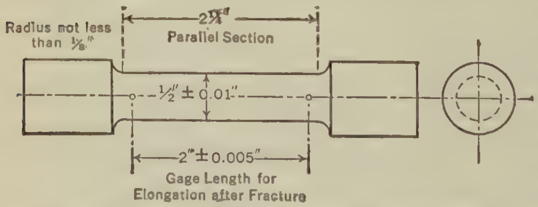
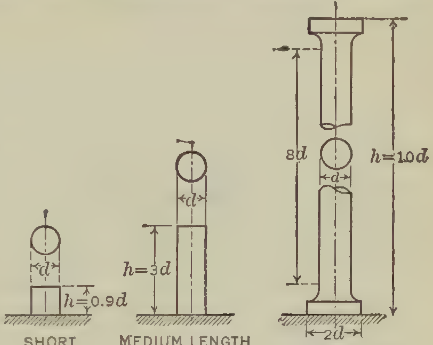


FIG. 3. Tension Test Specimen, 2-in. Gage Length

**SPECIMENS FOR COMPRESSION TESTS.**—A.S.T.M. Tentative Standards (1934) E9–34T recommends cylindrical specimens with plane ends, of the form shown in Fig. 4. The short specimen, Fig. 4*a*, is recommended for tests of bearing metals, or similar metals, which in service are in the form of a thin plate or shell, and which carry the load perpendicular to the surface. Medium length specimens, Fig. 4*b*, are recommended for tests to determine the general compressive strength qualities of metallic materials. Long specimens, Fig. 4*c*, are recommended for determining the modulus of elasticity in compression of metallic materials. Suggested diameters, with a tolerance of +0.01 in., are as follows: Short specimens, 1 in.; medium length specimens, 0.798 in., 1 in., and 1 1/8 in.; long specimens, 1 1/4 in. Dimensions of the test specimens should be given in reports of compression tests.

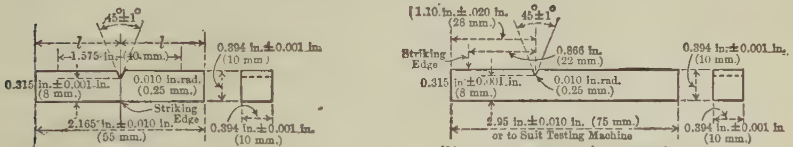


SHORT SPECIMEN (*a*) MEDIUM LENGTH SPECIMEN (*b*) LONG SPECIMEN (*c*)

FIG. 4. Compression Test Specimens

**SPECIMENS FOR IMPACT TESTS.**—A.S.T.M. Tentative Standard, 1934, E23–34T recommends the test specimens shown in Figs. 5*a* and 5*b* for use in Charpy type and Izod type impact machines, respectively.

**TENSION AND FLEXURE TESTS OF CAST IRON.**—See p. 20-07.


45°±1°
l
l
1.575 in. (40 mm.)
0.394 in.±0.001 in. (10 mm)
0.315 in.±0.001 in. (8 mm.)
0.010 in.rad. (0.25 mm.)
Striking Edge
2.165 in.±0.010 in. (55 mm.)
0.394 in.±0.001 in. (10 mm.)
1.10 in.±.020 in. (28 mm.)
45°±1°
Striking Edge
0.866 in. (22 mm.)
0.394 in.±0.001 in. (10 mm.)
0.315 in.±0.001 in. (8 mm.)
0.010 in.rad. (0.25 mm.)
2.95 in.±0.010 in. (75 mm.) or to Suit Testing Machine
0.394 in.±0.001 in (10 mm.)

(*a*) SIMPLE BEAM (CHARPY TYPE) SPECIMEN (*b*) CANTILEVER BEAM (IZOD TYPE) SPECIMEN

FIG. 5. Impact Test Specimens

## 3. PRECAUTIONS REQUIRED IN MAKING TESTS

**1. Inspection of Specimen.**—Examine the specimen for defects or irregularity of form. It should be of uniform section between gage marks on which measurements of elongation are to be made. The distance between gage marks, which usually is 8 in., should be marked off into half-inch divisions.

**2. Placing Specimen in Machine.**—The test-piece should be placed in an axial position, that is the line of stress of the machine should be in line with the axis of the specimen. Care should be taken to see that the ends of the specimen are properly gripped in the heads of the machine. When wedges are used, the full length of the wedge should bear against the ends of the specimen. The extensometer should be placed in position so that the clamps are at the gage marks and the specimen centered on the instrument.

**3. Speed of Testing.**—Regard must be had to the time occupied in making tests of certain materials. Wrought iron and soft steel can be made to show a higher strength than their actual apparent strength by keeping them under strain for a great length of time. A.S.T.M. Standard, E8–36, 1936, points out that the values of the yield point and ultimate strength may be increased by operating the testing machine at too high speed, and it recommends that the speed of the head of the testing machine shall be such that the load can be accurately weighed. In determining yield strength, the crosshead speed for the 2-in. gage length shall not exceed 0.125 in. per min.

In testing soft alloys, copper, tin, zinc, and the like, which flow under constant stress, their highest apparent strength is obtained by testing them rapidly. In recording tests of such materials the length of time occupied in the test should be stated.

**4. Increment of Load.**—A.S.T.M. Standard E8–33 suggests the following method for choosing increments in a tension test. An increment of load, about 1/10 of the estimated load corresponding to the knee of the stress-deformation diagram, is applied and the corresponding change of reading, Δ, of the extensometer noted. For the remainder of the test, the increments of load are made such that the extensometer readings change by approximately Δ.

## 4. INTERPRETATION OF TEST RESULTS

**DATA OBTAINED FROM TENSION TEST.**—The following data usually are obtained in testing by tension in a testing-machine a sample of a material:

*a.* The load at the elastic limit or yield point, lb.

*b.* The maximum load applied before rupture, lb.

*c.* The elongation of the piece, measured between gage points placed a stated distance apart before the test, usually 8 in. or 2 in.

*d.* Sometimes the area at the point of fracture.

The data now calculated are: 1. Tensile strength, lb. per sq. in. of original area; 2. Percentage elongation; 3. Elastic limit or yield point, lb. per sq. in. The modulus of elasticity is calculated either from the stress-strain graph, or as the ratio of the unit stress within the elastic limit to the corresponding unit strain. Due to the difficulty of making accurate measurements of the fractured area of a test piece and to the fact that elongation is more valuable than reduction of area as a measure of ductility, of toughness, or of energy absorbed before rupture, reduction in area frequently is not reported.

**Determination of Yield Point.**—Two methods for determining yield point of metals that have a " sharp knee " stress-deformation diagram are:

**1. " Drop of the Beam " Method.**—Load is applied to the specimen at a steady rate of increase, the beam being kept in balance by running out the poise. At the yield point the increase of load ceases temporarily, but the operator, running out the poise at a uniform rate, runs it a trifle beyond the balance position, and the beam of the machine drops for a brief, but appreciable, interval of time. In a machine fitted with a self-indicating weighing device there is a sudden halt in the motion of the load-indicating pointer, corresponding to the drop of the beam. The load at the " drop " or the " halt " is noted and the corresponding unit stress reported as the yield point.

**2. Total Deformation Method Using Dividers.**—An observer with a pair of dividers watches for visible elongation between two gage marks on the specimen. When visible stretch is observed the load is noted and the corresponding unit stress reported as the yield point.

**DETERMINATION OF YIELD STRENGTH.**—For determining yield strength, A.S.T.M. Standard E8–33, 1933, gives two methods:

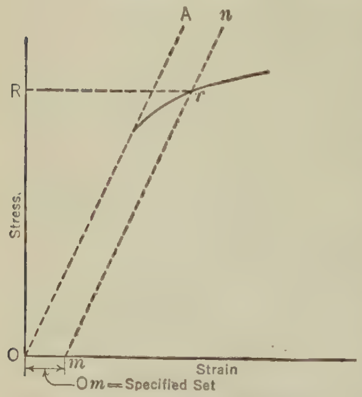


Fig. 6 Stress-deformation Diagram

**1. The Set Method.**—This method can be used, if desired, for materials having sharp-knee stress-deformation diagrams, but is especially adapted to materials whose stress-deformation diagram in the yield range is a smooth curve of gradual curvature.

For nearly all materials, if at any point on the stress-deformation diagram, such as $r$, Fig. 6, the load is released, the diagram for decreasing load will follow a line, $rm$, approximately parallel to the initial portion, $OA$, of the diagram for increasing load. $Om$ then is the approximate value of the permanent set after release of stress $OR$. The value of $Om$ is given in percentage of original gage length. Thus, to determine yield strength by the set method, it is necessary to secure data (autographic or numerical) from which a stress-deformation diagram may be drawn. Then with the stress-deformation diagram (Fig. 6) lay off $Om$ equal to the specified value of the set, and draw $mn$ parallel to $OA$ and thus locate $r$, the intersection of $mn$ with the stress-deformation curve. Draw $Rr$ parallel to the X-axis. $OR$ is the yield strength.

In reporting values of yield strength obtained by this method, the specified value of " set " used should be stated in parentheses after the term yield strength. Thus:

Yield strength (set = 0.1%) = 52,000 lb. per sq. in., indicates that, at a stress of 52,000 lb. per sq. in., the approximate permanent set of the material reached the value of 0.1% of the original gage length. (A value of 0.20% of the gage length has been used rather widely for metals.)

In using this method, an extensometer reading to 0.0001 in. per in. of gage length would be sufficiently sensitive for most materials. It will be observed that elastic limit and proportional limit are special values of yield strength, in which the specified limiting set is not measureable with the instruments used, and hence is considered to be zero.

**2. Approximate Method without Stress-deformation Diagram.**—For tests to determine the acceptance or rejection of material, whose stress-deformation characteristics are known from previous tests of similar material in which stress-deformation diagrams were plotted, the total deformation corresponding to the stress at which the specified permanent set occurs will be known within satisfactory limits; therefore, in such tests a specific total deformation may be used, and the stress on the specimen, when this total deformation is reached, is the value of the yield strength. The total deformation can be obtained satisfactorily by an extensometer, or in some cases by dividers, particularly if the surface of the specimen is so prepared that the line made by the dividers is a fine one, and if it is observed through a magnifying glass. This approximate method is recommended for use only after agreement between producer and consumer, with the understanding that check tests be made to obtain stress-deformation diagrams for use with the Set Method to settle any misunderstandings.

**DETERMINATION OF TENSILE STRENGTH (ULTIMATE).**—The tensile strength of a material is determined by dividing the maximum load carried in a tension test by the original area of cross-section of the specimen. It is important that the speed of testing should not be so high as to render uncertain the balancing of the beam of the testing machine, or to cause "surging" of a self-indicating load-measuring device.

**DETERMINATION OF ELONGATION AFTER FRACTURE** (A.S.T.M. Standard E8–33, 1933).—Measurement of elongation after fracture of tension test specimens can be made with sufficient accuracy by means of a pair of dividers and a scale. The elongation should not be reported for any tension test specimen which breaks outside the middle third of the gage length.

NOTE. If only a few specimens are available for testing, it is recommended that, in marking the gage length for measuring elongation after fracture, it be divided into eight parts (see Fig. 7), and that, if a specimen breaks inside the gage length but outside the middle third of the gage length, the requirement in the foregoing paragraph be waived, and the elongation be measured and reported as follows:

1. If specimen breaks in the middle half of the gage length (between *C* and *G*), elongation is measured directly over the stretched gage length. 2. If specimen breaks between *A* and *C* (or between *G* and *I*) but nearer *B* than *A* (or nearer *H* than *I*), elongation from *A* to *C* (or from *G* to *I*) is measured, and to this is added *twice* the measured elongation from *C* to *F* (or from *D* to *G*). 3. If specimen breaks within one-half a division of an end gage mark, the elongation from *A* to *E* (or from *E* to *I*) is measured and multiplied by 2. The above method can be applied to specimens of any gage length, to both turned and flat specimens, but is not applicable to brass specimens.

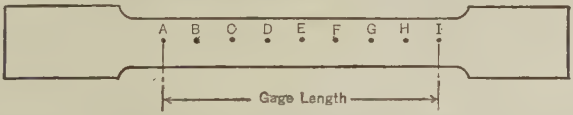


FIG. 7.—Method of Marking Gage Length for Measuring Elongation after Fracture

**DETERMINATION OF REDUCTION OF AREA.**—The measurement of reduction of the dimensions of a specimen after fracture may be made by the direct measurement with a micrometer of the smallest cross-section of the specimen. For round specimens this measurement best can be made by holding the broken pieces together in a vise or between centers and measuring the average diameter of the smallest section, using a micrometer fitted with rather sharp points. Reduction of area is calculated as the percentage reduction, based on original area.

**COMPRESSION TEST.**—The methods used to determine elastic limit, proportional limit, yield point, and yield strength in compression tests are, in general, the same as those used in tension tests. Compressive strength is determined by dividing the maximum load by the original area of cross-section of the specimen.

In a material which fails in compression with a shattering fracture, compressive strength has a very definite value. In materials which do not fail by a shattering fracture, the value obtained for compressive strength is an arbitrary value, depending on the degree of distortion which is regarded as indicating complete failure of the material.

## 5. MISCELLANEOUS TESTS

**PROOF TESTS.**—A proof test is one in which a part or machine, as a whole, is subjected to a load which equals or exceeds the design or working load but which does not test the part to destruction; for example, the hydrostatic test of cast-iron pipe at the foundry, where the pipe is subjected to a water pressure of about twice the normal working pressure.

**SHORT TIME TESTS FOR ENDURANCE LIMIT.**—Since the determination of the endurance limit for any given steel or metal takes considerable time, attempts have been made to devise short-time tests which will give, at least, some indication of the probable endurance limit. No method has been found which is entirely satisfactory or which has been generally accepted. Two methods have shown some promise of success, *viz.*, 1. The rise-in-temperature method; 2. The electrical resistance method.

**Rise-in-temperature Method.**—(Moore and Kommers, *Bull.* No. 124, Engg. Exp. Sta., Univ. of Ill., 1921). A standard rotating beam specimen is held at one end and, by an eccentric movement of the other end, a bending stress is produced in a manner similar to that obtained when the beam is rotated. A thermocouple is attached to the specimen at the section of greatest stress and connected to a galvanometer. The specimen is run at 1000 r.p.m. for 30 sec. at stresses which increase by stages; a reading of the galvanometer is taken for each value of the stress. The stress at which there is a sudden increase in the reading of the galvanometer is taken as the endurance limit. For ferrous metals the results varied from 14% below those found by the standard method to 18% above, but in the main they were within 5 or 6% of the standard values. The test does not give consistent results with non-ferrous metals.

**Electrical Resistance Method.**—This sometimes is known as the Ikeda method, after the Japanese who developed it. (See Moore and Konzo, *Bull.* 205, Engg. Exp. Sta., Univ. of Ill., 1930.) It consists in determining the electrical resistance corresponding to various stresses and noting a sudden change in the resistance. The stress where the change occurs in taken as the endurance limit. The method gives results comparable with those obtained by the rise-in-temperature method and gives consistent results with non-ferrous metals. It requires delicate instruments and careful manipulation.

**CREEP LIMIT TESTS.**—The following description of the apparatus for and the method of making creep tests is abstracted from Norton, The Creep of Metals at High Temperatures (McGraw-Hill, New York). Specimens 0.505 in. diam. with a 4-in. gage length are placed in an electric furnace. The load is applied by weights and a lever having a ratio of 10 to 1. The furnace, which holds 5 or 6 specimens at one time, is a hollow cylinder wrapped with asbestos paper and a heating coil which maintains a constant temperature through a regulating device. The extension of each specimen is measured by means of a dial connected to the lever arm which applies the load. To prevent oxidation of the specimen, a stream of nitrogen is introduced at the bottom of the apparatus.

The specimens are loaded nearly to the estimated creep limit for a given temperature, and this load is maintained for 400 hours, the elongation being measured twice a day. The load then is increased 10% and maintained for 400 hours, the elongation again being measured twice a day. This rate of increase in load is continued until the creep becomes large. Thus for each temperature, there will be a series of stress-elongation readings. A graph is plotted for each stress, with time as abscissa and percentage of elongation as ordinate. This usually is a straight line. The slopes of these time-percentage elongation lines are next plotted against stress in logarithmic paper. There will be one line for each temperature and, since the points fall on a straight line, the stress corresponding to any given rate of creep may be read directly from the graph. Rapid progress is being made (1937), in the development and revision of standard methods of creep limit tests. The latest issue of the A.S.T.M. standards should be consulted.

**X-RAY EXAMINATION OF CASTINGS AND FORGINGS.**—When x-rays are passed through castings or forgings which contain defects, the x-rays pass through the defects more rapidly than through the sound metal and thus outline the defects on a photographic plate.

The x-ray examination of castings or forgings has two functions: 1. Development of manufacturing technique; 2. Inspection of the finished product. In Recommended Practice for Radiographic Testing of Metal Castings (A.S.T.M. Standards, E15–29) the following defects are enumerated as being capable of detection by x-ray methods: Gas cavities, slag inclusions, sand inclusions, pipe cavities, porosity, cracks, metal segregation. Since there are no standards governing the number, extent and character of defects, no specific requirement can be laid down, but if the defects are known to be present the user can judge of their probable effect and act accordingly.

**Equipment.**—Equipment needed for x-ray examination consists of an x-ray tube, transformers, photographic apparatus, and a lead-lined operating chamber. The part to be examined is placed between the x-ray tube and the photographic film and an exposure made for the proper length of time. Since x-rays are very dangerous it is imperative that the operator be properly protected from them by lead-lined boxes or chambers. The operation is one requiring expensive apparatus and skilled technique, but it is very satisfactory for castings in which a positive knowledge of their soundness is essential. For metal not over 3 $^1/_2$ in. thick, a 200-kv. 8-milliampere Coolidge x-ray tube will be satisfactory. The Babcock & Wilcox Co. installed in 1933 a 300-kv., 10 milliampere tube for examining welded joints in pressure vessels (see Welding, p. 2–48). For detailed information see George L. Clark, Applied x-Rays (McGraw-Hill, New York); *Metals and Alloys;* St. John and Isenburger, Industrial Radiography (John Wiley & Sons, New York).

**SPECIAL TESTS** of materials, structures, and machines often are made to determine some particular characteristic. Typical tests of this character are given in Table 2. Other special tests are discussed below.

**Determination of Physical Properties from Magnetic Permeability.**—Tests of magnetic permeability of steel and iron seem to give some indication of the hardness, strength, and uniformity. They possess the great advantage that the test does not destroy the sample tested, and hence may be made on the actual parts to be used. Magnetic tests as an indication of hardness or strength are not standardized (1937), but hold promise of future usefulness.

The variation of magnetic qualities in a bar or disc of metal has been used to locate small invisible flaws. (See J. A. Capp, *Proc.* A.S.T.M., vol. xxvii, Pt. II, p. 268, 1927.) This principle has been utilized in the construction of a device, known as the transverse fissure detector car, for locating transverse fissures in railroad rails.

**Tests of Strength of Wood** have not been standardized (1937) to an extent which allows their inclusion in specifications for timber, but compression tests, flexure tests, shearing tests, and impact tests frequently are made. Compression tests are made both

**Table 2.—Special Tests of Materials, Structures, and Machines**

| Structural part, machine part, structure, machine or material tested | Test | Measurements |
|---|---|---|
| Floor panel of building. . | Proof test with dead load, not to destruction. | Deflections, tensile and compressive deformations in beams and columns. |
| Bridge. . . . . . . . . . . . . . . | Proof test with dead load or with moving load. | Deflections, tensile and compressive deformations in various members. |
| Riveted joints. . . . . . . . . | Tests of samples to destruction. | Ultimate load, slip of rivets. |
| Rivets, metal plates. . . . | Shearing test of samples to destruction, using special shearing tools. | Ultimate load, results depend on hardness of shearing tools used. |
| Bolts. . . . . . . . . . . . . . . . | Tests of samples to destruction in tension or torsion. | Ultimate load. |
| Boilers. . . . . . . . . . . . . . . | Tests with hydrostatic pressure. Proof tests with pressures somewhat above working pressures. | Observation of leaks, cracks, or permenent distortion of parts. |
| Car couplers and coupler yokes | Proof tests, tests of samples to destruction under tension and impact tension. | Ultimate load, distortion under proof load. |
| Wire rope. . . . . . . . . . . . . | Tests of samples to destruction in tension. | Ultimate load, observation of manner of fracture. |
| Brake beams for railway cars. | Tests of samples to destruction in flexure. | Ultimate load, deflections. |
| Chain. . . . . . . . . . . . . . . . | Proof tests of entire chain in tension, tests to destruction of sample sections. | Set after removal of proof load, ultimate strength. |
| Large pipe. . . . . . . . . . . . | Hydrostatic pressure proof test. | Observation of leaks, cracks, or other evidence of failure. |
| Gear teeth. . . . . . . . . . . . | Scleroscope or Rockwell test for uniform hardness. | Hardness for various teeth. |
| Engraver's plates. . . . . . . | Brinell, Rockwell, or scleroscope tests for uniform hardness of samples. | Hardness at various points. |
| Strain insulators for electric transmission lines. | Proof load in tension, tension test of samples to destruction. | Evidences of failure under proof load, ultimate strength. |
| Eyebars. . . . . . . . . . . . . . . | Tension test of sample eyebars to destruction. | Yield point, ultimate load, stretch. |
| Columns. . . . . . . . . . . . . . | Compression tests of models or samples to destruction. | Yield point, ultimate load, deflection, axial compression. |

along the grain and across the grain. Compression specimens usually are rectangular blocks, and should be tested with spherical-seated bearing blocks. Wood has a rather poorly defined proportional limit in compression. In compression along the grain there is a well-defined ultimate strength. Shearing tests along the grain are made by the use of shearing blocks placed in the testing machine, or for larger specimens by testing short, deep beams. The shearing strength of wood along the grain is of great importance. Flexure tests both of small selected specimens and of large beams are common. In flexure tests the failure may be by longitudinal shear along the neutral axis, by compression along the upper side of the beam, or by tension along the lower side. Impact flexure tests show the shock-resisting qualities of wood. Hardness tests are made by using a steel ball 0.444 in. in diameter, and noting the load which causes the ball to penetrate to one-half its diameter.

**Impact Testing of Wood.**—The Turner-Hatt impact testing machine is in common use in the U. S. for impact testing of wood. It consists of an anvil on which the specimen is placed, and of a weight which can be dropped from various heights. Attached to the weight is a pencil which draws a record on a rotating drum. From this record the deflections of the specimen can be measured, as the weight is dropped from successively increasing heights until rupture occurs, or until the deflections increase abnormally, showing that the proportional limit has been passed. Another method of making an impact test consists in dropping the weight from such a height that the specimen is fractured by one blow. The pencil attached to the weight traces a curve on the rotating drum whose steepness is a measure of the velocity of the falling weight. By measuring the velocity of the weight before and after fracture of the specimen, the energy absorbed in fracturing the specimen can be determined. For detailed description of tests of timber, see the A.S.T.M. Standards, D143–27 and D198–27.

**DUCTILITY TESTS.—Ductility as Determined from Tension Test.**—The usual measures of ductility are the percentage of elongation and the percentage of contraction of area of a specimen in a tension test. Since the percentage elongation varies widely over the length of a tension specimen, the gage length over which the deformation is measured should be specified. A tension test for determining ductility is not suitable for many metals, for example, a thin metal specimen, which in tension usually fails without any appreciable elongation.

**Bending Test for Determining Ductility.**—A.S.T.M. standard specification A9–36, for bend tests on structural steel requires that bend test specimens shall stand being bent cold through 180° without cracking on the outside of the bent portion, around a pin, the diameter of which shall have the following relation to the thickness of the specimen:

| Thickness of Material | Ratio: $\frac{\text{Pin Diameter}}{\text{Thickness of Specimen}}$ |
|---|---|
| Up to 3/4 in., inclusive | 1 |
| Over 3/4 to 1 in., inclusive | 1 1/2 |
| Over 1 to 1 1/2 in., inclusive | 2 |
| Over 1 1/2 to 2 in., inclusive | 2 1/2 |
| Over 2 in. | 3 |

**Cupping Tests for Determining Ductility.**—For determining the relative ductility of thin sheet metal, the so-called cupping test often is used. Various machines are on the market for performing this test. In these, a spherical-ended plunger or ball is forced into the test specimen, which is held over a suitable die. For some machines, the maximum depth of the cup which can be formed is taken as a measure of ductility; for others, the measure of ductility is the load required to produce the cup, just as visible " necking down " occurs.

## Section 8

# MECHANISM AND MECHANICS

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# MECHANISM AND MECHANICS

## ELEMENTS OF MACHINES

### 1. ELEMENTARY MACHINES

The object of a machine usually is to transform the work or mechanical energy, exerted at the point where the machine receives its motion, into work at the point where the final resistance is overcome. The specific result may be to change the character or direction of motion, as from circular to rectilinear, or *vice versa*, to change the velocity, or to overcome a great resistance by the application of a moderate force. In all cases the total energy exerted equals the total work done, the latter including the overcoming of all the frictional resistances of the machine as well as the useful work performed. No increase of power can be obtained from any machine, since this is impossible according to the law of conservation of energy. In a frictionless machine the product of the force exerted at the driving-point into the velocity of the driving-point, or the distance it moves in a given interval of time, equals the product of the resistance into the distance through which the resistance is overcome in the same time.

**ELEMENTARY MACHINES.**—The most simple machines, or elementary machines, are reducible to three classes, viz., the Lever, the Cord, and the Inclined Plane. The first class includes every machine consisting of a solid body capable of revolving on an axis, as the Wheel and Axle, or Crank. The second class includes every machine in which force is transmitted by means of flexible threads, ropes, belts, etc., as the Pulley.

The third class includes every machine in which a hard surface inclined to the direction of motion is introduced, as the Wedge, the Screw, the Cam, etc.

**THE EFFICIENCY OF A MACHINE** is a fraction expressing the ratio of the useful work to the whole work performed, which is equal to the energy expended. The limit to the efficiency of a machine is unity, denoting the efficiency of a perfect machine in which no work is lost. The difference between the energy expended and the useful work done, or the loss, usually is expended either in overcoming friction or in doing work on bodies surrounding the machine from which no useful work is received. Thus in an engine propelling a vessel, part of the energy exerted in the cylinder does the useful work of giving motion to the vessel, and the remainder is spent in overcoming the friction of the machinery and in making currents and eddies in the surrounding water.

**A LEVER** is an inflexible rod capable of motion about a fixed point, called a fulcrum. The rod may be straight, bent at any angle, or curved. It generally is regarded, at first, as without weight, but its weight may be considered as another force applied in a vertical direction at its center of gravity.

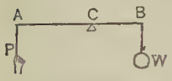


Fig. 1. 1st class

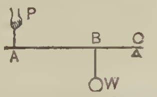


Fig. 2. 2nd class

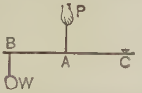


Fig. 3. 3rd class

Classes of Levers

The arms of a lever are the portions of it intercepted between the force, $P$, and fulcrum, $C$, and between the weight or load, $W$, and fulcrum. Levers are divided into three classes, according to the relative positions of the applied force, load, and fulcrum. In a lever of the first class, Fig. 1, the fulcrum lies between the points at which the force and the load act. In a lever of the second class, Fig. 2, the load acts at a point between the fulcrum and the point of application of the force. In a lever of the third class, Fig. 3, the point of application of the force is between the point of load and the fulcrum.

In all class of levers the relation between the force exerted, or the pull, $P$, and the load lifted, or resistance overcome, $W$, is expressed by the equation

$$P \times AC = W \times BC \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [1]$$

where $AC$ is the lever-arm of $P$, and $BC$ is the lever-arm of $W$; or moment of the force = moment of the resistance.

In cases in which the direction of the force (or of the resistance) is not at right angles to the arm of the lever on which it acts, the "lever-arm" is the length of a perpendicular from the fulcrum to the line of direction of the force (or of the resistance). If $\alpha$ be the angle which the direction of the applied force $P$ makes with the lever arm, the effective force $P' = P \sin \alpha$ and

$$P' \times AC = P \sin \alpha AC = W \times BC \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

Similarly the load $W$, acting at an angle $\beta$ to the lever arm would result in the equation

$$P \times AC = W \sin \beta \times BC \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

or

$$P \sin \alpha \times AC = W \sin \beta \times BC \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

The ratio of the resistance to the applied force is the inverse ratio of their lever arms or $W : P :: AC : BC$. Also if $V_w$ is the velocity of $W$, and $V_p$ is the velocity of $P$, then $W : P :: V_p : V_w$ and

$$P \times V_p = W \times V_w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

If $S_p$ is the distance through which the applied force acts, and $S_w$ is the distance through which the load is lifted, or through which the resistance is overcome, $W : P :: S_p : S_w$, and

$$W \times S_w = P \times S_p \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

These equations are general for all classes of machines as well as for levers, it being understood that friction, which in actual machines increases the resistance, is not at present considered.

**The Bent Lever.**—In the bent lever, Fig. 4, the lever-arm of the weight $m$ is $cf$ instead of $bf$. The lever is in equilibrium when $n \times af = m \times cf$, but it is to be observed that the action of a bent lever may be very different from that of a straight lever. In the latter, so long as the force and the resistance act in lines parallel to each other, the ratio of the lever-arms remains constant, although the lever itself changes its inclination with the horizontal. In the bent lever, however, this ratio changes: thus, in the illustration, if the arm $bf$ is depressed to a horizontal position, the distance $cf$ lengthens while the horizontal projection of $af$ shortens, the latter becoming zero when the position of $af$ becomes vertical. As the arm $af$ approaches the vertical, the weight $n$ which may be lifted with a given force $s$ is very great, but the distance through which it may be lifted is very small. In all cases the ratio of the weight $m$ to the weight $n$ is the inverse ratio of the horizontal projection of their respective lever-arms.

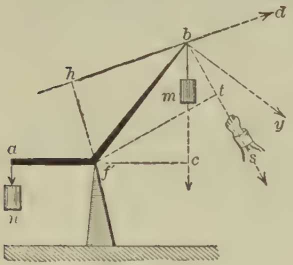


Fig. 4. Bent Lever

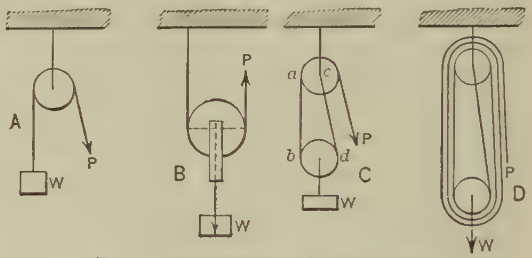


Fig. 5. Arrangements of Pulley Blocks

**THE CORD** is a flexible connector, used to change the direction of an applied force, usually by passing it over a pulley or wheel, or to transmit power from one source to another as in belt drives. For the application of the cord to power transmission, see pp. 24–14 to 24–52.

**Pulleys or Blocks.**—Let $P$ = force applied, or pull; $W$ = load lifted, or resistance. In the simple pulley $A$ (Fig. 5$A$), the point $P$ on the pulling rope descends the same amount that the load is lifted, therefore $P = W$. In Fig. 5$B$ and 5$C$ the point $P$ moves twice as far as the load is lifted, therefore, $W = 2P$. In Figs. 5$B$ and 5$C$ there is one movable block, and 2 plies of the rope engage with it. In Fig. 5$D$ there are 3 sheaves in the movable block, each with 2 plies engaged, or 6 in all. Six plies of the rope, therefore, are shortened by the same amount that the load is lifted and the point $P$ moves 6 times as far as the load; consequently $W = 6P$. In general, the ratio of $W$ to $P$ is equal to the number of plies of the rope that are shortened, and also is equal to the number of plies that engage the lower block. If the lower block has 2 sheaves and the upper 3, the end of the rope is fastened to a hook in the top of the lower block, then there are 5 plies shortened instead of 6, and $W = 5P$. If $V$ = velocity of $W$, and $v$ = velocity of $P$, then in all cases $VW = vP$, whatever the number of sheaves or their arrangement. If the hauling rope, at the pulling end, passes first around a sheave in the upper or stationary block, the direction in which the rope is led from this block to the point at which the pull on the rope is applied is immaterial; but if it first passes around the movable block, it is necessary that the pull be exerted in a direction parallel to the line of action of the resistance, or a line joining the centers of the two blocks, in order to obtain the maximum effect. If the rope pulls on the lower block at an angle, the block will be pulled out of the line drawn between the load and the upper block, and the effective pull will be less than the actual pull on the rope in the ratio of the cosine of the angle the pulling rope makes with the vertical, or line of action of the resistance, to unity.

**Differential Pulley.** (Fig. 6.)—Two pulleys, *B* and *C*, of different radii, rotate as one piece about a fixed axis, *A*. An endless chain, *BDECLKH*, passes over both pulleys. The rims of the pulleys are shaped so as to hold the chain and prevent it from slipping. One of the bights or loops *DE*, in which the chain hangs, passes under and supports the running block *F*. The other loop or bight, *HKL*, hangs freely, and is called the hauling part. It is evident that the velocity of the hauling part is equal to that of the pitch-circle of the pulley *B*.

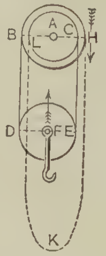


Fig. 6. Differential Pulley

In order that the velocity-ratio may be exactly uniform, the radius of the sheave *F* should be an exact mean between the radii of *B* and *C*.

Consider that the point *B* of the cord *BD* moves through an arc whose length = *AB*, during the same time the point *C* or the cord *CE* will move downward a distance = *AC*. The length of the bight or loop *BDEC* will be shortened by *AB* − *AC*, which will cause the pulley *F* to be raised half of this amount. If *P* = the pulling force on the cord *HK*, and *W* the weight lifted at *F*, then

$$P \times AB = W \times {}^1/_2(AB - AC) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

The length of chain required for a differential pulley is: Half the circumference of *A* + half the circumference of *B* + half the circumference of *F* + twice the greatest distance of *F* from *A* + the least length of loop *HKL*. The last quantity is fixed according to convenience.

**A Wheel and Axle, or Windlass,** resembles two pulleys on one axis, having different diameters. If a weight be lifted by means of a rope wound over the axle, the force being applied at the rim of the wheel, the action is like that of a lever of which the shorter arm is equal to the radius of the axle plus half the thickness of the rope, and the longer arm is equal to the radius of the wheel. A wheel and axle is, therefore, sometimes classed as a perpetual lever. If $P$ = the applied force, $D$ = diameter of the wheel, $W$ = the weight lifted, and $d$ = diameter of the axle + the diameter of the rope, $PD = Wd$.

**The Differential Windlass** (Fig. 7) is identical in principle with the differential pulley, the difference in construction being that in the differential windlass the running block hangs in the bight of a rope whose two parts are wound round, and have their ends respectively made fast to, two barrels of different radii, which rotate as one piece about the axis *A*. The differential windlass is little used in practice, because of the great length of rope which it requires.

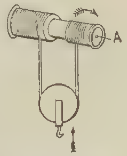


Fig. 7. Differential Windlass

**THE INCLINED PLANE,** as a mechanical element, is supposed to be perfectly hard and smooth, unless friction be considered. It assists in sustaining a heavy body by its reaction. This reaction, however, being normal to the plane, cannot entirely counteract the weight of the body, which acts vertically downward. Some other force, therefore, must be made to act upon the body, in order that it may be sustained.

If, in Fig. 8, the sustaining force *P* acts parallel to the plane, the ratio of the force to the weight *W* is equal to the ratio of the height of the plane to its length, measured on the plane. If the force acts parallel to the base, the ratio of the force to the weight is equal to the ratio of the height to the base.

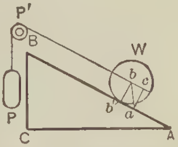


Fig. 8. The Inclined Plane

For forces acting at any other angle, let $\theta$ = angle of plane with the horizon and $\phi$ = angle of direction of applied force with the angle of the plane. Then $P : W :: \sin\theta : \cos\phi$, and

$$P \times \cos\phi = W \times \sin\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

If the force acts at any other angle, let $i$ = the angle of the plane with the horizon, and $e$ = the angle of the direction of the applied force with the angle of the plane. $P : W :: \sin i : \cos e$; $P \times \cos e = W \sin i$.

Problems of the inclined plane may be solved by the parallelogram of forces as follows:

Let the weight $W$ be kept at rest on the incline by the force $P$, acting in the line $bP'$, parallel to the plane. Draw the vertical line $ba$ to represent the weight; also $bb'$ perpendicular to the plane, and complete the parallelogram $b'c$. Then the vertical weight $ba$ is the resultant of $bb'$, the measure of support given by the plane to the weight, and $bc$, the force of gravity tending to draw the weight down the plane. The force required to maintain the weight in equilibrium is represented by this force $bc$. Thus the force and the weight are in the ratio of $bc$ to $ba$. Since the triangle of forces $abc$ is similar to the triangle of the incline $ABC$, the latter may be substituted for the former in determining the relative magnitude of the forces, and $P : W :: bc : ab :: BC : AB$.

**The Wedge** is a pair of inclined planes united by their bases. In the application of pressure to the head or butt end of the wedge, to cause it to penetrate a resisting body, the ratio of applied force to the resistance is equal to the ratio of the thickness of the wedge to its length. Let $t$ = thickness and $l$ = length of the wedge, $W$ = resistance, and $P$ = applied force or pressure on the head of the wedge. Then, friction neglected, $P : W :: t:l$, and

$$W = Pl/t \qquad [9]$$

## 2. APPLICATIONS OF THE LEVER

**THE MOVING STRUT** (Fig. 9) is similar to the bent lever, except that one of the arms is missing, and that the force and the resistance to be overcome act at the same end of the single arm. The resistance in the case shown is not the load $W$, but its resistance to being moved, $R$, which may be simply that due to its friction on the horizontal plane, or some other opposing force. When the angle between the strut and the horizontal plane changes, the ratio of the resistance to the applied force changes. When the angle becomes very small, a moderate force will overcome a very great resistance, which tends to become infinite as the angle approaches zero. If $\alpha$ = the angle that the strut makes with the plane on which $W$ rests,

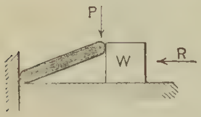


**Fig. 9.** Moving Strut

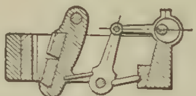


**Fig. 10.** Moving Strut Applied to Stone Crusher

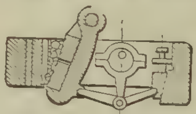


**Fig. 11.** Toggle Joint Applied to Stone Crusher

$$P \times \cos \alpha = R \times \sin \alpha \qquad [10]$$

If $\alpha$ = 5 deg., $\sin \alpha = 0.08716$, $\cos \alpha = 0.99619$ and $R = 11.44\ P$.

The stone crusher, Fig. 10, is a practical example of the use of two moving struts.

**THE TOGGLE-JOINT** is an elbow or knee joint, comprising two links or struts so connected that they may be brought into a straight line when a force is applied at the joint to move them into this position, thus producing great endwise pressure. The toggle-joint is a case of two moving struts placed end to end, the moving force being applied at their junction in a direction at right angles to the direction of the resistance. One end of one strut rests against a fixed abutment and the opposite end of the other strut rests against the body to be moved. In Fig. 12, if $\alpha$ = angle each strut makes with the straight line joining the points about which their outer ends rotate, the ratio of the resistance to the applied force is $R : P :: \cos \alpha : 2 \sin \alpha$, and

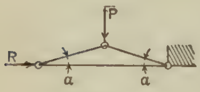


Fig. 12

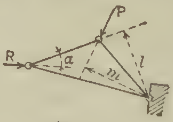


Fig. 13

$$2R \sin \alpha = P \cos \alpha \qquad [11]$$

The ratio varies as the angle varies. becoming infinite when the angle becomes zero.

If the direction of motion of the resisting body is not in the same straight line with the line joining the outer ends of the two struts, and if the applied force does not act at right angles to the line joining the outer ends of the struts, as shown in Fig. 13, the ratio of the applied force to the resistance is

$$P = R \times m \cos \alpha / l \qquad [12]$$

where $\alpha$ is the angle between the direction of motion of the resistance and the center line of the strut immediately adjacent to the resistance, $m$ is the perpendicular distance from the axis of rotation at the abutment to the line of action of the applied force, and $l$ is the perpendicular distance from the axis of rotation at the abutment to the center line of the strut adjacent to the resistance.

The toggle-joint is used where great resistances are to be overcome through very small distances, as in stone-crushers (Fig. 11).

**LINKWORK.**—Two levers, in the form of cranks, connected to two rigid members, one of which supports the crank bearings, constitute a four-bar linkage, generally known as linkwork. In general, any mechanism can be analyzed as a four-bar linkage or a combination of such linkages. Fig. 14 is a diagrammatic representation of a four-bar linkage, in which $A$ and $B$ are the cranks, and $C$ and $D$ the rigid members. Either $C$ or $D$ is fixed. Motion imparted to $A$ or $B$ then will cause definite determinate motion of the other movable members, the character of which depends on the position and length of the various links. The angular speed ratios of the cranks are inversely proportional to the lengths of the perpendiculars from the fixed centers of the cranks to the center line of the connecting-rod ($bn$ and $am$, Fig. 14). Various general combinations of the four-bar linkage are in common use, which are described below.

**Crank and Rocker** (Fig. 15).—In this combination the crank $A$ revolves in a complete circle, imparting an oscillating motion to the rocker arm $B$ through the connecting rod $C$. The link $D$ is fixed by being rigidly attached to the structure of which the linkage is a part. The following relations must hold to permit this linkage to operate:

$$1.\ A + B + C > D; \qquad 2.\ A + D + B > C$$

$$3.\ A + C - B < D; \qquad 4.\ C - A + B > D$$

Angle of oscillation $\alpha$ of the rocker arm $B = \cos^{-1} abd - \cos^{-1} abd$. Either $A$ or $B$ may be the driving crank. If $A$ drives, the mechanism always will operate. If $B$ drives, it will not operate unless a fly-wheel or other aid carries the crank over the dead-center points. An example of the application of this combination is the walking-beam of paddle-wheel steamers.

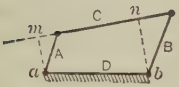


FIG. 14. Four-bar Linkage

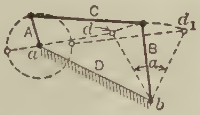


FIG. 15. Crank and Rocker

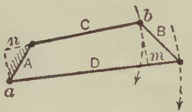


FIG. 16. Drag Link

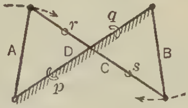


FIG. 17. Non-parallel Equal-crank 4-bar Linkage

**Drag Link** (Fig. 16).—The link $A$ is fixed, $D$ and $C$ are the cranks, and $B$ is the connecting-rod. To permit complete revolution of the cranks the following relations must hold: 1. $B > A + D - C$; 2. $B < D + B - A$. Examples of application are the feathering paddles of paddle-wheel steamers and the drives of slotters.

**Parallel Crank Four-bar Linkage.**—If in Fig. 14 the cranks $A$ and $B$ are of equal length and the connecting-rod $C$ is equal in length to the line of centers of the cranks, link $D$, the two cranks always will turn at the same angular speed. An example of the application of this linkage is the side rod and crank on the driving wheels of locomotives.

**Non-parallel Equal-crank Linkage.**—If the cranks are made of equal length, but pointing in opposite directions from their centers, as in Fig. 17, motion imparted to one crank will cause the other to rotate in the opposite direction. If crank $A$ is given a uniform angular motion, crank $B$ will have a varying angular speed. It is necessary, in order to obtain rotation in opposite directions, to cause the cranks to pass the dead center points in opposite directions by means of some device placed at the instantaneous axis of the connecting-rod when they are at the dead-center position. Such a device may be pins, $r$ and $s$, on the connecting-rod $C$, with open eyes at the corresponding points $p$ and $q$ on the fixed member $D$, with which the pins will mesh. Linkages of this type may be replaced by gears, Fig. 18, which are cut on the circumference of an ellipse whose foci are at $a$, $b$, and $c$, $d$, respectively, and whose major axes are $ad = bc$. The intersection of links $D$ and $C$ at all positions of the linkage is a point on the circumference of the two ellipses, and, therefore, is their point of tangency. An example of the application of such linkwork or elliptical gearing is the slow-advance, quick-return drive of slotters.

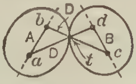


FIG. 18. Elliptical Gearing

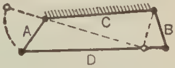


FIG. 19. Slow Motion Linkwork

**Slow Motion Linkwork.**—Fig. 19 shows a four-bar linkage arranged to produce slow motion in the crank $B$. As the crank $A$ is rotated upward it imparts motion to crank $B$, whose angular velocity decreases to zero when the crank $A$ reaches its dead-center position with the connecting-rod $C$ in line with the crank. An example of the application of this linkage is the valve motion of a Corliss engine.

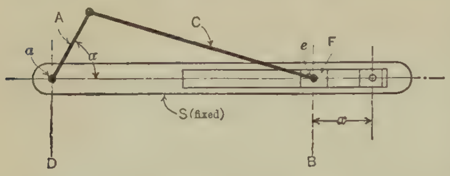


FIG. 20. Linkwork with Sliding Pair

**Linkwork with a Sliding Pair.**—If in Fig. 14, link $D$ be considered as two parallel lines through points $a$ and $b$, perpendicular to $D$ and meeting at infinity, and that on $D$ is mounted a slide in which moves a block attached to the end $e$ of $C$, we have the combination shown in Fig. 20 which is the common form of crosshead and connecting-rod in steam engines. It should be borne in mind that the slide $S$ is not one of the links, but that the slide and the block pivoted to the link $C$ are the equivalent of the infinite links represented by the parallel lines through $a$ and $b$. Two general classes of this combination exist: 1. One member of the sliding pair fixed. 2. One member of the sliding pair turning about a fixed axis. In group 1, the connecting-rod $C$ can be longer, shorter, or equal to the length of the crank $A$. In group 2 the crank can be longer, shorter or equal to the line of centers ($C$, Fig. 22). When one member of the sliding pair is fixed and the connecting-rod is longer than the crank, the movement of the block $F$ from its forward dead-center position is

$$x = A\,(1 - \cos\alpha) + B\{1 - \sqrt{1 - (A^2/B^2)\sin^2\alpha}\} \quad . \quad . \quad . \quad . \quad [13]$$

Another form of this arrangement sometimes used on crank-and-flywheel pumps is shown in Fig. 21. It is known as the Scotch yoke. The slide is at right angles to the center line of the piston rod and the follower block is pivoted to the rotating crank.

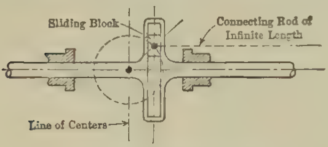


FIG. 21. Scotch Yoke

When the sliding follower is used to drive the crank, means must be provided to carry the crank over the dead-center positions, as by a fly-wheel or by two such linkages, the cranks of which are keyed to the same shaft and set at right angles to each other. An example is the driving cranks of locomotives. Another arrangement is to set the lines of movement of the follower block at an angle of approximately 90 deg. to each other and attach the connecting-rods to the same crank-pin on the crank. Examples of this arrangement are steering engines and some types of hoisting engines.

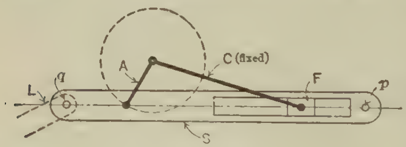


FIG. 22

If the follower $F$ in Fig. 20 is fixed to the frame and can neither turn nor slide, the combination then becomes a case where the line of centers is shorter than the crank. The connecting-rod $C$ then acts as a crank oscillating about the center $e$, the crank $A$ becomes the connecting-rod. If $A$ be given a complete rotation about its axis $a$, the axis will move along the line $ae$ and carry the slide $S$ along with it.

Fig. 22 illustrates a case of the second class noted above wherein the link $C$ is longer than the line of centers $B$. The pin in block $F$ in which link $C$ oscillates is fixed and the follower $F$ rotates around it. As the crank $A$ is rotated the slide $S$ oscillates about the pin, and at the same time slides back and forth over the follower $F$. This mechanism is used in a modified form for a quick-return motion in shapers and other machine tools. The slide $S$ swings on a pin fixed at $P$ and a link $L$, oscillating on the pin $q$, connects the slide $S$ with the head of the machine.

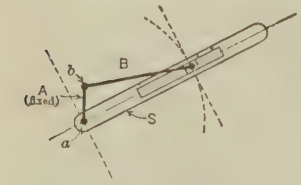


FIG. 23

**Whitworth Quick-return Motion** is a case of Class 2, noted above, where the line of centers is shorter than the crank. In Fig. 23, if the link $A$ is fixed, the link $B$ becomes a crank turning about the center $d$ and causing the slide $S$ to rotate about the axis $a$. The ratio of the angular speeds of the two rotating members varies, since the perpendiculars from them to the center line of the theoretical connecting-rod varies. In the Whitworth quick-return motion, Fig. 24, with the crank $B$ at position 1 the follower $F$ is at the extreme position at one end of the stroke. As the crank is rotated to position 2 through the angle $\alpha$, the follower is carried to its extreme position at the opposite end of the stroke. As the crank further rotates from position 2 to position 1 through the angle $\beta$ the slide is returned to its original position at the other end of the stroke. The ratio of the speed of the forward slow stroke and of the quick return stroke is the ratio $\alpha/\beta$.

For a complete discussion of linkwork, see Schwamb, Merrill and James, Elements of Mechanism (John Wiley & Sons).

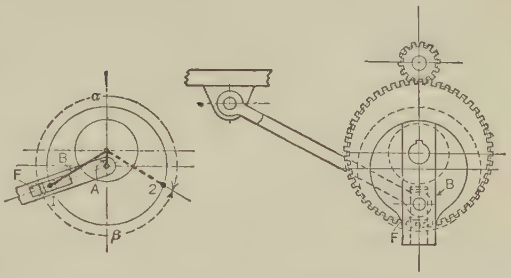


FIG. 24. Whitworth Quick-return Motion

## 3. APPLICATIONS OF THE INCLINED PLANE

**THE SCREW** is an inclined plane wrapped around a cylinder in such a way that the height of the plane is parallel to the axis of the cylinder. If the screw is formed on the internal surface of a hollow cylinder, it is called a nut. When force is applied to raise a weight or overcome a resistance by means of a screw and nut, either the screw or the nut may be fixed, the other being movable. The force generally is applied at the end of a wrench or lever-arm, or at the circumference of a wheel. If $r$ = radius of the wheel or lever-arm, and $p$ = pitch of the screw, or distance between threads, that is, the height of the inclined plane for one revolution of the screw, $P$ = the applied force, and $W$ = the resistance overcome, then, neglecting resistance due to friction,

$$2\pi r \times P = Wp \qquad [14]$$

$$W = 6.283\, Pr \div p \qquad [15]$$

The ratio of $P$ to $W$ is thus independent of the diameter of the screw. In actual screws, much of the power transmitted is lost through friction.

**Efficiency of a Screw.**—Let $\alpha$ = angle of the thread, that is, the angle whose tangent is the pitch of the screw divided by the circumference of a circle whose diameter is the mean of the diameters at the top and bottom of the thread. Then for a square thread

$$\text{Efficiency} = (1 - f \tan \alpha) \div (1 + f \cot \alpha), \qquad [16]$$

in which $f$ is the coefficient of friction. (For demonstration, see Cotterill and Slade, Applied Mechanics.) Since cotan = 1 ÷ tan, we may substitute for cotan $\alpha$ the reciprocal of the tangent, or if $p$ = pitch, and $c$ = mean circumference of the screw,

$$\text{Efficiency} = (1 - fp/c) \div (1 + fc/p) \qquad [17]$$

EXAMPLE.—Efficiency of square-threaded screws $^1/_2$ inch pitch.

| | | | | |
|---|---|---|---|---|
| Diameter at bottom of thread, in. | 1 | 2 | 3 | 4 |
| Diameter at top of thread, in. | $1\,^1/_2$ | $2\,^1/_2$ | $3\,^1/_2$ | $4\,^1/_2$ |
| Mean circumference of thread, in. | 3.927 | 7.069 | 10.21 | 13.35 |
| Cotangent $\alpha = c \div p$ = | 7.854 | 14.14 | 20.42 | 26.70 |
| Tangent $\alpha = p \div c$ = | 0.1273 | 0.0707 | 0.0490 | 0.0375 |
| Efficiency if $f = 0.10$ = | 55.3% | 41.2% | 32.7% | 27.2% |
| Efficiency if $f = 0.15$ = | 45% | 31.7% | 24.4% | 19.9% |

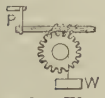


FIG. 25. Worm Gear

The efficiency thus increases with the steepness of the pitch.

The above formulas and examples are for square-threaded screws, and consider the friction of the screw-thread only, and not the friction of the collar or step by which end thrust is resisted, and which further reduces the efficiency. The efficiency is also further reduced by giving an inclination to the side of the thread, as in the V-threaded screw. For discussion of this subject, see paper by Wilfred Lewis, *Jour.* Frank. Inst., 1880; also *Trans.* A.S.M.E., vol. xii, 783, 1891.

**Endless Screw, or Worm-Gear** (Fig. 25) commonly is used to convert motion at high speed into motion at very slow speed. When the handle $P$ describes a complete circumference, the pitch-line of the worm-wheel moves through a distance equal to the pitch of the screw, and the weight $W$ is lifted a distance equal to the pitch of the screw multiplied by the ratio of the diameter of the axle to the diameter of the pitch-circle of the wheel. The ratio of the applied force to the weight lifted, or resistance, is inversely as the velocities, friction not being considered; but the friction in the worm-gear is usually very great, amounting sometimes to 3 or 4 times the useful work done.

If $v$ = the distance through which the force $P$ acts in a given time, say one second, and $V$ = distance the weight $W$ is lifted in the same time, $r$ = radius of the crank or wheel through which $P$ acts, $t$ = pitch of the screw, and also pitch of the teeth on the worm-wheel, $d$ = diameter of the axle, and $D$ = diameter of the pitch-line of the cog-wheel,

$$v = (6.283\,r/t)\,(D/d) \times V \quad [18]$$

$$V = v \times td \div 6.283\,rD \quad [19]$$

$$Pv = WV + \text{friction} \quad [20]$$

**COMPOUND AND DIFFERENTIAL SCREWS.***—**A Compound Screw** consists of a right-hand and a left-hand thread wound on the same or on different cylinders. When wound on the same cylinder (Fig. 27*a*) two nuts are employed. When wound on different cylinders (Fig. 27*b*) one nut is employed. Usually both threads have the same pitch. Each revolution of the screw causes the two nuts to move toward one another a distance equal to the sum of the pitches. Thus two fine threads will give the same relative advance as one coarse thread of double the pitch.

**A Differential Screw** comprises two threads of the same hand, but of different pitches, wound on the same or different cylinders. See Fig. 28*a* and 28*b*. One revolution of the screw causes the nuts, Fig. 28*a*, to move toward each other a distance equal to the difference of the pitches. One revolution of the nut, Fig. 28*b*, causes the screws to approach by the same distance. Thus the use of two coarse threads gives the same relative advance as one fine thread.

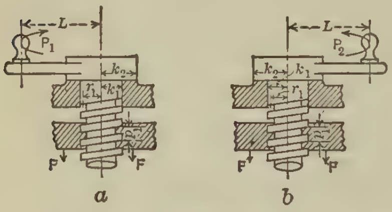


FIG. 26. Simple Screw

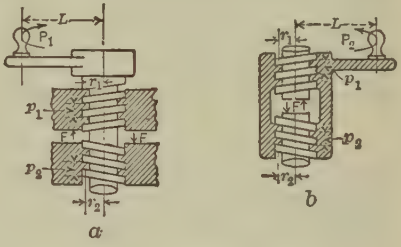


FIG. 27. Compound Screw

**Efficiency.**—The following formulas give the efficiency screws with square threads. In these formulas, $r_1$, $r_2$ = mean radius (to half depth of thread) in.; $c_1$, $c_2$ = mean circumference of screw, in.; $p_1$, $p_2$ = pitch of thread, in.; $p$ = equivalent pitch, in.; $f$ = coefficient of friction; $L$ = length of wrench, in.; $P_1$ = pull on wrench, lb.; $F$ = force transmitted by screw, lb.; $E$ = efficiency. Then

**For Simple Screws**

To raise load (Fig. 26*a*),
$$P_1 = \frac{F}{L}\left\{f\bar{x} + \frac{p_1 + fc_1}{c_1 - fp_1}\,r_1\right\} \quad [21]$$

To lower load (Fig. 26*b*),
$$P_2 = \frac{F}{L}\left\{f\bar{x} + \frac{fc_1 - p_1}{c_1 + fp_1}\,r_1\right\} \quad [22]$$

in which $\bar{x} = {}^2/_3\,\{(k_2^3 - k_1^3)/(k_2^2 - k_1)^2\}$.

If $f = 0$, $\quad P = (F/L)(p_1/6.2832)$, and $E = P/P_1$ [23]

**For Compound Screws** (See Fig. 27),

$$P_1 = \frac{F}{L}\left\{\frac{p_1 + fc_1}{c_1 - fp_1}\,r_1 + \frac{p_2 + fc_2}{c_2 + fp_2}\,r_2\right\} \quad [24]$$

If $f = 0$, $\quad P = (F/L)\{(p_1 + p_2)/6.2832\}$, and $E = P/P_1$ [25]

**For Differential Screws** (See Fig. 28),

$$P_1 = \frac{F}{L}\left\{\frac{p_1 + fc_1}{c_1 - fp_1}\,r_1 + \frac{fc_2 - p_2}{c_2 + fp_2}\,r_2\right\} \quad [26]$$

If $f = 0$, $\quad P = (F/L)\{(p_2 - p_1)/6.2832\}$ and $E = P/P_1$ [27]

For $V$ = threads use $1.15f$ instead of $f$.

---

* Contributed by William H. Boehm.

The load that may be lifted or the force that may be exerted by any of these screws is so much affected by friction that efficiency rarely is as high as 70%. It often is less than 30% for simple screws, and may be less than 5% for differential screws. The coefficient of friction is not uniform, increasing as pitch increases, and being greater with V- than with square threads. Also it varies with the kind of material, the relative smoothness of the contact surfaces and the quality and frequency of lubrication. For screws and nuts of mild steel with square threads, well lubricated, $f$ may be taken as 0.10.

The chief use of the differential screw is to obtain very small movements in precision equipment. Its inefficiency as a means of transmitting power is illustrated by the differential screw punch, below.

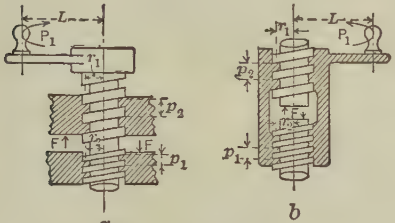


FIG. 28. Differential Screw

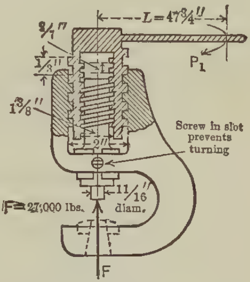


FIG. 29. Differential Screw Punch

**Differential Screw Punch.**—Fig. 29 is a diagram of a differential screw punch in which the pitch of the outer thread is $1/3$ in. and that of the inner thread $2/7$ in., giving an equivalent pitch of $1/21$ in. By jamming the bottom of the outer screw against the top of the inner screw, the device operates as a simple screw punch, with an advance of $1/3$ in. per revolution. The punch was used both as a differential screw punch and a simple screw punch to punch $11/16$ in. holes in $1/4$ in. plate, requiring a force of 27,000 lb. The results with the device used as a differential screw punch were disappointing. The efficiency as computed by formulas [26] and [27], and as actually measured were as follows:

| | Computed | | Actual | |
|---|---|---|---|---|
| | Pull on Lever | Efficiency | Pull on Lever | Efficiency |
| As a differential punch | 92 lb. | 4.6% | 95 lb. | 4.5% |
| As a simple punch | 83 lb. | 36.1% | 82 lb. | 36.7% |

**THE CAM** is a revolving inclined plane. It may be either an inclined plane wrapped around a cylinder in such a manner that the height of the plane is radial to the cylinder, such as the ordinary lifting cam used in stamp mills, Fig. 30, or it may be an inclined plane curved edgewise and rotating in a plane parallel to its base, Fig. 31. The relation of the weight lifted to the applied force is calculated in the same manner as in the case of the screw. See p. 8–09.

**Cam Motions.**—The motion imparted by a cam to its follower usually falls in one of three classes, *viz.*, uniform motion, uniformly accelerated and decelerated motion, and harmonic motion. These three types of motion are illustrated in Fig. 32. In the design of cams the first step is to determine the form of the pitch line, which is done by means of diagrams as illustrated in Fig. 32. Although the diagrams are confined to the three classes of motion noted above, it is obvious that the follower can be caused to conform to any cycle of motion desired. The principles involved in making the diagram for any type of motion are identical with those described below.

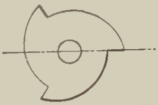


FIG. 30. Lifting Cam

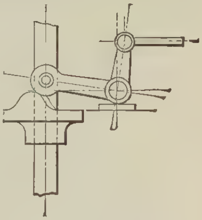
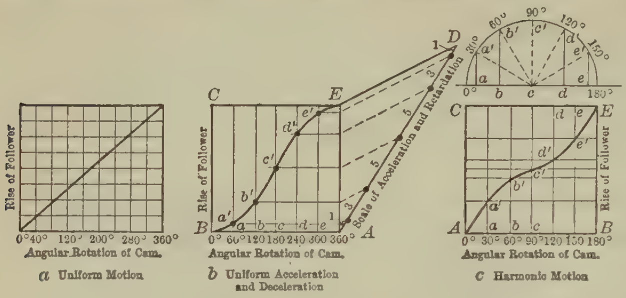


FIG. 31. Edgewise Cam

**Uniform-motion Cams**, as the name implies, produce a motion that is uniform from the beginning to the end of the cycle, each rotation of the cam through an equal number of degrees producing an equal increment of rise of the follower. The diagram of a uniform-motion cam is shown in Fig. 32$a$, in which abscissas represent the developed length of angular degrees of rotation, and ordinates represents increments of rise of the follower. Inasmuch as uniform-motion cams cause the follower to start and stop abruptly, they tend to produce shock in the mechanism, and the pitch line of the cam usually is made with a short curve at the beginning and end to gradually accelerate and retard the follower.

**Cams with Uniform Acceleration and Deceleration.**—The method of developing the pitch line is shown in Fig. 32*b*. Line *AB* represents the developed length of the circumference of the cam. Line *BC*, drawn to the same scale as *AB*, represents the total rise of the follower. A rate of acceleration and retardation first must be assumed, the successive steps increasing and decreasing by equal increments, as 1, 3, 5, 5, 3, 1, or 1, 3, 5, 7, 7, 5, 3, 1, etc. Line *AB* is divided into equal parts representing degrees of angular rotation. Line *AD* is drawn at any convenient angle to *AB*, and is divided in proportion to the scale of acceleration and deceleration adopted. A line is drawn from *D* to *E*, and parallels to *DE* are drawn from the points of division on *DA* to intersect *AE*. From these points of intersection abscissas are drawn parallel to *AB* until they intersect the corresponding ordinates at *a′*, *b′*, *c′*, etc., which are points on the pitch line of the cam. The ordinates *aa′*, *bb′*, *cc′*, etc., represents the rise of the follower at the corresponding degrees of angular rotation.



*a* Uniform Motion

*b* Uniform Acceleration and Deceleration

*c* Harmonic Motion

FIG. 32. Cam Motions

**Harmonic Motion Cams** are those in which the rise of the follower is equal to the sine of the angle through which the cam has rotated. The method of determining the pitch line is shown in Fig. 32*c*. As before, *AB* represents the developed length of the circumference of the cam and *AC* the total rise of the follower, both drawn to the same scale. *AB* is divided into equal parts representing equal numbers of degrees of angular rotation. A semi-circle is drawn, of diameter *AC*, and subdivided by the radii *a′c*, *b′c*, etc., to correspond with the angular degrees of rotation into which *AB* has been divided. Perpendiculars to the diameter from the extremities of these radii represent the rise of the follower. These ordinates are transferred to their corresponding position on the diagram as shown and a smooth curve drawn through them, which is the pitch line of the cam.

**Laying Out a Uniform-motion Cam.**—In Fig. 33 a circle of radius *OC*, equal to the smallest radius of the cam is drawn. The vertical radius is prolonged beyond the circle to *B*. The distance 0–6, equal to the maximum movement of the follower, is laid off on *OB* from its intersection with the circle, and divided into any convenient number of equal parts. The semi-circle to the right of *OB* is divided into the same number of equal arcs, and radii are drawn through the points of division. Then with *O* as a center, and with radii equal to *O*-1, *O*-2, *O*-3, etc., arcs are drawn intersecting the corresponding radii at

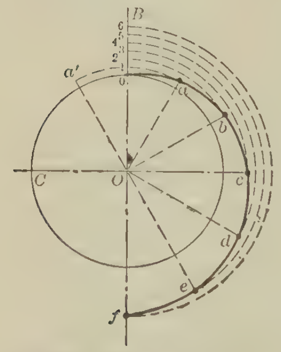


FIG. 33. Uniform-motion Cam

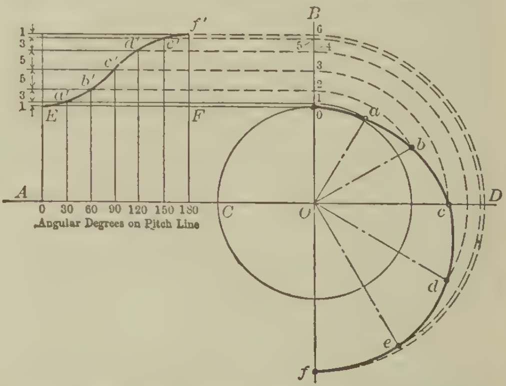


FIG. 34. Cam with Uniform Acceleration and Retardation

*a*, *b*, *c*, etc. The smooth curve drawn through these points is the profile of the cam. The profile of the cam to the left of *OC* is identical, and is drawn in the same manner.

**Laying-out a Cam with Uniform Acceleration and Retardation.**—See Fig. 34. Draw *AD*, and from *O* as a center, with radius *OC*, equal to the smallest radius of the cam, draw a circle. Draw *fOB* perpendicular to *AD*. Draw the tangent *OFE* parallel to *AD*. Divide the semi-circle to the right of *fOB* into any convenient number of equal arcs, and draw radii through the points of division. Lay off on *AC* equal spaces corresponding to the arcs, and erect ordinates at the points of division. Assume a value for the increment or decrement of motion. In the present case it is 2. Hence, with the semi-circle divided into 30-deg. arcs, the relative motion of the follower over the successive arcs will be 1, 3, 5, 5, 3, 1. Note that the motion over the first and last arcs is one-half the value of the assumed increment. Make *f′F* equal to the total movement of the follower. It also is equal to the sum of the increments and decrements of motion, *viz.*, $1 + 3 + 5 + 5 + 3 + 1 = 18$. The length above *EF* of the several ordinates will be

| Angular degrees.................. | 0 | 30 | 60 | 90 | 120 | 150 | 180 |
|---|---|---|---|---|---|---|---|
| Length, fraction of *fF*............ | 0 | $^{1}/_{18}$ | $^{4}/_{18}$ | $^{9}/_{18}$ | $^{14}/_{18}$ | $^{17}/_{18}$ | $^{18}/_{18}$ |

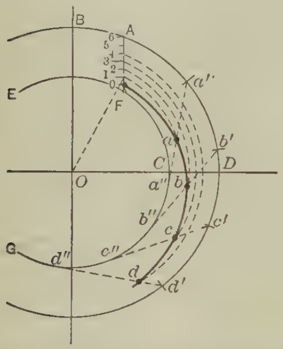


FIG. 35. Cam with Offset Follower

Denote these lengths by points *a′*, *b′*, *c′*, etc., and draw *a′*1, *b′*2, etc. parallel to *EF*, intersecting *OB* at 1, 2, 3, etc. The procedure from this point is the same as for the uniform-motion cam.

**Cam with Offset Follower** (Fig. 35).—Draw the path of the follower *AF* in its position relative to the center *O* of the cam, and divide it according to the character of motion desired. (See Fig. 32.) With *O* as a center and *AO* as a radius describe a circle. Starting at *A*, divide it at *a′*, *b′*, *c′*, etc., to correspond to the positions 1, 2, 3, etc. of the follower. With radius *OC*, equal to the smallest radius of the cam, draw the circle *ECG*. With *O* as a center, with radius equal to *O*1, draw from point 1 on the follower path, the arc 1*a*. From point *a′* draw a tangent *a′a″* to circle *ECG*, intersecting the arc 1*a* at *a*. Similarly draw arcs 2*b*, 3*c*, etc., and tangents *b′b″*, *c′c″*, etc. The intersection of the arcs and tangents at *b*, *c*, etc. are points on the pitch line of the cam.

**Cam with Follower on Rocker Arm** (Fig. 36).—From center *O*, with radius *OC* equal to smallest radius of cam, draw the circle *CG*. Draw the path of the crosshead *JK* in its proper relative position to cam center *O*, and subdivide it according to the type of motion desired. (See Fig. 32.) Locate the center *E* of the rocker arm in its position relative to cam center *O*. Join the point *L*, on the line *AO*, with *E* and prolong *LE* to the opposite end *H* of the rocker arm. With *E* as a center, describe the arcs *HF* and *AL*. Divide arc *HF* to correspond to divisions on the path of the crosshead, by striking arcs from points 0, 1, 2, etc., on *JK* with a radius *JH*. Transfer these points to arc *AL*. With *O* as a center, and *OE* as a radius, draw circle *DEM*, and divide each semi-circle at *a′*, *b′*, *c′*, etc., to correspond with the divisions on *JK*, starting the divisions at *E*, which corresponds to point 0 on *AL*. From *O* as a center, with radii of *O*1, *O*2, *O*3, etc., draw arcs 1*a*, 2*b*, 3*c*, etc. From points *a′*, *b′*, *c′*, etc., with radius equal to *LE*, strike arcs intersecting arcs 1*a*, 2*b*, 3*c*, etc., at *a*, *b*, *c*, etc. The intersections *a*, *b*, *c*, etc., are points on the pitch line of the cam.

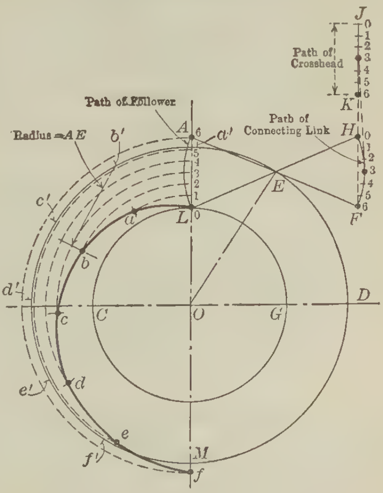


FIG. 36. Cam with Rocker Follower

NOTE.—In the examples of cam layout, Figs. 35 and 36 a uniform-motion cam has been considered. The procedure for any other motion would be identical, except that the points of division of the path of the follower would be determined as illustrated in Fig. 32.

**Correction for Roller Follower.**—A roller follower must be used with cams, unless the force to be transmitted is small. The pitch line of the cam as determined by the above methods then must be corrected. Fig. 37 shows the method of making such corrections. The pitch line as determined above is *ab*. From the center *O* draw radii *Oc*, *Od*, etc. With the points of intersection of the radii with *ab* as centers, draw arcs of radius equal to the radius of the roller follower. The points of intersection of the arcs with the radii are points on the profile of the cam. In Fig. 37 the line *HG* is the profile of a plain cam, and the lines *HG* and *JK* are the boundaries of the slot of a slotted cam.

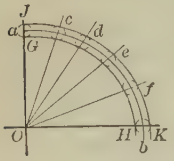


Fig. 37. Profile for Roller Follower

**Cylindrical Cams** comprise a cylinder with a slot cut in the cylindrical surface. The variation in the slope of the slot causes a roller projecting into it to be moved as the cylinder rotates about its axis. Any desired variation in motion can be obtained by appropriate variation of the slot. Fig. 38 shows the method of laying out a cam to give a uniformly accelerated and retarded forward motion, and a uniform return motion in two revolutions of the cam. The line *AB* represents to scale the total movement in one direction of the follower. The line *AA′*, to the same scale, represents the circumference of the cam. The rectangle *AA′B′B* therefore represents the developed surface of the cam. Divide *AB* into the number of equal spaces necessary to represent the desired sequence of acceleration and deceleration increments. Erect ordinates at the division points and locate on them the points of the pitch line. (See Cams with Uniform Acceleration and Retardation, p. 8-13.) Draw the pitch line *AB′* for the forward motion of the follower. The pitch line of the return path is a straight line from *B* to *A′*. Inasmuch as strictly uniform motion would cause shock, the return pitch line is drawn between points slightly short of *B* and *A′*, and joined to *B* and *A′* by arcs. Fig. 39 shows the layout of a cam that gives a harmonic forward motion and a uniform return motion to the follower in a single revolution.

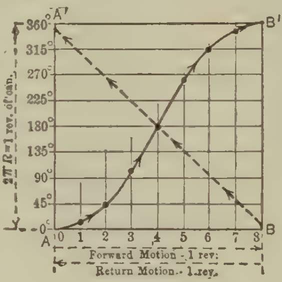


Fig. 38

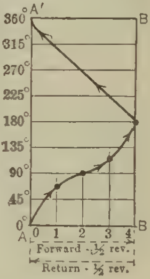


Fig. 39

Cylinder cams also can be designed to give a variety of motions in sequence, during several revolutions. The rectangle representing the developed surface of the cam is divided into half as many parts as the number of revolutions per cycle. The pitch line for that portion of the forward motion of the follower that is to take place during each revolution, then is laid out, by the methods described above, in the portion of the rectangle corresponding to the revolution under consideration. Fig. 40 shows the layout of a cam to give the following cycle of motions in four revolutions: 1. $^3/_4$ revolution, uniform acceleration and deceleration; 2. $^1/_4$ revolution, dwell; 3. $^3/_4$ revolution, uniform acceration and deceleration; 4. $^1/_4$ revolution, dwell; 5. 2 revolutions, return.

**Minimum Radius of Curvature** of the pitch line at any point should not be less than $d + {}^1/_8$ in. where $d$ = diameter of cam roller, in. Otherwise there will be a loss of rise of the follower which will vary in accordance with the sharpness of the angle of intersection of two adjoining curves on the cam.

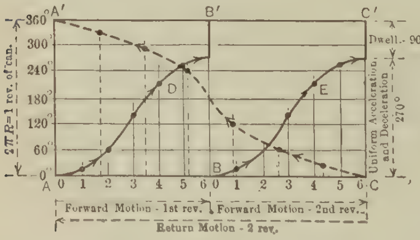


Fig. 40

**Rollers for Cylinder Cams.**—The preferred form of rollers that work in the slot of a cylinder cam is the frustum of a cone, with a corresponding taper of the cross-section of the slot. Straight rollers do not have a true rolling action because of the difference in linear velocity of the top and the bottom of the slot. This sliding of a portion of the roller tends to wear both roller and slot, resulting in eventual backlash and faulty action.

The method of determining the taper of roller and slot is shown in Fig. 41. Draw *AB* equal to the total throw of the cam. Draw *AC* perpendicular to *AB* and of a length equal to the length of pitch line on the surface of the cam through which correct cam action is necessary; for example, lengths *AD* or *BE* in Fig. 40. Lay out *CD* equal to the length of the pitch line at the bottom of the slot. This may be determined by redrawing, for example, Fig. 40, making $AA' = 2\pi r$, where $r$ = diameter of cam cylinder at bottom of slot. At $D$, draw $DE$ parallel to $AB$ and equal in length to $AB$. Then $CB$ equals the developed length of the helix traversed by the top of the roller and $CE$ equals the developed length of the helix traversed by the bottom of the roller. If $CB = a$, $CE = b$, $D$ = diameter of roller at top, and $d$ = diameter of roller at bottom, $d = bD/a$.

The head of the pin on which the roller is mounted should be recessed into the roller at least $^1/_{64}$ in. to avoid sliding contact with the bottom of the slot.

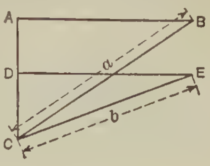


Fig. 41

## 4. THE GYROSCOPE

The Gyroscope is in effect a heavy disc spinning rapidly on its axis, and so mounted that its center is stationary while the axis may take any position whatever. See Fig. 42. A characteristic of the gyroscope, which is utilized in stabilizers to prevent the rolling of ships, in the gyro-compass, in monorail cars, etc., is that when the disc is spinning in a given plane, it resists any attempt to cause it to spin in a different plane. In Fig. 43, the disc of radius $r$ spins about the axis $ZZ$. The disc lies in the plane $QRST$, and the axis $ZZ$ in plane $ABCD$, perpendicular to $QRST$. Now let a force $F$ be applied at one end of the axis $ZZ$, tending to move the disc about its stationary center $O$ into a plane at an angle to $QRST$. The movement of the axis in the direction of $F$ will set up a force $G$, perpendicular to plane $ABCD$ and parallel to plane $QRST$. If this *gyroscopic reaction* $G$ be resisted by some means, as for instance, having the ends of the axis bear against a guide parallel to $ABCD$, the resistance $G'$, equal and opposite to $G$ will tend to impede movement in the direction of $F$. The direction of $G$ is determined by considering $F$ as a pull applied to the end of the axis. Then $G$ will act in the same direction as a point on the disc, in the plane of $F$ and on the same side of the axis, is moving.

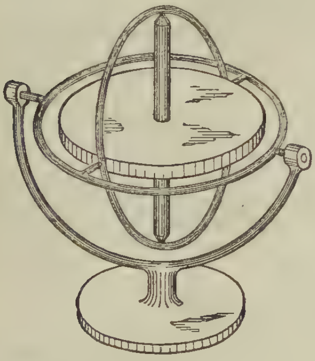


Fig. 42. Gyroscope

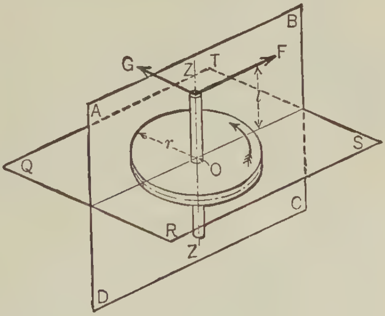


Fig. 43. Forces in the Gyroscope

The magnitude of the forces is as follows. Let $F$ = force applied to move the axis, $m$ = mass of the disc, $= w/g$, $R$ = its radius of gyration, $a$ = acceleration at any instant due to application of force $F$, $l$ = length of lever arm $= OZ$, $N$ = r.p.m. of disc, $v$ = velocity corresponding to acceleration $a$. Then $F = {}^1/_2\, mR^2\, a$, and

$$Gl = mR^2\,(\pi N/30)\,(vl).$$

For a mathematical analysis of the forces in the gyroscope and a discussion of its applications, see Seeley and Ensign, Analytical Mechanics for Engineers, p. 346 (John Wiley & Sons) and Deimel, Mechanics of the Gyroscope (Macmillan).

# MECHANICS

In the design of structures and machines, certain equations and relations, developed by the science of mechanics, frequently must be used. The more commonly used formulas are given below. For a mathematical discussion of these relations, see Eshbach, Handbook of Engineering Fundamentals, Section 4, forming Vol. 1 of this series.

## 1. CENTER OF GRAVITY

The center of gravity of a body or system of bodies rigidly connected is the point about which, if suspended, all parts will be in equilibrium, *i.e.*, without tendency to rotate. In bodies of uniform density, the center of gravity is the center of magnitude.

The center of gravity of **two bodies** is at a point between them, whose distance from their respective centers of gravity is inversely proportional to their weights. The center of gravity of **more than two bodies** is located by determining the center of gravity of two of them; then considering these two as a single body whose weight is concentrated at their common center of gravity, the center of gravity with reference to the third body is located. The process is repeated until all the bodies in the system have been considered.

The center of gravity of regular figures, whether plane or solid, is the same as their geometrical center. The center of gravity of symmetrical figures lies on the axis of symmetry. The center of gravity may be determined by suspending a body or figure from two different points and dropping a plumb line from the point of suspension. The intersection of the plumb lines will be the center of gravity.

The location of the center of gravity of the more common lines, areas and solids is as follows:

**LINES.—Straight Line.**—At middle point of the line.

**Semi-circumference.**—On middle radius $R$, at distance 0.6366 $R$ above diameter.

**Quadrant.**—On middle radius $R$, at distance 0.90028 $R$ from center.

**Circular Arc** of subtended angle $\alpha$, on middle radius $R$, at distance from center of ($R \sin {}^1/_2\,\alpha / {}^1/_2\,\alpha$ in radians).

**Cycloid,** generated by circle of diameter $D$, on perpendicular to middle of base, and ${}^2/_3\,D$ above base.

**PLANE FIGURES.—Triangle.**—At the intersection of two medians $L_1$ and $L_2$, or at a distance of ${}^1/_3\,L$ above the base.

**Parallelogram.**—At intersection of diagonals.

**Trapezium or Trapezoid.**—At the intersection of the two lines joining the centers of gravity of the two triangles formed by drawing first one diagonal and then the other.

**Sector of Circle** of radius $R$, chord $C$, arc $L$; on radius bisecting arc, at ($2\,CR/3\,L$) from center.

**Semi-circle.**—On middle radius, 0.4244 $R$ from center.

**Quadrant.**—On middle radius, 0.600 $R$ from center.

**Segment of Circle** of chord $C$ and area $A$, $C^3/12\,A$ from center.

**Parabolic Surface.**—In the axis, ${}^3/_5$ of its length from vertex.

**Semi-parabola** (surface).—${}^3/_5$ of axis from vertex, and ${}^3/_8$ of semi-base from axis.

**Quadrant of Ellipse** of semi-axes $a$ and $b$. At intersection of parallels to the axes $a$ and $b$, and distant from them, respectively, $\{b - {}^4/_3\,(b/\pi)\}$ and $\{a - {}^4/_3\,(a/\pi)\}$.

**SOLID BODIES.—Prism or Cylinder** (parallel bases).—At middle point of line joining centers of gravity of the bases.

**Oblique Frustum of Right Circular Cylinder** of radius $R$, mean height $H$. In the plane of symmetry at $\{{}^1/_2\,H + (R^2 \tan\alpha / 8\,H)\}$ above base, where $\alpha$ is angle of oblique section to the base, and at (${}^1/_4\,R^2 \tan\alpha / H$) from the axis.

**Cone or Pyramid.**—In the axis, ${}^1/_4$ of its length above the base.

**Paraboloid.**—In the axis, ${}^2/_3$ of its length from the vertex.

**Frustum of Cone or Pyramid.**—In axis, at distance above base of

$$[(a/4) - \{3\,a'^3/4(a^2 + aa' + a'^2)\}],$$

where $a$ = length of line from vertex of complete cone to center of gravity of base, and $a'$ = that portion of it between vertex and top of frustum.

**Hemisphere.**—On radius $R$, perpendicular to base, at ${}^3/_8\,R$ above base.

**Hollow Hemisphere.**—Radii $R$ and $r$. On radius perpendicular to base at

$$\{{}^3/_8\,(R^4 - r^4)/(R^3 - r^3)\} \text{ above base.}$$

**Segment of Sphere.**—Radius $R$; height of segment, $H$. On radius perpendicular to base, at $\{^3/_4\,(2\,R - H)^2/(3\,R - H)\}$.

**Sector of Sphere.**—Radius $R$; height of segment subtended by sector, $H$. On the middle radius, at $^3/_8\,(2\,R - H)$ from center.

**Ellipsoid** of semi-axes, $A$, $B$, and $C$. If an ellipsoid be divided into 8 parts by planes passing through each axis, the center of gravity of each part will lie in the point of intersection of three planes parallel to $A$, $B$, and $C$, and at a distance from them respectively, of $^3/_8\,B$, $^3/_8\,C$ and $^3/_8\,A$. See Fig. 1.

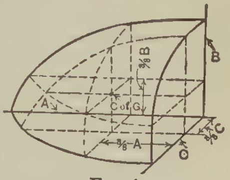


FIG. 1

## 2. MOMENT OF INERTIA

**THE MOMENT OF INERTIA OF A BODY** with respect to an axis is the algebraic sum of the products of the weight of each elementary particle multiplied by the square of its distance from the axis. If $w_1, w_2 \ldots w_n$ represent the weights of the several particles, and $r_1, r_2 \ldots r_n$, their respective distances from axis $XX$, then the moment of inertia with respect to the axis $XX$ is $I = w_1 r_1^2 + w_2 r_2^2 + \ldots w_n r_n^2$. $I$ is least when the axis passes through the center of gravity of the body.

Moment of inertia as applied to areas, as the cross-section of structural beams, equals $ar^2$, where $a$ is any elementary area. Moments of inertia of the more commonly used bodies, about different axes, are given in Table 1. Moments of inertia of structural shapes are given on pp. 17–50 to 17–72.

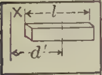
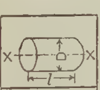
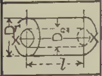
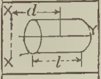
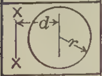
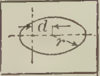
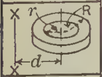
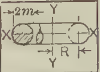
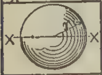
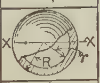
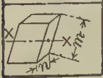
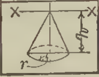
**Moment of Inertia about Parallel Axes.**—If the moment of inertia $I$ of an area or solid, about an axis through the center of gravity, is known, its moment of inertia $I'$ about any parallel axis may be found by the formula $I' = I + (d^2 \times A)$, where $d$ = distance between the two axes and $A$ = the area or weight.

**Polar Moment of Inertia.**—If the moments of inertia of an area with respect to two rectangular axes in the same plane are known, its moment of inertia about an axis perpendicular to this plane, and passing through the intersection of the axes is equal to the sum of the moments of inertia about the two rectangular axes. This latter is known as the polar moment of inertia, and is designated by $J$. If $I$ and $I'$ are the moments of inertia about the rectangular axes $XX$ and $YY$, $J = I + I'$.

**Moment of Inertia about Any Axis.**—If $b$ = breadth and $h$ = depth of a rectangular section, its moment of inertia about its central axis (parallel to the breadth) is $^1/_{12}\,b\,h^3$; and about one side is $^1/_3\,b\,h^3$. If a parallel axis exterior to the section is taken, and $d$ = distance of this axis from the farthest side and $d_1$ = its distance from the nearest side, $(d - d_1) = h$, the moment of inertia about this axis is $^1/_3\,b\,(d^3 - d_1^3)$.

### Table 1.—Moment of Inertia of Commonly Used Bodies

$W$ = weight of body; $w$ = weight of elementary unit of body; $d$ = distance of center of gravity of body from axis.

| | | | |
|---|---|---|---|
|  | Solid rod, axis perpendicular to axis of rod,<br>$I = W\left\{(l^2/3) + d^2\right\}$ |  | Solid cylinder, axis coincident with axis of cylinder,<br>$I = WD^2/8$ |
|  | Hollow cylinder about its axis,<br>$I = {}^1/_8\,W(D_1^2 + D_2^2)$.<br>Thin hollow cylinder,<br>$I = {}^1/_4\,WD_1^2$ |  | Solid cylinder, axis perpendicular to axis of cylinder,<br>$I = W\left\{\frac{D^2}{16} + \frac{l^2}{12} + d\right\}$ |
|  | Thin circular plate, axis in its own plane,<br>$I = W\left\{(r^2/4) + d^2\right\}$ |  | Thin circular plate, axis perpendicular to plate,<br>$I = W\left\{(r^2/2) + d^2\right\}$ |
|  | Circular ring, axis perpendicular to plane,<br>$I = W\left\{{}^1/_2(R^2 + r^2) + d^2\right\}$ |  | Circular ring about axis XX,<br>$I = w\,\pi R m^2(R^2 + {}^5/_4 m^2)$.<br>About axis YY,<br>$I = {}^1/_2\,w\,\pi^2 R^2 m^2(4R^2 + 3m^2)$ |
|  | Solid sphere about diameter,<br>$I = {}^2/_5\,Wr^2$ |  | Hollow sphere about diameter,<br>$I = {}^8/_{15}\left\{w\,\pi(R^5 - r^5)\right\}$<br>Thin sphere, $I = {}^2/_3\,WR^2$ |
|  | Rectangular prism, axis through c. of g. perpendicular to face $mn$,<br>$I = {}^1/_{12}\,W(m^2 + n^2)$ |  | Right circular cone, axis XX,<br>$I = {}^3/_5\,W({}^1/_4\,r^2 + h^2)$<br>About its axis of revolution,<br>$I = {}^3/_{10}\,W\,r^2$ |

**Moment of Inertia of Compound Shapes.**—The moment of inertia of a compound shape about any axis equals the sum of the moments of inertia with reference to the same axis of all the rectangular portions composing it. It also equals the moment of inertia about a parallel axis through its center of gravity + (area of section × square of distance between axes). From this, the moments of inertia or radii of gyration of any single section being known, values may be found for any combination of these sections.

## 3. CENTER AND RADIUS OF GYRATION

**THE CENTER OF GYRATION** with reference to an axis is that point at which the entire weight of a body may be concentrated without changing its moment of inertia. In a revolving body, it is the point at which the whole weight of the body may be conceived to be concentrated.

**RADIUS OF GYRATION** is the distance from the axis to the center of gyration. Let $W$ = weight of the body; $I$ = its moment of inertia; $k$ = its radius of gyration. Then $I = W k^2 = \Sigma w\, r^2$; $k = \sqrt{I/W} = \sqrt{\Sigma w\, r^2 / W}$. The radius of gyration is least when the axis passes through the center of gravity, and is then known as the principal radius of gyration. Let $k$ = principal radius of gyration; $k'$ = any other radius of gyration; $d$ = distance between the axes of $k$ and $k'$. Then $k' = \sqrt{k^2 + d^2}$. Table 2 gives principal radii of gyration of the more common areas and solids, except structural shapes, for which see pp. 17–50 to 17–72.

## 4. VELOCITY AND ACCELERATION

**VELOCITY** is the rate of motion, or speed of a body at any instant. If $s$ = space, ft., passed over in $t$ sec., and $v$ = velocity, ft. per sec., when $v$ is uniform, $s = vt$; $v = s/t$. If $v$ varies uniformly, the mean velocity $v_m = 1/2\,(v_1 + v_2)$, where $v_1$ and $v_2$ are the velocities at beginning and end of time $t$, respectively. $s = 1/2\,(v_1 + v_2)t$.

**Table 2.—Principal Radii of Gyration and Squares of Radii of Gyration**

(For radii of gyration of sections of columns, see pp. 17–50 and 17–77)

| Surface or Solid | Radius of Gyration | Square of Radius of Gyration |
|---|---|---|
| Parallelogram: height $h$, axis at its base | $0.5773\,h$ | $1/3\,h^2$ |
| Parallelogram: height $h$, " mid-height | $0.2886\,h$ | $1/12\,h^2$ |
| Straight rod: length $l$, or thin rectangular plate, axis at end | $0.5773\,l$ | $1/3\,l^2$ |
| Straight rod: length $l$, or thin rectangular plate, " mid-length | $0.2886\,l$ | $1/12\,l^2$ |
| Rectangular prism: axes $2a$, $2b$, $2c$, referred to axis $2a$ | $0.577\sqrt{b^2 + c^2}$ | $(b^2 + c^2) \div 3$ |
| Parallelopiped: length $l$, base $b$, axis at one end, at mid-breadth | $0.289\sqrt{4l^2 + b^2}$ | $\dfrac{4l^2 + b^2}{12}$ |
| Hollow square tube: | | |
| outer side $h$, inner $h'$, axis mid-length | $0.289\sqrt{h^2 + h'^2}$ | $(h^2 + h'^2 \div 12$ |
| very thin, side $= h$, axis mid-length | $0.408\,h$ | $h^2 \div 6$ |
| Thin rectangular tube: sides $b$, $h$, axis mid-length | $0.289\sqrt{\dfrac{h + 3b}{h + b}}$ | $\dfrac{h^2}{12} \cdot \dfrac{h + 3b}{h + b}$ |
| Thin circular plate: radius $r$, diameter $h$, axis on diameter | $1/2\,r$ | $1/4\,r^2 = h^2 \div 16$ |
| Flat circular ring: diameters $h$, $h'$, axis on diameter | $1/4\sqrt{h^2 + h'^2}$ | $(h^2 + h'^2) \div 16$ |
| Solid circular cylinder: length $l$, axis on diameter at mid-length | $0.289\sqrt{l^2 + 3r^2}$ | $\dfrac{l^2}{12} + \dfrac{r^2}{4}$ |
| Circular plate: solid wheel of uniform thickness, or cylinder of any length, referred to axis of cylinder | $0.7071\,r$ | $1/2\,r^2$ |
| Hollow circular cylinder, or flat ring: $l$, length; $R$, $r$, outer and inner radii. Axis: 1, longitudinal axis; 2, diameter at mid-length | $0.7071\sqrt{R^2 + r^2}$ | $(R^2 + r^2) \div 2$ |
| | $0.289\sqrt{l^2 + 3(R^2 + r^2)}$ | $\dfrac{l^2}{12} + \dfrac{R^2 + r^2}{4}$ |
| Same: very thin, axis its diameter | $0.289\sqrt{l^2 + 6R^2}$ | $\dfrac{l^2}{12} + \dfrac{R^2}{2}$ |
| " radius $r$; axis, longitudinal axis | $r$ | $r^2$ |
| Circumference of circle, axis its center | $r$ | $r^2$ |
| " " " " " diameter | $0.7071\,r$ | $1/2\,r^2$ |
| Sphere: radius $r$, axis its diameter | $0.6325\,r$ | $2/5\,r^2$ |
| Spheroid: equatorial radius $r$, revolving polar axis $a$ | $0.6325\,r$ | $2/5\,r^2$ |
| Paraboloid: $r$ = radius of base, revolving on axis | $0.5773\,r$ | $1/3\,r^2$ |
| Ellipsoid: semi-axes $a$, $b$, $c$; revolving on axis $2a$ | $0.4472\sqrt{b^2 + c^2}$ | $\dfrac{b^2 + c^2}{5}$ |
| Spherical shell: radii $R$, $r$, revolving on its diameter | $0.6325\sqrt{\dfrac{R^5 - r^5}{R^3 - r^3}}$ | $\dfrac{2}{5}\,\dfrac{R^5 - r^5}{R^3 - r^3}$ |
| Same: very thin, radius $r$ | $0.8165\,r$ | $2/3\,r^2$ |
| Solid cone: $r$ = radius of base, revolving on axis | $0.5477\,r$ | $0.3\,r^2$ |

If the velocity varies, but not uniformly, $v$ for a very short interval of time $= s/t$, or in the calculus, $v = ds/dt$.

**ACCELERATION,** $a$, is the change in velocity occurring in unit time. The unit of $a$ is 1 ft. per sec. per sec. For uniformly accelerated motion, $a$ is constant.

$a = (v_2 - v_1)/t$; $v_2 = (v_1 + at)$; $v_1 = (v_2 - at)$; $t = (v_2 - v_1)/a$.

If the body starts from rest, $v_1 = 0$, and the mean velocity $v_m = v_2/2$; $v_2 = 2v_m$; $a = v_2/t$; $v_2 = at$; $v_2 - at = 0$; $t = v/a$.

For acceleration due to gravity, and of falling bodies, see p. 8–21.

**Deceleration or Retarded Motion.** — If a body starts with a velocity $v_1$ and comes to rest, $v_2 = 0$, and $s = {}^1/_2\, v_1 t$. In any case, if the change in velocity is $v$, $s = {}^1/_2\, vt = v^2/2a = {}^1/_2\, at^2$.

For a body starting from or ending at rest, $v = at$; $s = {}^1/_2\, vt = {}^1/_2\, at^2$; $v^2 = 2as$.

**CURVES OF VELOCITY AND ACCELERATION.**—If the velocity or acceleration are not uniform, or are represented by complex equations, they may be studied by plotting space, velocity or acceleration against time. In Fig. 2, the space passed over is plotted as ordinates with seconds of time as abscissas. The angle made by the tangent to the curve at any point determines the velocity at that instant. Thus, the tangent $BC$ at point $A$ intersects the axis of abscissas at $C$ and the velocity at $A$ is $(AD/CD) = {}^3/_2$ ft. per sec. Since the slope of the tangent is upward, the velocity is increasing, *i.e.*, the point under consideration is being accelerated. The slope of tangent at $F$ is downward, and hence the motion of the point is being retarded. If the curve at any point is parallel to the axis of abscissas, the motion is uniform.

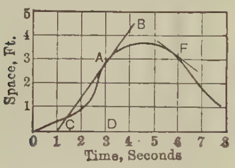


Fig. 2

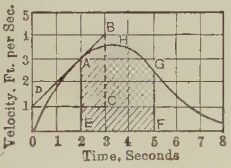


Fig. 3

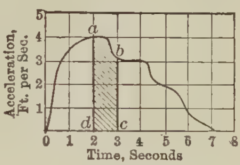


Fig. 4

In Fig. 3, velocity is plotted as ordinates and time as abscissas. The angle of the tangent to the curve at any point will represent the acceleration at that instant. The acceleration at $A$ is $(BC/CD) = {}^3/_3 = 1$ ft. per sec. per second. As before, an upward slope represents acceleration, while a downward slope would represent retardation or deceleration. The area under the curve between any two ordinates will represent the space passed over in that interval of time. Thus area $AEFGH \div EF = 10 \div 3 = 3\,{}^1/_3$ ft.

In Fig. 4, acceleration is plotted as ordinates and time as abscissas. The angle of the tangent to the curve at any point will represent the velocity at that instant and the area under the curve between two ordinates $ad$ and $bc$ will represent the change in velocity during the interval.

For *uniform motion* the space-time curve is a straight line, inclined to the axis of abscissas, and the velocity-time curve is a straight line parallel to the axis of abscissas.

For *constant acceleration* or *constant retardation* the space-time curve is a parabola, and the velocity-time curve is a straight line, inclined away from or toward the axis of abscissas, depending upon whether the velocity increases or decreases. The acceleration-time curve is a straight line, parallel to the axis of abscissas.

**ANGULAR VELOCITY OF A TURNING BODY.**—Let $r$ = radius of a turning body, ft.; $n$ = rev. per min.; $v$ = linear velocity of a point on the circumference, ft. per sec., and $60\,v$ = velocity in ft. per min. Then $v = 2\pi rn/60$; $60\,v = 2\pi rn$.

**ANGULAR VELOCITY** denotes the angle through which any radius of a body turns in 1 sec., or the rate at which any point in it, of unit radius, is moving in ft. per sec. The unit of angular velocity is the *Radian* $= 180/\pi$ deg. $= 57.3$ deg. If $A$ = angular velocity, $v = Ar$; $A = v/r = 2\pi rn/60$. The linear velocity of a point in a rotating body is proportional to its distance from the axis.

**ANGULAR ACCELERATION** is the rate at which angular velocity varies with respect to time.

## 5. FORCE, WORK AND ENERGY

**UNIFORMLY ACCELERATED MOTION.**—Let $f$ be a constant force, acting for $t$ sec. on a body, at rest at the beginning of the time, but free to move without friction, the weight of the body being $w$ lb. The body will move at a gradually increasing velocity $v$, which varies as $ft/w$.

If $v$ is in ft. per sec., $f$ in standard pounds of force,* $t$ in seconds, and $w$ in pounds, the equations that govern the motion of the body are as follows:

Let $K$ = a constant derived by experiment. Then $v = Kft/w$. The value of $K$ at sea level at latitude 45 deg. is 32.1740, and is twice the distance that a body *in vacuo* would fall in 1 sec. It usually is designated as $g$.† The fundamental equation then is

$$v = ftg/w. \qquad [1]$$

Let $S$ = distance through which the body moves. If $v$ varies directly as time (uniformly accelerated motion), then $S = v_m t$, where $v_m$ = mean velocity during time $t$. If $v_0$ = initial velocity, and $v$ = final velocity at end of time $t$, $v_m = (v_0 + v)/2$. For a body starting from rest $v_0 = 0$, and $v_m = v/2$. Hence, $S = {}^1/_2\, vt$; $v = 2\,S/t$; $t = 2S/v$. Substituting $v = 2S/t$ in equation [1],

$$S = ft^2\, g/2\,w. \qquad [2]$$

If $f = w$, which is true of a falling body at latitude 45 deg. at sea level, equation [1] becomes $v = gt$. Substitute in this equation $t = S/v$; then

$$v = 2\,gS/v, \quad \text{and} \quad v = \sqrt{2\,gS}. \qquad [3]$$

In equation [2], put $f = w$. Then $S = {}^1/_2\, gt^2$.

From 1, by transposition,

$$ft = w \times v/g = v \times w/g. \qquad [4]$$

$w/g$, the mass of the body frequently is designated as $m$, whence [4] becomes

$$ft = mv. \qquad [5]$$

Substitute in [4], $t = 2S/v$. Then $f\,2\,S/v = v \times w/g$, whence,

$$fS = {}^1/_2\, mv^2 \qquad [6]$$

By definition acceleration $a = v/t$, whence from [5],

$$f = ma‡. \qquad [7]$$

If two forces $f$, $f_1$, act on a body at different times, producing accelerations $a$, $a_1$, then $\dfrac{f = ma}{f_1 = ma_1}$, whence

$$f/f_1 = a/a_1. \qquad [8]$$

These eight equations and their transformations will solve all problems of uniformly accelerated motion.

**FORCE OF ACCELERATION.**—To produce an acceleration of $g$ (*i.e.*, 32.174 ft. per sec. per sec.), force required, $f = mg = (w/g)g = w$, or the weight of the body. Since $f = ma$, $f = m(v_2 - v_1)/t$, where $v_2$ and $v_1$ = respectively, initial and final velocities at beginning and end of time $t$. Also $f = mg = (w/g)/\{(v_2 - v_1)/t\} = (w/g)a$; $f/w = a/g$. In problems where the local attraction of gravity is a factor, the local value of $g$ must be used if accuracy is necessary.

Example.—Required the tension in a cord lifting 100 lb. vertically through 80 ft., at velocity uniformly increasing from 0 to 40 at the end of 4 sec. Local value of $g$ = 0.998 × standard value = 32.108; $v_m = 20$; final velocity $v_2 = 2v_m = 40$. Acceleration $a = v_2/4 = 10$. Force required to produce acceleration $= f_a = ma = wa/g = (100 \times 10)/32.174 = 31.08$ lb. The standard value of $g$ (32.174) is used here, since force required for acceleration is independent of gravity. Force required to lift the weight without acceleration $= f_w = 100 \times 0.998 = 99.8$ lb. The factor 0.998 is used in this equation because the force of gravity in the given locality is less than standard. Total tension in cord $= f_a + f_w = 130.88$ lb.

**Resistance to Acceleration** equals the force necessary to produce acceleration $= (w/g)\{(v_2 - v_1)/t\}$.

**FORMULAS FOR ACCELERATED MOTION.**—For cases of uniformly accelerated motion, other than those of falling bodies, we have the formulas already given, $f = (w/g)a$;

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* The pound avoirdupois is the unit of force used by American and British engineers. It is the force with which gravity attracts 1 lb. of matter at sea level at 45 deg. latitude, and which would give to 1 lb. of matter an acceleration of 32.1740 ft. per sec.

In the centimeter-gram-second system, the unit of force is the *dyne*. It is that force which acting on a mass of 1 gram will produce in 1 second a velocity of 1 centimeter per second. A dyne = 1/980.665 gram.

The absolute unit of force, used in physical laboratories, is the *Poundal*. It is that force which acting for 1 second on a mass whose weight is 1 lb. at London will produce a velocity of 1 ft. per second.

† $g$, commonly called acceleration due to gravity, may be considered as: an abstract figure, the constant in equation [1]; the velocity acquired at the end of 1 sec. by a freely falling body; the distance that a body would travel in 1 sec. at its then velocity if the force ceased to act and velocity remained constant.

‡ Equation [7] sometimes is read "force equals mass times acceleration," which is strictly true in the dyne-centimeter-gram-second, or "absolute" system of measurements, in which force is measured in dynes, but it is not true in the pound-foot-second system, nor in the metric system where the kilogram is used as a unit of both force and quantity of matter, unless it is understood that the word "mass" means the quotient of $W$ divided by $g$.

$f = (w/g)\{(v_2 - v_1)/t\}$. If the body starts from rest, $v_1 = 0$, $v_2 = v$, and $f = (w/g)(v/t)$; $f g t = w v$. We also have $S = {}^1/_2\, v t$. Transforming and substituting 32.174 for $g$,

$$f = w v^2/64.35\, S = w v/32.17\, t = w S/16.09\, t^2. \quad . \quad . \quad . \quad . \quad . \qquad [9]$$

$$w = 32.17\, f t/v = 64.35\, f S/v^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [10]$$

$$S = w v^2/64.35\, f = 16.09\, f t^2/w = v t/2. \quad . \quad . \quad . \quad . \quad . \qquad [11]$$

$$v = 8.02\sqrt{f S/w} = 32.17\, f t/w. \quad . \quad . \quad . \quad . \quad . \quad . \qquad [12]$$

$$t = w v/32.17\, f = (1/4.01)\sqrt{w S/f}. \quad . \quad . \quad . \quad . \quad . \qquad [13]$$

For varying velocity, $f = w\{(v_2^2 - v_1^2)/64.35\, S\}$. . . . . . . . . . . . . . . . [14]

**WORK OF ACCELERATION.**—The work $P$ done in accelerating a body is $f \times S$, where $f$ = force producing acceleration and $S$ = distance covered in time $t$. The force, $f = m a = (w/g)\{(v_2 - v_1)/t\}$, and

$$P = (w/g)\{(v_2 - v_1)/t\}S \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [15]$$

EXAMPLE.—Work required to move horizontally 80 ft. in 4 sec. a frictionless body weighing 100 lb., at uniformly increasing velocity. Mean velocity, $v_m = 20$; initial velocity $v_1 = 0$; final velocity $v_2 = 2\, v_m = 40$; $a = (v_2 - v_1)t = 40/4 = 10$; $f = (w/g)a = (100/32.16) \times 10 = 31.1$ lb.; $S = 80$; $P = f S = 31.1 \times 80 = 2488$ ft.-lb. Energy stored in the body at the final velocity $v = 40$, equals the work of acceleration.

$E = {}^1/_2\, m v^2 = {}^1/_2\, (w/g)v^2 = (100 \times 40^2)/(2 \times 32.16) = 2488$ ft. lb.

**WORK OF ACCELERATED ROTATION.**—Let $A$ = angular velocity, radians, of a particle of weight $w$, rotating at unit radius $r$ around an axis. The velocity $v_1 = A_1 r$. If angular velocity is accelerated to $A_2$, velocity of particle becomes $v_2$. Increase in velocity is $(v_2 - v_1) = r(A_2 - A_1)$, and the work of acceleration is

$$P_a(w/g)\{(v_2^2 - v_1^2)/2\} = (w r^2/g)\}(A_2^2 - A_1^2)/2\}. \quad . \quad . \quad . \quad . \quad . \qquad [16]$$

The work of a rotating body around an axis is

$$\Sigma(w/g) \times {}^1/_2\, (v_2^2 - v_1^2) = (A_2^2 - A_1^2)/2g \times \Sigma w r^2, \quad . \quad . \quad . \quad . \quad . \qquad [17]$$

where $\Sigma w r^2$ = moment of inertia of the body.

**MEASURE OF ENERGY** in a body is the product of its weight and the height from which it must fall to acquire actual velocity. If $v$ = velocity, ft. per sec., $h$ = height, ft., corresponding to velocity $= v^2/2g$ (see Falling Bodies below), and if $w$ = weight, lb., energy, $E = {}^1/_2\, m v^2 = w v^2/2g = w h$. Since energy is the capacity for doing work, the units of work and energy are equivalent, or $FS = {}^1/_2\, m v^2 = w h$.

In a rotating body, $E = A^2 I/2g$, where $A$ = angular velocity, radians, and $I$ = moment of inertia. Also, $E = w v^2/2g$, where $v$ = velocity of center of gyration.

**THE LAW OF GRAVITATION** states that two bodies attract each other with a force directly proportional to their masses and inversely proportional to the square of the distance between them. Let $F$ = force with which two bodies of masses $m_1$ and $m_2$ attract each other, and $d$ = distance between them. Then $F = C m_1 m_2/d^2 = C(w_1/g)(w_2/g)/d^2 = C w_1 w_2/d^2 g^2$ where $C$ = gravitation constant of two spheres of unit mass, *i.e.*, the force with which they attract each other through unit distance. $C = 3.31 \times 10^{-11}$ if $F$, $w$ and $d$ are in pounds and feet.

**MOMENTUM. IMPULSE.**—Momentum is the product of the mass and velocity of a body, *i.e.*, momentum $= m v = (w/g)v$. In equation [5], $f t = m v$, the expression $f t$ is sometimes termed the impulse, whence Impulse = Momentum.

**FALLING BODIES.**—The formulas for falling bodies are derived from the equations [9] to [14] by putting $f = w$. If $g$ be taken as 32.16 (value at 40 deg. latitude), $v$ = velocity, ft. per sec. at end of time $t$; $h$ = height of fall, ft., then

$$v = g t = 32.16\, t = \sqrt{2\, g h} = 8.02\sqrt{h} = 2h/t. \quad . \quad . \quad . \quad . \quad . \quad . \qquad [18]$$

$$h = g t^2/2 = 16.08\, t^2 = v^2/2g = v^2/64.32 = v t/2. \quad . \quad . \quad . \quad . \quad . \qquad [19]$$

$$t = v/g = v/32.16 = \sqrt{2h/g} = \sqrt{h}/4.01 = 2h/v. \quad . \quad . \quad . \quad . \quad . \qquad [20]$$

$u$ = space fallen through during $T$th sec.

$$= h_t - h_{t-1} = (g t^2/2) - g(t - 1)^2/2 = (g/2)(2t - 1) = g(t - 1)/2. \qquad [21]$$

Solving the above equations, the values of $v$, $h$, $t$, $u$, are as follows:

| $t =$ | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|
| $g = 32.16 \times$ | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| $v = 32.16 \times$ | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| $u = (32.16/2) \times$ | 1 | 3 | 5 | 7 | 9 | 11 | 13 | 15 | 17 | 19 |
| $h = (32.16/2) \times$ | 1 | 4 | 9 | 16 | 25 | 36 | 49 | 64 | 81 | 100 |

**Value of $g$** increases with latitude and decreases with elevation above sea-level. At Philadelphia, latitude 40 deg., $g = 32.16$. Everett's formula for $g$ is

$$g = 32.173 - 0.082 \cos 2 \text{ lat. } 45 \text{ deg.} - 0.000003 \text{ height in ft.} \quad . \quad . \quad . \qquad [22]$$

The standard value of $g$ at latitude 45 deg. given by International Conference on Weights and Measures, Paris, 1901, is 32.1740. The value of $\sqrt{2g}$ decreases about 0.0004 per 1000 ft. elevation above sea-level. For all ordinary calculations in the U. S., $g = 32.16$, and $\sqrt{2g} = 8.02$. In England, $g = 32.2$; $\sqrt{2g} = 8.025$. C. S. Pierce gives limiting values in the U. S. as: At latitude 49 deg., at sea-level, $g = 32.186$; at latitude 25 deg., 10,000 ft. above sea-level, $g = 32.089$.

Local values of $g$ are used in calculations involving local gravitational force, as falling bodies, lifting weights, and water powers. In all cases involving the expression $w/g$, the standard value, $g = 32.1740$ should be used.

**CENTRIFUGAL FORCE.**—A body revolving in a curved path of radius $= R$ ft. exerts a force, called centrifugal force, on the arm or cord that restrains it from moving in a straight line. If $W$ = weight of body, lb., $N$ = rev. per min., $v$ = linear velocity of center of gravity of body, ft. per sec., $g = 32.174$,

$$v = \frac{2\,\pi RN}{60}. \qquad [23]$$

$$F = \frac{Wv^2}{gR} = \frac{Wv^2}{32.174\,R} = \frac{W4\,\pi^2\,RN^2}{3600\,g} = \frac{WRN^2}{2933.9} = 0.00034084\,WRN^2 \text{ lb.} \qquad [24]$$

If $n$ = rev. per sec., $F = 1.2270\,W\,r\,n^2$.

The force that restrains the body from moving in a straight line, *i.e.*, the tension in the cord or arm, is called *centripetal force*, and is equal and opposite to the centrifugal force.

## 6. OSCILLATION AND PERCUSSION

**CENTER OF OSCILLATION.**—A body oscillates about a fixed axis, outside of its center of gravity. The center of oscillation is a point on the line drawn from the center of gravity of the body, perpendicular to the axis, whose motion is the same as it would be were the entire mass of the body there concentrated and allowed to vibrate as a pendulum.

**RADIUS OF OSCILLATION** is distance from center of oscillation to point of suspension $= r = R^2/l$; $R$ = radius of gyration, $l$ = distance from point of suspension to center of gravity. Centers of suspension and oscillation are mutually convertible. The center of oscillation of a straight line, or uniform thin bar or cylinder of length $L$ suspended from and oscillating about one end, is $^2/_3\,L$ from the axis. If the point of suspension is $^1/_3\,L$, the center of oscillation is again $^2/_3\,L$ from the axis, *i.e.*, at end of bar. The time of oscillation $t$ is the same in both cases. If point of suspension is at center of gravity, in the simple equivalent pendulum $L = \infty$ and $t = \infty$.

A sphere of radius $R$ is suspended by a cord. Let $h$ = distance of axis of motion to center of sphere, $h'$ = distance of center of oscillation to center of sphere, $r$ = radius of oscillation $= h + h' = h + {}^2/_5\,(R^2/h)$. If the sphere oscillates about an axis tangent to the surface $h = R$ and $r = R + {}^2/_5\,R$.

**Table 3.—Length of Radius of Oscillation of Plane Figures**

| Figure | Direction of Vibrations | |
|---|---|---|
| | Perpendicular to Plane of Figure | In Plane of Figure |
| Isosceles triangle | $^3/_4$ altitude | |
| Circle | $^5/_8$ diameter | $^3/_4$ height |
| Parabola | $^5/_7$ height | $^5/_7$ height + $^1/_3$ parameter *<br>$^4/_7$ height + $^1/_2$ parameter † |
| Rectangle | | $^2/_3$ diagonal ‡ |

* Suspended by vertex. † Suspended by middle of base. ‡ Suspended by one angle.

**THE PENDULUM.**—The *Simple Pendulum* is an ideal body concentrated at the center of oscillation, suspended from a fixed point by a weightless string. A *Compound Pendulum* is a body of any form suspended from a fixed axis about which it oscillates by force of gravity. The equivalent simple pendulum has the same weight and the same moment of inertia referred to an axis through point of suspension, and it vibrates in the same time. The ordinary pendulum of given length vibrates in equal times when the angle of vibration does not exceed 2 deg. or 2 $^1/_2$ deg. on each side of the vertical. This property is *isochronism*.

Let $T$ = time of vibration, sec., $l$ = length of simple pendulum, $g$ = acceleration due to gravity, which is constant at a given locality. $T = \pi\sqrt{l/g}$; since $\pi$ = constant, $T \propto \sqrt{l/g}$, and in a given locality $T \propto \sqrt{l}$. If $l$ = constant, then for any locality $T \propto l\sqrt{g}$.

If $T$ = constant, $gT^2 = \pi^2\,l$; $l \propto g$, $g = \pi^2\,l/T^2$. This equation gives the force of gravity at any locality if $l$ of a simple pendulum vibrating seconds at that locality is known.

At New York, $l = 39.1017$ in. = 3.2585 ft., and $g = 32.16$ ft. The time of vibration of any pendulum at New York = $T\sqrt{l/39.1017} = 0.16\sqrt{l}$. The length of a pendulum at New York, having a given time of vibration is $l = T^2 \times 39.1017$ in.

The time of vibration may be varied by adding a weight or *bob* above the point of suspension, which lengthens time $T$. By varying the height of the upper bob, $T$ is varied.

Let $W$, $w$ = weight of lower and upper bobs, lb., respectively, $D$, $d$ = distance, in., of lower and upper bobs, respectively, from point of suspension. Then $w = W\{(39.1\,D - D^2)/(39.1\,d - d^2)\}$. Thus by adding a second bob, short pendulums may be made to vibrate as slowly as long ones.

**Conical Pendulum.**—A weight suspended by a cord and revolving at a uniform speed in the circumference of a circular horizontal plane whose radius is $r$, the distance of the plane below the point of suspension being $h$, is held in equilibrium by three forces—the tension in the cord, the centrifugal force, which tends to increase the radius $r$, and the force of gravity acting downward. If $v$ = velocity, ft. per sec. of the center of gravity of the weight, as it describes the circumference, $g = 32.16$, and $r$ and $h$ are taken in feet, the time in seconds of performing one revolution is (at New York or other place where $g = 32.16$)

$$t = 2\pi\, r/v = 2\pi\sqrt{h/g}; \qquad h = g\,t^2/4\pi^2 = 0.8146\,t^2.$$

If $t = 1$ second, $h = 0.8146$ foot = 9.775 inches.

**CENTER OF PERCUSSION** is identical with center of oscillation, and is the point at which the percussive action of a blow is the same as if the whole mass of the body were concentrated at the point.

**IMPACT.**—Let $m_1\ m_2$ = mass of two inelastic bodies; $v_1$, $v_2$ = their respective velocities. Then if these two bodies collide, they will move on as one mass with a common velocity $v$. $(m_1 + m_2)v = m_1 v_1 + m_2 v_2$, and $v = (m_1 v_1 + m_2 v_2)/(m_1 + m_2)$. If the bodies move in opposite directions, $v = (m_1 v_1 - m_2 v_2)/(m_1 + m_2)$. If $m_1 v_1 = m_2 v_2$, and the bodies impinge directly from opposite directions, $v = 0$; that is the bodies will be brought to rest.

**The Loss of Energy** resulting from the impact of inelastic bodies is the sum of the energies lost and gained by the bodies respectively.

$$^1/_2\, m_1 v_1^2 + {}^1/_2\, m_2 v_2^2 - {}^1/_2\, (m_1 + m_2)v_2 = {}^1/_2\, m_1(v_1 - v)^2 + {}^1/_2\, m_2(v_2 - v)^2.$$

Example.—Let $m_1 = 10$, $m_2 = 8$, $v_1 = 12$, $v_2 = 15$. $v = \{(10 \times 12) - (8 \times 15)\}/(10 + 8) = 0$.

$$\text{Energy loss} = ({}^1/_2 \times 10 \times 12^2) + ({}^1/_2 \times 8 \times 15^2)$$
$$= \{{}^1/_2 \times 10 \times (12 - 0)^2\} + \{{}^1/_2 \times 8 \times (15 - 0)^2\} = 1620 \text{ ft.-lb.}$$

The lost energy does internal work on the bodies by changing their shape and heating them.

**For Imperfectly Elastic Bodies** of elasticity $e = f_r/f_c$,

$$v'_1 = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2} - \frac{m_2\, e\,(v_1 - v_2)}{m_1 + m_2}; \qquad v'_2 = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2} + \frac{m_2\, e\,(v_1 - v_2)}{m_1 + m_2},$$

where $v'_1$ and $v'_2$ = respective velocities of the bodies after impact, $f_r$ = internal force tending to restore shape of body after compression, $f_c$ = force of compression, other notation being as before.

**For Perfectly Elastic Bodies,** $v'_1 - v'_2 = v_2 - v_1$; that is, relative velocities before and after impact are the same, and $m_1 v'_1 + m_2 v'_2 = m_1 v_1 + m_2 v_2$.

# FRICTION *

## 1. THE LAWS OF FRICTION

**References.**—Thomsen, The Practice of Lubrication (McGraw-Hill); Battle, Industrial Oil Engineering (Lippincott); Archbutt and Deeley, Lubrication and Lubricants (Griffin, London); Thurston, Friction and Lost Work (John Wiley & Sons); Kimball and Barr, Elements of Machine Design (John Wiley & Sons).

**FRICTION** is defined by Rankine as that force which acts between two bodies at their surface of contact so as to resist their sliding on each other, and which depends on the force with which the bodies are pressed together.

Friction is either useful or harmful, depending on its relation to the mechanism or process in which it occurs. Examples of useful friction are brakes, self-locking mechanisms, bolted or riveted joints, heaping loose materials, etc. Friction is harmful in bearings, as it wastes power, and in pipes where it retards flow. Friction in static structures tends to preserve them; in brakes it destroys material; in well-lubricated bearings it is accompanied neither by wear of the bearings nor by deterioration of the lubricant.

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* Staff revision.

**COEFFICIENT OF FRICTION.**—The ratio of the force required to slide a body along a horizontal plane surface to the weight of the body is called the coefficient of friction. It is equivalent to the tangent of the *angle of repose*, which is the angle of inclination, to the horizontal, of an inclined plane on which the body will just overcome its tendency to slide. The angle usually is denoted by $\theta$, and the coefficient by $f$ or $\lambda$: $f = \tan\theta$. If the inclined surface, Fig. 1*a*, upon which a loaded block rests without motion, is made gradually steeper until the block starts to slide, the angle at which the sliding starts is the angle of repose for static friction. After the motion has begun, if the angle is not reduced, the block usually will accelerate down the incline. If the angle then be reduced until the block slides with uniform velocity, the angle at which this occurs may be used to determine the friction opposing that particular velocity. The angle $\Delta$ shown in Fig. 1*b* is used as a measure of the manner in which granular materials may be heaped. This angle varies widely with the character of the material. Values of $f$ for a wide variety of materials, both at rest and in motion, are given in Table 1.

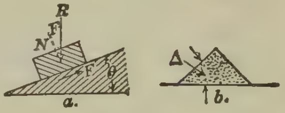

FIG. 1

**Table 1.—Coefficients of Friction of Various Materials***

| Material | In Contact with | At Rest or in Motion† | Lubrication | Coefficient of Friction $f$ | Authority‡ |
|---|---|---|---|---|---|
| | | **Metal and Metal** | | | |
| Brass | Brass | M | Dry | .173 | 1 |
| " | Cast Iron | M | Dry | .188 | 2 |
| " | " | M | Unct. | .111 | 2 |
| " | Wrt. Iron[1] | R | Dry | .136 | 1 |
| Bronze | Bronze | M | Dry | .199 | 2 |
| " | " | M | Unct.[44] | .133 | 2 |
| " | Cast Iron | M | Dry | .213 | 2 |
| " | " | M | Unct.[44] | .098 | 2 |
| " | Wrt. Iron[1] | M | Dry | .167 | 2 |
| Cast Iron | Bronze | M | Dry | .147 | 2 |
| " | " | M[2] | [3] | .054 | 2 |
| " | " | M | Unct.[44] | .132 | 2 |
| " | Cast Iron | R | Dry | .162 | 2 |
| " | " | A | Dry | .48 | 1 |
| " | " | M | Dry | .152 | 2 |
| " | Steel[4] | R | Dry | .155 | 1 |
| " | Wrt. Iron[1] | R | Dry | .166 | 1 |
| "[5] | Bronze | M | Unct.[44] | .140–.169 | 2 |
| "[5] | Cast Iron | .. | Ord.[6] | .073–.082 | 2 |
| "[5] | " | M | Unct.[44] | .130–.140 | 2 |
| Steel | Bronze | M | Dry | .152 | 2 |
| " | "[7] | M | Bath | .0004–.0008[8] | 3 |
| " | "[7] | M[9] | Bath[10] | .0011 | 3 |
| " | "[7] | M[11] | Bath[10] | .0023 | 3 |
| " | "[7] | M[12] | Bath[10] | .0044 | 3 |
| Steel | Bronze[7] | M[13] | Bath[10] | .0062 | 3 |
| " | Cast Iron | M | A† | .4 | 1 |
| " | Cast Iron[14] | M[15] | ...... | .0035–.0047 | 4 |
| " | Cast Iron[16] | M[17] | ...... | .036–.064 | 5 |
| " | Cast Iron | M | Dry | .204 | 2 |
| " | " | M | Unct.[44] | .108 | 2 |
| " | Steel | R | Dry | .149 | 2 |
| "[4] | Wrt. Iron[1] | R | Dry | .194 | 1 |
| Steel | W.M.[18] | M[19] | ...... | .002–.030 | 6 |
| " | Gun Metal | M | S[20] | .0054–.0177 | 5 |
| " | " | M | Pad[21] | .0082–.0154 | 5 |
| " | W.M.[18] | M[22] | ...... | .0016–.020 | 4 |
| " | Gun Metal | M[23] | Bath[21] | .0011–.0070 | 5 |
| Tin | Cast Iron | M | Dry | .176 | 1 |
| Wrt. Iron[1] | Bronze | M | Unct.[44] | .160 | 2 |
| " | Cast Iron | M | Dry | .192 | 2 |
| " | " | R | Dry | .190 | 2 |
| " | Wrt. Iron[1] | M | Dry | .138 | 2 |
| " | " | A | Dry | .4 | 1 |
| " | " | R | Dry | .152 | 1 |
| Wrt. Iron[5] | " | .. | Ord.[6] | .07–.08 | 2 |
| | | **Metal and Non-metal** | | | |
| Brass | Oak[24] | M | Dry | .616 | 2 |
| " | "[24] | R | Dry | .619 | 2 |
| Cast Iron | Elm[24] | M | Dry | .193 | 2 |
| " | Leather[25] | M | Wet | .237 | 2 |
| " | "[25] | M | Dry | .221 | 1 |
| " | "[25] | M | Wet | .284 | 1 |
| " | "[25] | R | Wet | .613 | 2 |
| " | "[26] | M | Dry | .561 | 2 |
| " | "[26] | M | Gr.[27] | .229 | 2 |
| " | "[26] | M | Wet | .359 | 2 |
| " | Lig. Vit.[28] | M | [29] | .093 | 2 |
| " | "[28] | M | Unct.[44] | .100 | 2 |
| " | Oak[24] | M | Dry | .37–.491 | 2 |
| " | "[24] | M | Unct.[44] | .106 | 2 |
| " | "[24] | M | Wet | .22 | 2 |
| " | "[24] | R | Wet | .642 | 2 |
| Metal | Hemp | .. | ...... | .279 | 9 |
| Steel | Asb.[30] | R | ...... | .27–.38 | 7 |
| " | Cork | R | Dry | .455 | 8 |
| " | " | R | Oily | .421 | 8 |
| " | " | R | Wet[33] | .564 | 8 |
| " | " | R | Wet | .694 | 8 |
| " | " comp.[32] | R | Dry | .514 | 8 |
| " | " comp.[32] | R | Gr.[27] | .55 | 8 |
| " | " comp.[32] | R | Oily | .526 | 8 |
| " | " comp.[32] | R | Wet | .894 | 8 |
| " | " comp.[32] | R | Wet | .793 | 8 |
| " | Ice | R | Dry | .266 | 1 |
| " | Stone[33] | R | ...... | .420–.491 | 10 |
| Wrt. Iron[1] | Elm[24] | M | Dry | .252 | 2 |
| " | Oak[24] | M | Dry | .605 | 2 |
| " | "[24] | R | Dry | .619 | 2 |
| " | "[24] | M | Soapy | .213 | 2 |
| " | "[24] | M | Wet | .255 | 2 |

**Table 1.—Coefficients of Friction of Various Materials*** — *Continued*

| Material | In Contact with | At Rest or in Motion † | Lubrication | Coefficient of Friction $f$ | Authority ‡ | Material | In Contact with | At Rest or in Motion † | Lubrication | Coefficient of Friction $f$ | Authority ‡ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **Non-metal and Non-metal** | | | | | | | | | | | |
| Brick | Limestone $^{34}$ | R | Dry | .674 | 2 | Masonry | Brick | R | $^{38}$ | .741 | 11 |
| " | " $^{4}$ | R | Dry | .665 | 2 | " | Brick | R | Dry | .65 | 11 |
| Cork | Cork | R | Dry | .492 | 8 | " | Clay $^{39}$ | R | Dry | .508 | 11 |
| " | Glass | R | Dry | .516 | 8 | " | Concrete | R | Dry | .785 | 10 |
| " | Pine $^{24}$ | R | ...... | .49 | 8 | " | Earth $^{37}$ | R | ...... | .65 | 10 |
| Cork comp. $^{32}$ | Cork | R | ...... | .839 | 8 | " | " $^{37}$ | R | Wet | .30 | 10 |
| " " $^{32}$ | Glass | R | ...... | .98 | 8 | Oak $^{40}$ | Limestone $^{37}$ | M | Dry | .381 | 2 |
| Earth | Earth | R | ...... | .25–1.0 | 11 | Oak | Oak $^{41}$ | M | Dry | .48 | 2 |
| " | " $^{34}$ | R | ...... | .81– | 11 | " | | M | Soapy | .164 | 2 |
| | | | | 1.11 | 1 | " | " $^{41}$ | M | $^{29}$ | .075 | 2 |
| Granite | Granite | M | Dry | .304 | 2 | " | " $^{41}$ | R | Dry | .62 | 2 |
| Hemp rope | Oak | R | Dry | .462 | 10 | " | " $^{41}$ | R | Unct. $^{44}$ | .441 | 2 |
| " | Wood $^{35}$ | R | Dry | .331 | | " | " $^{42}$ | M | Dry | .315 | 2 |
| " | " $^{36}$ | R | Dry | .50 | 10 | " | " $^{42}$ | M | Unct. $^{44}$ | .143 | 2 |
| Leather | Oak | M | Dry | .35 | 8 | " | " $^{42}$ | R | Dry | .543 | 2 |
| " $^{25}$ | " | M | Dry | .335 | 2 | " | " $^{42}$ | R | Wet | .71 | 2 |
| " $^{25}$ | " | M | Wet | .291 | 2 | " | " $^{43}$ | M | Dry | .19 | 2 |
| " $^{25}$ | " | R | Dry | .43 | 2 | " | " $^{43}$ | R | Dry | .43 | 2 |
| " $^{25}$ | " | R | Wet | .79 | 2 | Sandstone | Sandstone | M | Dry | .367 | 1 |
| " $^{26}$ | " | M | Dry | .516 | 2 | Stone | Stone | R | Dry | .63 | 1 |
| " $^{26}$ | " | R | Dry | .595 | 2 | " $^{33}$ | Wood | R | Dry | .46–.60 | 10 |
| Limestone $^{37}$ | Limestone $^{37}$ | M | Dry | .376 | 2 | Wood | Stone | R | ...... | .40 | 11 |
| " $^{37}$ | " $^{37}$ | R | Dry | .702 | 2 | Wood | Wood | R | Soapy | .358 | 12 |
| " $^{4}$ | " $^{4}$ | R | Dry | .636–.744 | 2 | Woolen Cloth | Woolen Cloth | M | Dry | .439 | 1 |

* Compiled from data assembled by J. F. Spiegel for Kingsbury Machine Works.

† A = abrasion; M = motion; R = rest.

‡ Authorities: 1. Rennie, Experiments on Friction, *Phil. Trans.*, Royal Soc., London, Vol. 119, 1829. 2. Morin, Nouvelles Expériences sur le Frottement; faites a Metz, 1831 Annales des Mines 3rd ser. vol. 4; faites a Metz 1832 Annales des Mines 3rd. ser. vol. 6; faites a Metz 1833 Annales des Mines 3rd. ser. vol. 10; or Paris, Bachelier 1832, 1834, 1835. Also Nouvelles Expériences sur l'Adherence des Pierres et des Briques, le Frottement des Axes de Rotation, la Variation du Tension des Courroies ou Cordes sans Fin, Paris, Carillan Goeury 1838; Summary of the results of Coulomb, Rennie and Morin may be found in the paper of Brix, "Über die Reibung," Verhandl. d. Vereins z. Forderung des Gewerbefleisses in Preussen, vol. 16, 1837. 3. Kingsbury, Experiments on the Friction of Screws, *Trans.* A.S.M.E., vol. xvii, 1896. 4. Stribeck, "Eigenschaften der Gelit- und Rollenlager, *Zeit.* V.D.I., 1902. 5. Tower, First Reports on Friction Experiments, *Proc.*, I.M.E., March 1891. 6. Lasche, Die Reibungsverhältnisse von Lagern mit hoher Umfangsgeschwindigkeit, *Zeit.* V.D.I., 1902. 7. Chase, Practice and Theory in Clutch Design, *Trans.* S.A.E., 1921, Pt. II, 8. Wenrich, University of Pittsburgh. 9. Unwin, Elements of Machine Design. 10. Hütte. Ingenieurs Taschenbuch, 26th Edition. 11. Rankine, Applied Mechanics. 12. Lesley, Inst. Soc. Nav. Arch. & Mar. Eng., 1904.

[1] Wrought Iron. [2] Slow. [3] Copious. [4] Soft. [5] Journal. [6] Ordinary. [7] Thrust bearing. [8] Low practical value. [9] 50 r.p.m. [10] Oil viscosity, 150 Saybolt Universal sec. [11] 500 r.p.m. [12] 2000 r.p.m. [13] 4000 r.p.m. [14] Shell. [15] Velocity, 13.2 ft. per sec.; pressure, 14.2–356 lb. per sq. in. [16] Collar bearing. [17] 50–145 r.p.m.; pressure, 75–90 lb. per sq. in. [18] W.M. = white metal. [19] Velocity, 3–66 ft. per sec.; pressure, 14.2–214 lb. per sq. in. [20] Siphon lubrication; rape oil. [22] Velocity, 13.2 ft. per sec.; pressure, 14.2–712 lb. per sq. in. [23] Velocity, 1.75–7.85 ft. per sec.; pressure, 100–520 lb. per sq. in. [24] On lengthwise grain. [25] On edge. [26] On flat. [27] Greasy. [28] Lignum Vitae. [29] Tallow or lard. [30] Asbestos fabric. [31] Water-soaked. [32] Cork composition. [33] Stone, gravel. [34] Including shingle and gravel. [35] Polished. [36] Rough. [37] Hard. [38] Damp mortar. [39] Dry. [40] End grain. [41] Grains parallel. [42] Grains crossed. [43] End grain or parallel grain. [44] Unctuous.

**FRICTION OF REST AND OF MOTION.**—The force required to start a body sliding is called the friction of rest, and the force required to continue its sliding after having started is called the friction of motion. Except at very low velocities, sliding friction is less than the friction of rest. If $P$ is the force with which two bodies, sliding on each other, are pressed together, their motion is resisted by a force $R$ acting in a direction opposite to the direction of motion, which is called the frictional resistance, and $R = fP$, where $f$ is the coefficient of sliding friction.

**Minimum Starting Friction.**—Starting friction, neglecting acceleration, can be reduced to zero if the surfaces are separated by a liquid of low viscosity, as shown in Fig. 2. Liquid is pumped into the recess at the center of the disc under such pressure that when the flow reaches the edge of the plate the load is supported entirely by a fluid film. This method is used for starting oil film bearings, the lubricating oil being the pressure liquid.

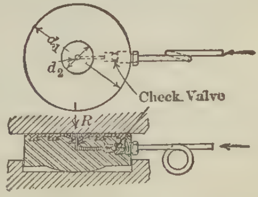

FIG. 2

**Starting Friction of Newly-finished Bearing Surfaces.**—The following summarizes recent (1937) tests of newly finished surfaces to determine the influence of method of finishing on the coefficient of starting friction in the absence of vibration. The tests were made with a babbitt-faced shoe, of 60.7 sq. in. area, scraped to a surface plate, opposed to a close-grained cast-iron surface, finish turned and lapped with fairly coarse carborundum. The surfaces were immersed in light oil, and subjected to a load of over 300 lb. per sq. in. for several hours. The coefficient of starting friction was as high as 0.66. It was reduced to as low as 0.21 by more refined finishes. The lowest coefficients were found when the cast-iron surfaces had been scrubbed by rotation, pressed by a leather-covered block, using lubricating oil impregnated with a very fine abrasive. The direction of motion during test (so long as it is concentric), seems to have little influence on the coefficient of friction.

Some of the values of starting coefficient of friction determined were: Lard oil, load applied for 2 hr., $f = 0.18$; castor oil, load applied for 113 hr., $f = 0.25$; light mineral oil, viscosity 150 Saybolt universal seconds, load applied 14 to 18 hr., $f = 0.30$. Kerosene gave higher values than the mineral oil. The conclusions drawn from the test were: In the absence of vibration, the starting friction coefficient is a function of the newness of the surfaces, of the methods of finishing employed, of the time which the surfaces have been pressed together, of the pressure acting on them, of the nature and viscosity of the lubricating oil in which the surfaces are immersed, and of the area of the smaller of the two faces. When the surfaces are loaded to around 300 lb. per sq. in. and have an area of 60 sq. in., the coefficient of static friction in the absence of vibration may be expected to be below 0.30 for new faces, and below 0.20 for those in service. The influence of electrical vibration is believed to reduce these values 20% or more.

**Influence of Surface Finish on Starting Friction and Friction at Low Speed.**—Howarth's experiments on a low speed and other tests of Kingsbury thrust bearings (*Trans.* A.S.M.E., xli, p. 685, 1919) show the variation, with speed and load, of friction of motion in a small thrust bearing of a mean diameter of 9.375 in., having four babbitt-faced shoes, and a revolving cast-iron runner. The total shoe area was varied from 71.5 sq. in. to 10 sq. in. in order to test pressures ranging from 140 to 1000 lb. per sq. in. In a typical test, the load was 600 lb. per sq. in. Machine oil, of 5.345 Saybolt universal seconds viscosity at the test temperature, was used as lubricant, the surfaces being always submerged. The test was started at 16 r.p.m., speed being reduced gradually to 1/3 r.p.m. The friction coefficient increased gradually, but irregularly, from about 0.0015 at 16 r.p.m. to about 0.132 at 1/3 r.p.m. The speed then was gradually increased without changing the load. The friction coefficients determined for the increasing speeds were much lower than that for decreasing speeds. This was explained by the improvement of bearing surfaces due to the long running at low speed. During these tests friction coefficients for starting from rest were found to vary from 0.15 to 0.20. These values, therefore, may be considered satisfactory for bearing surfaces that have been in operation for some time under favorable conditions in which a number of starts and stops have gradually brought the surfaces to a condition favorable to such starting when immersed in an oil bath.

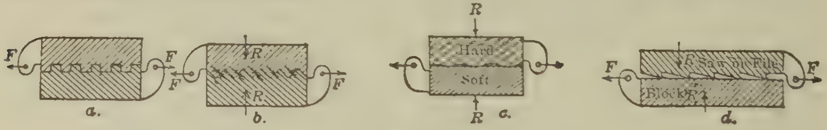

FIG. 3

**Maximum Friction.**—When two metals are sweated or brazed together, the force required to separate them transversely at the joint is less a function of the pressure than of the transverse shearing strength of the metal bond. This might be a combination of the breaking of the bond between the soft and hard material and the actual shearing of the soft material on itself. The force required for such separation cannot consistently be expressed by means of a coefficient of friction.

INTERLOCKING FRICTION.—If two surfaces interlock, as in a square thread screw, Fig. 3*a*, the force required to slide one on the other depends more on strength of the materials and proportion of interlocking than on the load acting between them. If the surfaces are free to move vertically against a load, Fig. 3*b*, the force required to slide them laterally would be a combination of sliding friction and lifting of the load. When an irregular hard surface presses against a soft one, Fig. 3*c*, the resistance offered to sliding one over the other will depend on the normal load, on the sliding friction, on resistance to deformation of the soft surface, and on the strength of the softer surface as it might be broken down because of interlocking. Fig. 3*d* illustrates abrading frictional resistance to sliding of one surface over another, as a saw or file. The greater the normal pressure on the saw or file, the more will it cut into the lower block and the greater will be the resistance offered to transverse motion.

INFLUENCE OF VIBRATION.—Tests to determine starting friction show coefficients of friction to be much higher in the absence of vibration than in its presence, and that friction decreases as the disturbance by vibration increases. Vibration similarly reduces the friction of motion so long as metallic contact is present in any considerable degree. With perfect lubrication, bearing surfaces are completely separated by lubricating films, and vibration does not appear to change the coefficient of friction of motion.

**ROLLING FRICTION** is the force required to roll a cylindrical or spherical body on a plane or on a curved surface. It depends on the nature of the surfaces and on the force with which they are pressed together, but is different from ordinary, or sliding, friction.

Rolling friction is a consequence of the irregularities of form and the roughness of surface of bodies rolling one over the other. Its laws are not yet definitely established in consequence of the uncertainty which exists in experiment as to how much of the resistance is due to roughness of surface, how much to original and permanent irregularity of form, and how much to distortion under the load.

**Coefficients of Rolling Friction.**—Let $R$ = resistance applied at the circumference of the wheel, $W$ = total weight, $r$ = radius of the wheel, and $f$ = a coefficient. Then $R = fW \div r$. The value of $f$ is very variable. Coulomb gives 0.06 for wood, 0.005 for metal, where $W$ is in pounds and $r$ in feet. Tredgold made the value of $f$ for iron on iron 0.002. For wagons on soft soil Morin found $f$ = 0.065, and on hard smooth roads 0.002.

A Committee of the Society of Arts (Clark, R. T. D.) reported a loaded omnibus to exhibit a resistance on various loads as below:

| Pavement | Speed per hr. | Coefficient | Resistance |
|---|---|---|---|
| Granite | 2.87 miles | 0.007 | 17.41 per ton |
| Asphalt | 3.56 " | 0.0121 | 27.14 " |
| Wood | 3.34 " | 0.0185 | 41.60 " |
| Macadam, graveled | 3.45 " | 0.0199 | 44.48 " |
| Macadam, granite, new | 3.51 " | 0.0451 | 101.09 " |

Thurston gives the value of $f$ for ordinary railroads, 0.003; well-laid railroad track, 0.002; best possible railroad track, 0.001. The friction of pneumatic tires on roads is discussed in Kent's Mechanical Engineers' Handbook—Power, p. 14–55, forming Vol. 2 of this series.

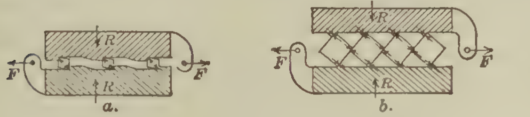

FIG. 4

**GRANULAR FRICTION.**—The friction required to move one surface over another when granular material is placed between them probably is a combination of several kinds of interaction. The simplest condition is that in which the granular particles turn over and over without interference (Fig. 4*a*). Fig. 4*b* indicates how the granular particles may become interlocked so they will abrade each other and may cut both surfaces when motion occurs. Fig. 4*c* represents granular particles embedded in the softer surface and acting as cutters when moved on the harder surface.

**SKIDDING FRICTION** results from slipping of rubber tires on a pavement. The friction depends on the type of tire surface and conditions of the pavement. With rough pavement and smooth tires, high spots on the pavement will penetrate any film that is formed and project into the rubber, thus diminishing tendency to skid. With a smooth pavement and tires, the film of water, clay or pulpy material found on roads, will completely separate the surfaces. The tire then will slide freely in any direction. See Fig. 5*a*.

In the non-skid tire, Fig. 5*b*, the rubber surface is broken up into small faces, with deep grooves between. The theory of lubrication shows that the thickness of film formed when sliding occurs decreases as the supporting area for a given load decreases, and the films will more quickly leak from between the surfaces and permit them to come together. On rough roads the difference between the action of smooth and non-skid tires is small. Even non-skid tire surfaces are apt to slip on pulpy material or on clay, since such materials build up much thicker films than are possible with clear water.

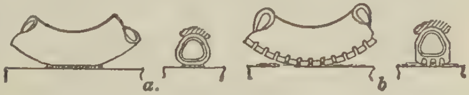

FIG. 5

**FLUID FRICTION** includes the frictional resistance to motion of: Fluids in containers, as the flow of water or gas in pipes; of solid bodies completely separated by a film of fluid, as a perfectly lubricated bearing; of solid partially or wholly immersed in a fluid, as a boat moving through water.

**LAWS OF FRICTION.**—The laws of friction, first formulated by Morin about 1830, have been modified from time to time by the work of later experimenters. The following paragraphs form a brief statement of the generally-accepted laws, based on present-day practice.

**Laws of Friction for Unlubricated Surfaces.**—1. Friction varies approximately as the normal pressure with which the rubbing surfaces are pressed together.

2. Friction is approximately independent of the area of the surfaces, but is slightly greater for small surfaces than for large surfaces.

3. Friction decreases with increase of velocity, except at very low velocity, and with soft surfaces.

**Laws of Friction for Lubricated Surfaces.**—*Perfect Lubrication, i.e.*, surfaces completely separated by a film of lubricant:

4. The coefficient of friction is independent of the materials of the surfaces.

5. The coefficient of friction varies directly with the viscosity of the lubricant, which varies inversely with temperature of the lubricant.

6. The coefficient of friction varies inversely as the unit pressure, and directly as the velocity.

7. The coefficient of friction varies inversely as the mean film thickness of the lubricating medium.

8. Mean film thickness varies directly with velocity and inversely as the temperature and unit pressure.

For a full discussion of the friction of perfectly lubricated bearings, see p. 13–44.

*Imperfect Lubrication, i.e.*, surfaces partially separated by a film of lubricant, may range from almost complete separation of the surfaces to almost complete contact.

9. The coefficient of friction increases with increase of pressure between surfaces.

10. The coefficient of friction decreases with increase of relative velocity between the surfaces.

**Laws of Fluid Friction.**—Thurston states that for all fluids, whether liquid or gaseous, the resistance is: 1. Independent of the pressure between the masses in contact; 2. Directly proportional to the area of rubbing-surface; 3. Proportional to the square of the relative velocity at moderate and high speeds, and to the velocity nearly at low speeds; 4. Independent of the nature of the surfaces of the solid against which the stream may flow, but dependent to some extent upon their degree of roughness; 5. Proportional to the density of the fluid, and related in some way to its viscosity.

Fluid friction, as related to the flow of liquids in pipes, is discussed in Kent's Mechanical Engineers' Handbook—Power, p. 4–56, forming Vol. 2 of this series.

**LOW FRICTION COEFFICIENTS.**—Friction coefficients frequently are as low as 0.001 in journal and thrust bearings in operation with perfect lubrication. Much lower values have been obtained experimentally. Fig. 6 shows coefficients obtained by S. J. Needs in the laboratory of Albert S. Kingsbury, using a 120-deg. fitted bearing 4 in. wide,

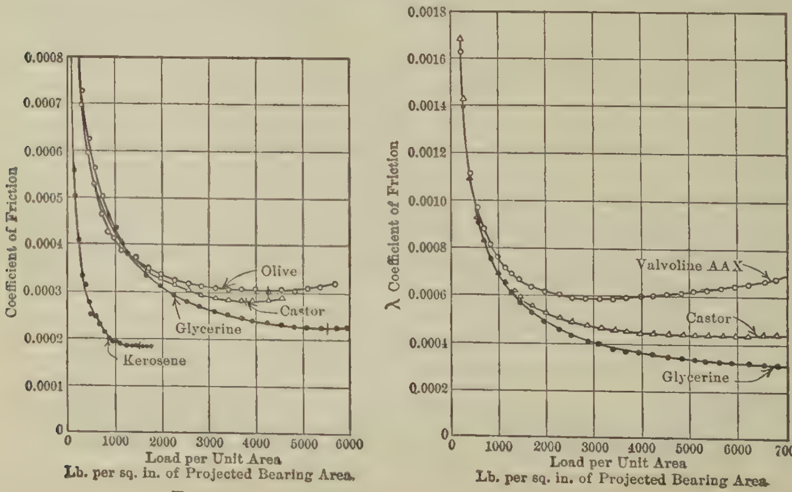

Fig. 6 Fig. 7

Variation of Viscosity with Pressure

4 in. diameter. The value 0.000183 is the lowest known value obtained in a bearing lubricated by oil.

Fig. 7 indicates that the viscosity of the lubricant may increase considerably with pressure. The three oils had identical viscosities at the beginning of the test, were subjected to the same series of loads, and run at the same speed of 11.06 r.p.m. The Valvoline AAX oil showed the greatest increase of viscosity with load. The least increase was shown by glycerine. Castor oil viscosity increased an intermediate amount. In these tests the bearing always operated under fluid film conditions, and the bearing surfaces were completely separated.

## 2. FRICTION OF MACHINE ELEMENTS

**FRICTION OF BEARINGS.**—For a detailed discussion of the friction of bearings, see pp. 13–03 to 13–58. The following data, from various sources, are for general guidance only. For design, in cases where friction is an important factor, reference should be made to the pages above cited.

**FRICTION OF JOURNALS.**—The friction of journals usually is expressed in terms of units of projected area, that is, $d \times l$, where $d$ = diameter and $l$ = length of journal. Unwin states that the friction of journals must lie between $\pi fP/2$ and $4fP/\pi$, where $f$ = coefficient of friction and $P$ = load on journal. For journals in ordinary condition the value of $fP$ usually is sufficiently accurate. Values of $f$ vary according to different authorities. Unwin gives the following:

| Authority. | $f$ | Conditions. |
|---|---|---|
| Morin | 0.05 to 0.07 | Good journals, well lubricated |
| Rühlmann & Waltjen | 0.02 to 0.03 | Wrought-iron on gun metal |
| Kirchweger | 0.009 to 0.01 | " " " white metal* |
| " | 0.014 | " " " gun metal* |
| Bokelberg & Welkner | 0.0028 | Small loads, low velocities† |
| " " | 0.013 | Heavy loads, high velocities† |

The work lost in friction, in ft.-lb. per minute is $W = fP\pi n(d/12)$, where $n$ = rev. per minute, $d$ = diam. of journal, in., other notation as above.

The friction of a journal is influenced by the character and quantity of lubrication. Below are the results of experiments on lubricated journals.

**Experiments on Friction of a Journal Lubricated by an Oil-bath** (reported by the Committee on Friction, *Proc.* Inst. M. E., Nov., 1883) show that the absolute friction, that is, the absolute tangential force per square inch of bearing, required to resist the tendency of the brass to go round with the journal, is nearly a constant under all loads, within ordinary working limits. Most certainly it does not increase in direct proportion to the load, as it should do according to the ordinary theory of solid friction. The results of these experiments seem to show that the friction of a perfectly lubricated journal follows the laws of liquid friction much more closely than those of solid friction. They show that under these circumstances the friction is nearly independent of the pressure per square inch, and that it increases with the velocity, though at a rate not nearly so rapid as the square of the velocity. See Table 2.

The experiments indicate a great diminution in the friction as the temperature rises. With lard-oil, at a speed of 450 r.p.m., the coefficient of friction at 120° F. is only one-third of the coefficient at 60° F. See Table 3.

In the experiments, the journal was of steel, 4 in. diameter and 6 in. long. A gun-metal brass, embracing somewhat less than half the circumference of the journal, rested on its upper side, on which the load was applied. When the bottom of the journal was immersed in oil, and the oil therefore carried under the brass by rotation of the journal, the greatest load carried with rape-oil was 573 lb. per sq. in., and with mineral oil 625 lb.

**Table 2.—Variation of Coefficient of Friction with Speed and Pressure**

| Feet per Minute | 209 | 262 | 314 | 366 | 419 | 471 | Nominal Load per sq. in. |
|---|---|---|---|---|---|---|---|
| Coefficient of friction | 0.0010 | 0.0012 | 0.0013 | 0.0014 | 0.0015 | 0.0017 | 520 lb. |
| " " " | .0013 | .0014 | .0015 | .0017 | .0018 | .002 | 468 lb. |
| " " " | .0014 | .0015 | .0017 | .0019 | .0021 | .0024 | 415 lb. |

**Table 3.—Variation of Coefficient of Friction with Temperature**

| Temperature, deg. F. | 110 | 100 | 90 | 80 | 70 | 60 |
|---|---|---|---|---|---|---|
| Observed | 0.0044 | 0.0051 | 0.006 | 0.0073 | 0.0092 | 0.0119 |
| Calculated | 0.00451 | 0.00518 | 0.00608 | 0.00733 | 0.00964 | 0.01252 |

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* Railway journals well lubricated.

† In these experiments $f$ was less for gun metal than for white metal.

**Table 4.—Coëfficients of Friction of Journal with Oil-bath**

| Lubricant in Bath | Nominal Load, lb. per sq. in. | | | | | | |
|---|---|---|---|---|---|---|---|
| | 625 | 520 | 415 | 310 | 205 | 153 | 100 |
| | Coefficient of Friction | | | | | | |
| Lard oil: 157 ft. per min. | ........ | 0.0009 | 0.0012 | 0.0014 | 0.0020 | 0.0027 | 0.0042 |
| " " 471 " " | ........ | .0017 | .0021 | .0029 | .0042 | .0052 | .009 |
| Mineral grease: 157 ft. per min. | 0.001 | .0014 | .0016 | .0022 | .0034 | .0038 | .0076 |
| " " 471 " " | 0.002 | .0022 | .0027 | .004 | .0066 | .0083 | .0151 |
| Sperm-oil: 157 ft. per min. | ........ | seized | .0015 | .0011 | .0016 | .0019 | .003 |
| " " 471 " " | ........ | ....... | .0021 | .0019 | .0027 | .0037 | .0064 |
| Rape-oil: 157 ft. per min. | (573 lb.)<br>.001 | .001 | .0009 | .0008 | .0014 | .002 | .004 |
| " " 471 " " | ........ | .0015 | .0016 | .0016 | .0024 | .004 | .007 |
| Mineral oil: 157 ft. per min. | .0013 | .0012 | .0012 | .0014 | .0021 | ....... | .004 |
| " " 471 " " | ........ | .0018 | .002 | .0024 | .0035 | ....... | .007 |
| Rape-oil fed by siphon lubricator: 157 ft. per min. | ........ | ....... | ....... | .0056 | .0098 | ....... | .0125 |
| 314 " " | ........ | ....... | ....... | .0068 | .0077 | ....... | .0152 |
| Rape-oil pad under journal: 157 ft. per min. | ........ | ....... | ....... | .0099 | .0105 | ....... | .0099 |
| 314 " " | ........ | ....... | ....... | .0099 | .0078 | ....... | .0133 |

In experiments with ordinary lubrication, the oil being fed in at the center of the top of the brass, and a distributing groove being cut in the brass parallel to the axis of the journal, the bearing would not run cool even with a load of but 100 lb. per sq. in., the oil being pressed out from the bearing-surface and through the oil-hole, instead of being carried in. Introducing the oil at the sides through two parallel grooves, gave apparently satisfactory lubrication, but the bearing seized at a pressure of 380 lb. per sq. in.

When the oil was introduced through two oil-holes, one near each end of the brass, and each connected with a curved groove, the brass refused to take its oil or run cool, and seized with a load of only 200 lb. per sq. in.

With an oil-pad under the journal feeding rape-oil, the bearing fairly carried 551 lb. Mr. Tower's conclusion from these experiments is that the friction depends on the quantity and uniformity of distribution of the oil, and may be anything between the oil-bath results and seizing, according to the perfection or imperfection of the lubrication. The lubrication may be very small, giving a coefficient of $1/100$; but it appeared as though it could not be diminished and the friction increased much beyond this point without imminent risk of heating and seizing. The oil-bath probably represents the most perfect lubrication possible, and the limit beyond which friction cannot be reduced by lubrication. The experiments show that with speeds of from 100 to 200 ft. per minute, by properly proportioning the bearing-surface to the load, it is possible to reduce the coefficient of friction to as low as 0.001. A coefficient of 0.00066 is easily attainable, and probably is frequently attained, in ordinary engine-bearings in which the direction of the force is rapidly alternating and the oil has an opportunity to get between the surfaces, while the duration of the force in one direction is not sufficient to allow time for the oil film to be squeezed out.

Observations on the behavior of the apparatus gave reason to believe that with perfect lubrication the speed of minimum friction was from 100 to 150 ft. per minute, and that this speed of minimum friction tends to be higher with an increase of load, and also with less perfect lubrication. By the speed of minimum friction is meant that speed in approaching which from rest the friction diminishes, and above which the friction increases. Table 4 gives coefficients of friction as determined in these experiments.

Comparative friction of different lubricants under same circumstances, temperature 90° F., oil-bath: sperm-oil, 100; rape-oil, 106; mineral oil, 129; lard, 135; olive oil, 135; mineral grease, 217.

**Value of Anti-friction Metals.**—According to Prof. J. E. Denton the various white metals available for lining brasses do not afford coefficients of friction lower than can be obtained with bare brass, but they are less liable to "overheating," because of the superiority of such material over bronze in ability to permit of abrasion or crushing, without excessive increase of friction. Thurston (Friction and Lost Work) says that gun-bronze, Babbitt, and other soft white alloys have substantially the same friction; in other words, the friction is determined by the nature of the unguent and not by that of the rubbing-surfaces, when the latter are in good order. The soft metals run at higher temperatures than the bronze. This, however, does not necessarily indicate a serious defect, but simply deficient conductivity. The value of the white alloys for bearings lies mainly in their ready reduction to a smooth surface after any local or general injury by alteration of either surface of form. See Bearing metals, pp. 4–62 to 4–67.

**Ball Bearings.**—The friction of ball bearings varies with the design and construction of the bearing. In general, it decreases with increase of load, and is independent of speed and temperature. Stribeck found the coefficient of friction in one type of bearing at 65 r.p.m. to vary from $f = 0.0033$ with a load of 838 lb. to $f = 0.0015$ with a load of 6614 lb. At 780 r.p.m. the same bearings showed $f = 0.0037$ and $f = 0.0013$, respectively, at 838 lb. and 6614 lb. loads.

Goodman gives the following ranges for $f$ in various types of ball bearings: Cylindrical, 2-point bearing, $f = 0.0012$ to $0.0018$; thrust, 2-point bearing, $f = 0.0008$ to $0.0012$; thrust, 3-point bearing, $f = 0.0018$ (average); thrust, 4-point bearing, $f = 0.0055$.

**Roller Bearings.**—The coefficient of friction of roller bearings decreases with increase of load and with increase of temperature. It seems to be independent of the speed. Goodman gives values at loads of 2000 and 10,000 lb. respectively, of $f = 0.0131$ to $0.0072$ for 40 r.p.m., and of $f = 0.0053$ to $0.0024$ for 400 r.p.m.

**BELTING.**—For data on the coefficient of friction of belting, see pp. 24–15 and 24–36.

**GEARING.**—The friction of gearing depends on the type, the material of the gears, the quality of workmanship, quality of lubrication, and the form of the teeth. Values given below should be used with caution, as they are applicable only under conditions identical with those under which the determination was made.

**Spur Gears.**—Wilfred Lewis gives the following as the friction losses $F$ of various types of spur gears, in which $p$ = circular pitch: 20-deg. stub gear, addendum of $0.24\,p$, $F < 1\%$; $22\,{}^1/_2$-deg. gear, addendum of $0.28\,p$, $F = 1\%$ approx.; $14\,{}^1/_2$-deg. gear, addendum of $0.32\,p$, $F = 1.3\%$ approx. Other investigators report friction losses of cut gears in the neighborhood of 1%. The length of addendum has greater effect on the friction than has the angle of obliquity.

**Bevel Gears.**—Friction losses have been reported by various investigators, ranging from 1% under exceptionally good conditions of workmanship and lubrication to 8.6% under less favorable conditions.

**Helical Gears.**—Friction losses in helical gears are composed of the tooth friction losses, which correspond to those of spur gears, and the losses resulting from the transverse sliding of the teeth on each other. The ratio $F_s$ of lost work, due to this latter cause, to the useful work may be expressed as

$$F_s = \{\sin \alpha(\beta + \theta)/\sin \beta \sin (\alpha + \theta)\} - 1$$

in which $\alpha$ and $\beta$ are, respectively, the helical angles of the gears and $\theta$ is the angle of repose, determined from the coefficient of friction, *i.e.*, $f = \tan \theta$.

**Worm Gearing.**—Let $\beta$ = angle of inclination of worm thread to its axis, $P$ = force acting on pitch line of worm in the plane of rotation, $R$ = resistance acting at pitch line of worm-wheel in its plane of rotation, $\theta$ = angle of repose of metal of worm on metal of wheel, $f$ = coefficient of friction = $\tan \theta$, $r_1$ = radius of worm = $1.5\,p$ to $3\,p$ usually, $r_2$ = radius of wheel, $p$ = pitch of teeth of wheel, and $p/n$ = axial pitch of worm, $n$ being the number of threads on the worm. Then

$$\frac{P}{R} = \frac{1 + f \tan \beta}{\tan \beta - f} = \cot (\beta - \theta)$$

$$\text{Efficiency} = \frac{\cot \beta}{\cot (\beta - \theta)} = \left(1 - f\frac{np}{2\pi r_1}\right) \div \left(1 + f\frac{r_1}{np}\right).$$

Taking $f = 0.15$, Efficiency $= np \div (np + r_1)$, approximately.

**Friction Gearing.**—See p. 14–03.

**FRICTION OF HYDRAULIC PLUNGER PACKING.**—Hütte (15th edition, vol. 1, p. 202) says: "For stuffing-boxes with hemp, cotton or leather packing, with water pressures between 1 and 50 atmospheres, the frictional loss depends on the water pressure, the circumference of the packed surface, and a coefficient $\mu$, which is constant for this range of pressure. The loss is independent of the depth of stuffing-box or leather ring, and is given by the formula $F = Kpd$, where $F$ = total frictional loss, lb., $p$ = pressure, lb. per sq. in., $d$ = diameter of plunger, in. $K$ is a coefficient which depends on the kind and condition of the packing, and is given as follows for various cases:

For cotton or hemp, loose or braided, dipped in hot tallow, plungers smooth, glands not pulled down too tight, packing therefore retaining its elasticity, dimensions such as usually occur, $K = 0.072$.

Same conditions, after packing is some months old, $K = 0.132$.

Materials the same, but with hard packing, unfavorable conditions, etc., $K$ = as much as 0.299.

Leather packing, soft leather, well made, etc., $K = 0.036$ to $0.084$.

Hard, stiffly tanned leather, $K = 0.12$ to $0.156$.

Unfavorable conditions, rough plungers, gritty water, etc., $K$ = up to 0.239.

Weisbach-Hermann, in Mechanics of Hoisting Machinery, gives a formula which when translated into the same notation as the one in Hütte is

$$F = 0.0312\,pd \text{ to } 0.0767\,pd.$$

Since the total pressure on a plunger is ${}^1/_4\,\pi d^2\,p$, the ratio of the loss of pressure to the total pressure is $K\,p\,d\,/\,{}^1/_4\,\pi\,d^2\,p$, or using the extreme values of $K$, 0.0312 and 0.299, the ratio ranges from $0.04/d$ to $0.38/d$, or from 4 to 38% divided by the diameter in inches.

Walter Ferris (*Am. Mach.*, Feb. 3, 1898) derives from the formula given above the following formula for the pressure produced by a hemp-packed hydraulic intensifier made with two plungers of different diameters:

$$p_2 = p_1\{(A - KD) \div (a + kd)\},$$

where $p_2$ = pressure per sq. in. produced by the intensifier, $p_1$ = initial pressure, $A$ = area and $D$ = diam. of the larger plunger, $a$ = area and $d$ = diam. of the smaller plunger, and $K$ an experimental coefficient. He gives the following results of tests of an intensifier with a small plunger 8 in. diam. and two large plungers, 14 1/4 and 17 3/4 in., either one of which could be used as desired.

| | | | | |
|---|---|---|---|---|
| Diam. of large plunger, in. | 14 1/4 | 14 1/4 | 17 3/4 | 17 3/4 |
| Initial pressure, lb. per sq. in. | 285 | 475 | 335 | 350 |
| Intensified pressure, lb. per sq. in. | 750 | 1450 | 1450 | 1510 |
| Intensified if there were no friction | 905 | 1505 | 1650 | 1725 |
| Intensified calculated by formula* | 806 | 1433 | 1572 | 1643 |
| Efficiency of machine | 0.83 | 0.965 | 0.88 | 0.875 |

* Assuming $K$ = 0.2. Efficiency calculated by formula in each case = 0.953.

**FRICTION OF KEYS AND COTTERS.**—In a taper key, the angle between the tapered sides of which is $\alpha$, the force required to drive the key home is

$$F = P\{\tan(\theta + \alpha) + \tan\theta\},$$

where $P$ = pressure on the key normal to one of the tapered sides, and $\theta$ = angle of repose of the surfaces in contact.

The force required to drive the key out is

$$F_1 = P\{\tan(\theta - \alpha) + \tan\theta\}.$$

The limiting value of $\alpha$ is $2\theta$, for if $\alpha > 2\theta$ the key will not remain in position. If $f$ = 0.08, $\theta$ = 4.5 deg. The greatest angle of taper, therefore, is 9 deg. or a taper of approximately 1 in 7. Similarly for cotters, where $\alpha_1$, and $\alpha_2$, respectively, are the angles made by the two edges of the cotter with a center line normal to the line of pressure.

$$F = P\{\tan(\theta + \alpha_1) + \tan(\theta + \alpha_2)\},$$

and

$$F_1 = P\{\tan(\theta - \alpha_1) + \tan(\theta - \alpha_2)\}.$$

As before, the cotter will slip out if $(\alpha_1 + \alpha_2) > 2\theta$, which gives a limiting value of $(\alpha_1 + \alpha_2)$ = 9 deg. when $f$ = 0.08.

**FRICTION OF ROPE SHEAVES.**—Unwin gives $f$ = 0.28 for rope on a metal pulley. With ropes in grooves whose sides make an angle of 45 deg. the pressure between the rope and the sides of the groove is greater than the force pressing the rope into the groove, the ratio being $(\operatorname{cosec}\theta/2):1$, where $\theta$ = angle of inclination of groove. The actual coefficient of friction of a rope in a sheave groove then is

$$f = 0.28 \times \operatorname{cosec} {}^1/_2\,\theta,$$

giving for a 45-deg. groove $f$ = 0.7.

**FRICTION OF SCREWS.**—See also p. 8–11. The coefficient of friction of square thread screws 1.425 in., outside diam., 0.33 in. pitch was determined by Kingsbury to range from $f$ = 0.009 to 0.25 with lard oil lubrication, $f$ = 0.11 to 0.19 with heavy machinery oil, and $f$ = 0.03 to 0.15 with heavy oil and graphite. Variation in axial pressure apparently had no effect on values of $f$, nor did $f$ change appreciably with variation of the metals in contact.

**FRICTION OF STEAM ENGINES.**—For data on the friction of steam engines, see Kent's Mechanical Engineers' Handbook—Power, p. 7–28, forming Vol. 2 of this series.

**FRICTION OF V GUIDES.**—Let $P$ be the force pressing a V-shaped block into V-guides, the angle between the faces of the guides being $\beta$. The frictional resistance $R$ to the movement of the block along the guides will be

$$R = fP/\sin({}^1/_2\,\beta),$$

where $f$ = coefficient of friction between the metals of the guides and blocks.

# Section 9

# FASTENINGS

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## SCREWS AND SCREW FASTENINGS

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# FASTENINGS

## SCREWS AND SCREW FASTENINGS

**CLASSIFICATION.**—Screws and bolts may be classified, according to use, into two general classes: A. Screws for transmitting power. B. Screw fastenings. Class B may be further subdivided into: 1. Bolts; 2. Studs; 3. Machine screws; 4. Cap-screws or Tap-bolts; 5. Set-screws; 6. T-head bolts; 7. Coupling bolts; 8. Hook bolts; 9. Automobile bolts; 10. Carriage bolts; 11. Stove bolts; 12. Expansion bolts; 13. Foundation bolts; 14. Eye-bolts; 15. Stay-bolts; 16. Patch bolts; 17. Wood screws; 18. Lag screws and other special screws and bolts.

### 1. FORMS OF THREAD

The form of thread depends on the service required and on the engineering standards of the country in which the thread is used. In general, the full V-thread and the modified V-thread are best adapted for fastenings. These combine strength and cheapness; the inefficiency of the V-thread in transmitting power lessens the liability of unscrewing.

**POWER-TRANSMISSION THREADS.**—Three forms of threads are adapted to the transmission of power: *a.* The square thread, Fig. 1, has the highest efficiency in transmitting power. It is expensive to make and is not easily compensated for wear. *b.* The so-called Acme thread, Fig. 2, is not so expensive to cut as the square thread, and wear can be compensated by means of a longitudinally split nut. Its efficiency in transmitting power is less than that of the square thread. *c.* The buttress thread, Fig. 3, is used for the transmission of power in one direction only. The driving face is perpendicular to the axis of the screw; the back face makes an angle, usually 45 deg., with the axis. This combination gives the efficiency of the square thread and the strength of the V-thread.

Besides the threads referred to, the standard pipe thread may be considered a form of fastening. See **Kent's Mechanical Engineers' Handbook—Power, p. 5–40, forming Vol. 2 of this series.**

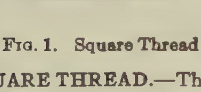


**Fig. 1. Square Thread**

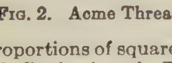


**Fig. 2. Acme Thread**

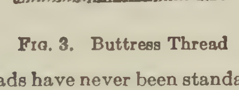


**Fig. 3. Buttress Thread**

**SQUARE THREAD.**—The proportions of square threads have never been standardized, but the practice of Wm. Sellers & Co. is given in Table 1.

**THE ACME SCREW THREAD.**—The Acme thread is an adaptation of the commonly-used style of worm thread and is intended to take the place of the square thread. It is a little shallower than the worm thread, but the same depth as the square thread, and much

**Table 1.—Proportions of Square Threads**

William Sellers & Co., Philadelphia

| Size, in. | Threads per in. | Root Diam., in. | Size, in. | Threads per in. | Root Diam., in. | Size, in. | Threads per in. | Root Diam., in. |
|---|---|---|---|---|---|---|---|---|
| 1/4 | 10 | 0.1625 | 1 1/4 | 3 1/2 | 1.0 | 3 | 1 3/4 | 2.5 |
| 3/8 | 8 | .2658 | 1 1/2 | 3 | 1.2084 | 3 1/4 | 1 3/4 | 2.75 |
| 1/2 | 6 1/2 | .3656 | 1 3/4 | 2 1/2 | 1.4 | 3 1/2 | 1 5/8 | 2.962 |
| 5/8 | 5 1/2 | .466 | 2 | 2 1/4 | 1.612 | 3 3/4 | 1 1/2 | 3.168 |
| 3/4 | 5 | .575 | 2 1/4 | 2 1/4 | 1.862 | 4 | 1 1/2 | 3.418 |
| 7/8 | 4 1/2 | .6806 | 2 1/2 | 2 | 2.0626 | | | |
| 1 | 4 | .7813 | 2 3/4 | 2 | 2.3126 | | | |

**Table 2.—Proportions of Acme Standard Thread.** (See Fig. 2)

Dimensions in inches

| Pitch, $p$ | Threads per in. | Depth of Thread, $d$ | Width at Top of Thread, $f$ | Width at Bottom of Thread, $f'$ | Pitch, $p$ | Threads per in. | Depth of Thread, $d$ | Width at Top of Thread, $f$ | Width at Bottom of Thread, $f'$ |
|---|---|---|---|---|---|---|---|---|---|
| 2 | 1/2 | 1.010 | 0.7414 | 0.7362 | 2/3 | 1 1/2 | 0.3433 | 0.2471 | 0.2419 |
| 1 7/8 | 8/15 | 0.9475 | .6950 | .6897 | 5/8 | 1 3/5 | .3225 | .2316 | .2264 |
| 1 3/4 | 4/7 | .8850 | .6487 | .6435 | 1/2 | 2 | .260 | .1853 | .1801 |
| 1 5/8 | 8/13 | .8225 | .6025 | .5973 | 1/3 | 3 | .1766 | .1235 | .1183 |
| 1 1/2 | 2/3 | .7600 | .5560 | .5508 | 1/4 | 4 | .1350 | .0927 | .0875 |
| 1 3/8 | 8/11 | .6975 | .5097 | .5045 | 1/5 | 5 | .110 | .0741 | .0689 |
| 1 1/4 | 4/5 | .635 | .4633 | .4581 | 1/6 | 6 | .0933 | .0617 | .0565 |
| 1 1/8 | 8/9 | .5725 | .4170 | .4118 | 1/7 | 7 | .0814 | .0530 | .0478 |
| 1 | 1 | .510 | .3707 | .3655 | 1/8 | 8 | .0725 | .0463 | .0411 |
| 7/8 | 1 1/7 | .4475 | .3243 | .3191 | 1/9 | 9 | .0655 | .0413 | .0361 |
| 3/4 | 1 1/3 | .385 | .2780 | .2728 | 1/10 | 10 | .060 | .0371 | .0319 |

stronger than the latter. The angle of the thread is 29 deg. The proportions are given by the following formulas:

$$p = \text{pitch} = (1/\text{No. of threads per inch}) \quad [1]$$

$$d = \text{depth} = 0.5p + 0.010 \quad [2]$$

$$f = \text{flat on top of thread} = p \times 0.3707 \quad [3]$$

$$f' = \text{flat at bottom of thread} = p \times 0.3707 - 0.0052 \quad [4]$$

See also Fig. 2 and Table 2.

## 2. SCREW THREAD STANDARDS

The screw threads generally used in the United States conform to the dimensions of the American Standard Screw Threads as published by the American Standards Association. Tables 3 to 11 show the dimensions and tolerances of the various classes of threads covered by this standard. Five series of screw threads are covered, differing in the diameter-pitch relations of the threads, but all using the American National Standard form of thread.

**FORM OF THREAD.**—The American National Standard form of thread profile is the form previously known as the United States Standard or Sellers profile. The basic angle of thread between the sides of the thread, measured in an axial plane, is 60 deg. The basic width of flat at the root and crest of the thread is

$$F = {}^1/_8\, p = 0.125\, p \quad [5]$$

where $F$ = width of flat, and $p$ = pitch of thread, both in inches.

The basic depth of the thread form is

$$h = 0.649519\, p = 0.649519/n \quad [6]$$

where $h$ = depth of thread, in.; $n$ = number of threads per inch.

**Clearance in Nut at Minor Diameter.**—The screw thread in the nut is so truncated that the basic depth of the thread is reduced by one-sixth or more, depending on the size and pitch, to provide a clearance at the minor diameter of the thread.

**Clearance in Nut at Major Diameter** is provided by decreasing the depth of the truncation triangle any desired amount down to one-third of its theoretical value.

**DEFINITIONS.**—The more important terms are defined below. See also Figs. 4 and 5.

**Major Diameter** is the largest diameter of a screw thread. It applies both to internal and external threads.

**Minor Diameter** is the smallest diameter of a screw thread. The term applies to both internal and external threads.

**Pitch Diameter** is the diameter of an imaginary cylinder whose surface would pass through the threads at such points as to make equal the width of the threads and the width of the spaces cut by the surface of the cylinder.

**Pitch** is the distance from a point on a screw thread to a corresponding point on the next thread, measured parallel to the axis. Pitch, in inches, equals (1/ No. of threads per inch).

**Lead** is the distance a screw thread advances axially in one turn. On a single-thread screw, lead and pitch are identical. On double-thread screws, lead is twice the pitch. On triple-thread screws, lead is three times the pitch, etc.

**Angle of Thread** is the angle included between the sides of the thread, measured in an axial plane.

**Helix Angle** is the angle made by the helix of the thread, at the pitch diameter, with a plane perpendicular to the axis. The tangent of the helix angle $S = L/\pi E$, where $L$ = lead, and $E$ = the pitch diameter.

**Crest** is the top surface joining the two sides of a thread.

**Root** is the bottom surface joining the sides of two adjacent threads.

**Side** is the surface of the thread connecting the crest with the root.

**Depth of Thread** is the distance between crest and root, measured normal to the axis.

**Length of Engagement** is the length of contact between two mating parts, measured axially.

**Depth of Engagement** is the depth of thread contact of two mating parts, measured radially.

**DEFINITIONS RELATING TO FITS.**—**Basic Size** is the theoretical or nominal standard size from which all variations are made.

**Crest Clearance** is the space between the crest of a thread and the root of its mating thread.

**Allowance** is an intentional difference in the dimensions of mating parts. It is the minimum clearance or the maximum interference between mating parts. See Table 4.

**Tolerance** is the amount of variation permitted in the size of a part. In Tables 5 to 11 tolerance is the difference between the maximum and the minimum dimensions given.

**Limits** are the extreme permissible dimensions of a part.

**Fit** is the relation between two mating parts with reference to ease of assembly. Quality of fit depends both on the relative size and the quality of finish of mating parts.

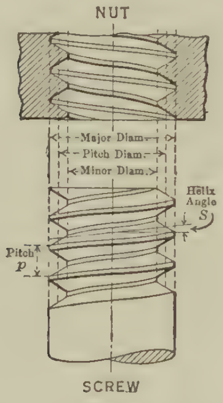


Fig. 4

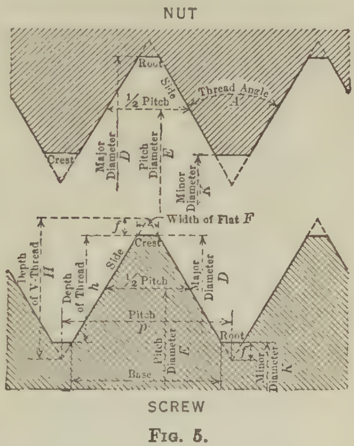


Fig. 5.

**SCREW THREAD SERIES.**—The five series of screw threads are as follows:

**Coarse Thread Series** is the former United States Standard, supplemented in the sizes below $1/4$ in. by a part of the standard proportions for machine screws established by the A.S.M.E. in 1907. General dimensions are given in Table 3. This series is recommended for general use.

**Fine Thread Series** is in accordance with the former regular screw thread series established by the S.A.E. in 1911, supplemented below $1/4$ in. by sizes selected from the standard proportions for machine screws established by the A.S.M.E. in 1907. General dimensions are given in Table 3.

**8-pitch-thread Series** is a series in which the number of threads per inch is constant at 8 irrespective of the diameter. Used on bolts for high-pressure pipe flanges, cylinder-head studs, and cylinder fastenings against pressure, in which it is required that an initial tension be set up in the fastening by elastic deformation of the fastening. To procure proper initial tension it is not practicable that the pitch should increase with the diameter of the thread because the torque required to assemble the fastening would be excessive. Limiting dimensions are given in Table 10.

**12-pitch-thread Series.**—12-pitch threads from $1/2$ in. to and including 1 $3/4$ in. are used in boiler practice. This requires that a worn stud hole be retapped with a tap of the next larger size, the increment being $1/16$ in. throughout most of the range. These threads are widely used also in machine construction for thin nuts on shafts and sleeves. Shoulder diameters should be limited to steps of not less than $1/8$ in. 12-pitch is the coarsest pitch in general use that will permit a threaded collar which screws on a threaded shoulder to fit over a shaft, the difference between shoulder and shaft being $1/8$ in.

**16-pitch-thread Series** is a uniform-pitch series for applications requiring a very fine thread. It is intended primarily for use on threaded adjusting collars and bearing retaining nuts. Limiting dimensions are given in Table 11.

**Table 3.—General Dimensions of Coarse-thread and Fine-thread Series Screw Threads**
(A.S.A. Standard B1.1—1935) For Limiting Dimensions, see Tables 5 to 9

| Size | Threads per Inch, $n$ | Basic Diameters | | | Thread Data | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Diameter, $D$, in. | Pitch Diameter, $E$, in. | Minor Diameter, $K$, in. | Pitch *, $p$, in. | Basic Depth of Thread, $h$, in. | Basic Width of Flat, $p/8$, in. | Min. Width of Flat at Major Diameter of Nut, $p/24$, in. | Helix Angle at Basic Pitch Diam. | Basic Area † of Section at Root, sq. in. |
| | | | | | **Coarse-thread Series** | | | | | |
| No. 1 | 64 | 0.0730 | 0.0629 | 0.0527 | 0.0156250 | 0.01015 | 0.00195 | 0.00065 | 4° 31′ | 0.0022 |
| 2 | 56 | .0860 | .0744 | .0628 | .0178571 | .01160 | .00223 | .00074 | 4° 22′ | .0031 |
| 3 | 48 | .0990 | .0855 | .0719 | .0208333 | .01353 | .00260 | .00087 | 4° 26′ | .0041 |
| 4 | 40 | .1120 | .0958 | .0795 | .0250000 | .01624 | .00312 | .00104 | 4° 45′ | .0050 |
| 5 | 40 | .1250 | .1088 | .0925 | .0250000 | .01624 | .00312 | .00104 | 4° 11′ | .0067 |
| 6 | 32 | .1380 | .1177 | .0974 | .0312500 | .02030 | .00391 | .00130 | 4° 50′ | .0075 |
| 8 | 32 | .1640 | .1437 | .1234 | .0312500 | .02030 | .00391 | .00130 | 3° 58′ | .0120 |
| 10 | 24 | .1900 | .1629 | .1359 | .0416667 | .02706 | .00521 | .00174 | 4° 39′ | .0145 |
| 12 | 24 | .2160 | .1889 | .1619 | .0416667 | .02706 | .00521 | .00174 | 4° 1′ | .0206 |
| 1/4 | 20 | .2500 | .2175 | .1850 | .0500000 | .03248 | .00625 | .00208 | 4° 11′ | .0269 |
| 5/16 | 18 | .3125 | .2764 | .2403 | .0555556 | .03608 | .00694 | .00231 | 3° 40′ | .0454 |
| 3/8 | 16 | .3750 | .3344 | .2938 | .0625000 | .04059 | .00781 | .00260 | 3° 24′ | .0678 |
| 7/16 | 14 | .4375 | .3911 | .3447 | .0714286 | .04639 | .00893 | .00298 | 3° 20′ | .0933 |
| 1/2 | 13 | .5000 | .4500 | .4001 | .0769231 | .04996 | .00962 | .00321 | 3° 7′ | .1257 |
| 9/16 | 12 | .5625 | .5084 | .4542 | .0833333 | .05413 | .01042 | .00347 | 2° 59′ | .1620 |
| 5/8 | 11 | .6250 | .5660 | .5069 | .0909091 | .05905 | .01136 | .00379 | 2° 56′ | .2018 |
| 3/4 | 10 | .7500 | .6850 | .6201 | .1000000 | .06495 | .01250 | .00417 | 2° 40′ | .3020 |
| 7/8 | 9 | .8750 | .8028 | .7307 | .1111111 | .07217 | .01389 | .00463 | 2° 31′ | .4193 |
| 1 | 8 | 1.0000 | .9188 | .8376 | .1250000 | .08119 | .01562 | .00521 | 2° 29′ | .5510 |
| 1 1/8 | 7 | 1.1250 | 1.0322 | .9394 | .1428571 | .09279 | .01786 | .00595 | 2° 31′ | .6931 |
| 1 1/4 | 7 | 1.2500 | 1.1572 | 1.0644 | .1428571 | .09279 | .01786 | .00595 | 2° 15′ | .8898 |
| 1 3/8 | 6 | 1.3750 | 1.2667 | 1.1585 | .1666667 | .10825 | .02083 | .00694 | 2° 24′ | 1.0541 |
| 1 1/2 | 6 | 1.5000 | 1.3917 | 1.2835 | .1666667 | .10825 | .02083 | .00694 | 2° 11′ | 1.2938 |
| 1 3/4 | 5 | 1.7500 | 1.6201 | 1.4902 | .2000000 | .12990 | .02500 | .00833 | 2° 15′ | 1.7441 |
| 2 | 4 1/2 | 2.0000 | 1.8557 | 1.7113 | .2222222 | .14434 | .02778 | .00926 | 2° 11′ | 2.3001 |
| 2 1/4 | 4 1/2 | 2.2500 | 2.1057 | 1.9613 | .2222222 | .14434 | .02778 | .00926 | 1° 55′ | 3.0212 |
| 2 1/2 | 4 | 2.5000 | 2.3376 | 2.1752 | .2500000 | .16238 | .03125 | .01042 | 1° 57′ | 3.7161 |
| 2 3/4 | 4 | 2.7500 | 2.5876 | 2.4252 | .2500000 | .16238 | .03125 | .01042 | 1° 46′ | 4.6194 |
| 3 | 4 | 3.0000 | 2.8376 | 2.6752 | .2500000 | .16238 | .03125 | .01042 | 1° 36′ | 5.6209 |
| 3 1/4 | 4 | 3.2500 | 3.0876 | 2.9252 | .2500000 | .16238 | .03125 | .01042 | 1° 29′ | 6.7205 |
| 3 1/2 | 4 | 3.5000 | 3.3376 | 3.1752 | .2500000 | .16238 | .03125 | .01042 | 1° 22′ | 7.9183 |
| 3 3/4 | 4 | 3.7500 | 3.5876 | 3.4252 | .2500000 | .16238 | .03125 | .01042 | 1° 16′ | 9.2143 |
| 4 | 4 | 4.0000 | 3.8376 | 3.6752 | .2500000 | .16238 | .03125 | .01042 | 1° 11′ | 10.6084 |
| | | | | | **Fine-thread Series** | | | | | |
| No. 0 | 80 | 0.0600 | 0.0519 | 0.0438 | 0.0125000 | 0.00812 | 0.00156 | 0.00052 | 4° 23′ | 0.0015 |
| 1 | 72 | .0730 | .0640 | .0550 | .0138889 | .00902 | .00174 | .00058 | 3° 57′ | .0024 |
| 2 | 64 | .0860 | .0759 | .0657 | .0156250 | .01015 | .00195 | .00065 | 3° 45′ | .0034 |
| 3 | 56 | .0990 | .0874 | .0758 | .0178571 | .01160 | .00223 | .00074 | 3° 43′ | .0045 |
| 4 | 48 | .1120 | .0985 | .0849 | .0208333 | .01353 | .00260 | .00087 | 3° 51′ | .0057 |
| 5 | 44 | .1250 | .1102 | .0955 | .0227273 | .01476 | .00284 | .00095 | 3° 45′ | .0072 |
| 6 | 40 | .1380 | .1218 | .1055 | .0250000 | .01624 | .00312 | .00104 | 3° 44′ | .0087 |
| 8 | 36 | .1640 | .1460 | .1279 | .0277778 | .01804 | .00347 | .00116 | 3° 28′ | .0128 |
| 10 | 32 | .1900 | .1697 | .1494 | .0312500 | .02030 | .00391 | .00130 | 3° 21′ | .0175 |
| 12 | 28 | .2160 | .1928 | .1696 | .0357143 | .02320 | .00446 | .00149 | 3° 22′ | .0226 |
| 1/4 | 28 | .2500 | .2268 | .2036 | .0357143 | .02320 | .00446 | .00149 | 2° 52′ | .0326 |
| 5/16 | 24 | .3125 | .2854 | .2584 | .0416667 | .02706 | .00521 | .00174 | 2° 40′ | .0524 |
| 3/8 | 24 | .3750 | .3479 | .3209 | .0416667 | .02706 | .00521 | .00174 | 2° 11′ | .0809 |
| 7/16 | 20 | .4375 | .4050 | .3725 | .0500000 | .03248 | .00625 | .00208 | 2° 15′ | .1090 |
| 1/2 | 20 | .5000 | .4675 | .4350 | .0500000 | .03248 | .00625 | .00208 | 1° 57′ | .1486 |
| 9/16 | 18 | .5625 | .5264 | .4903 | .0555556 | .03608 | .00694 | .00231 | 1° 55′ | .1888 |
| 5/8 | 18 | .6250 | .5889 | .5528 | .0555556 | .03608 | .00694 | .00231 | 1° 43′ | .2400 |
| 3/4 | 16 | .7500 | .7094 | .6688 | .0625000 | .04059 | .00781 | .00260 | 1° 36′ | .3513 |
| 7/8 | 14 | .8750 | .8286 | .7822 | .0714286 | .04639 | .00893 | .00298 | 1° 34′ | .4805 |
| 1 | 14 | 1.0000 | .9536 | .9072 | .0714286 | .04639 | .00893 | .00298 | 1° 22′ | .6464 |
| 1 1/8 | 12 | 1.1250 | 1.0709 | 1.0167 | .0833333 | .05413 | .01042 | .00347 | 1° 25′ | .8118 |
| 1 1/4 | 12 | 1.2500 | 1.1959 | 1.1417 | .0833333 | .05413 | .01042 | .00347 | 1° 16′ | 1.0238 |
| 1 3/8 | 12 | 1.3750 | 1.3209 | 1.2667 | .0833333 | .05413 | .01042 | .00347 | 1° 9′ | 1.2602 |
| 1 1/2 | 12 | 1.5000 | 1.4459 | 1.3917 | .0833333 | .05413 | .01042 | .00347 | 1° 3′ | 1.5212 |

* This column is given to seven decimal places for computation purposes only.

† Equals $\pi K^2/4$. Basic areas are given for roughly computing ultimate strength of thread. Actual strength, computed from ultimate strength of the material, will depend on actual cross-sectional area at root of thread and an increment contributed by the thread itself.

**SCREW THREAD FITS.**—The four classes of screw thread fits covered by the American Standard range from Class 1, the loosest, to Class 4, the closest. They are produced by the application of specific tolerances, or tolerances together with allowances. Allowances are applicable, in the case of the screw only, to the basic pitch diameter of the thread which is the same for both external and internal threads of like size and pitch. Tolerances are applied *plus* to the hole, and *minus* to the screw. The several classes of fits are described below.

**Class 1 Fit.**—A screw and hole threaded to Class 1 specifications never will produce a close fit, and there always will be " play " or looseness. This fit is recommended for screw thread work where clearance between mating parts is essential for rapid assembly, and where shake or play is unobjectionable. In the Class 1 fit, the maximum nut is basic and the maximum screw is below basic. The major diameter and pitch diameter of the maximum screw are below the basic dimensions by the amount of the allowance specified in Table 4. The tolerance on the nut is *plus*. The tolerance on the screw is *minus*. Pitch-diameter tolerances for a screw or nut of a given pitch are given in Table 4. Limiting dimensions are given in Table 5.

**Class 2 Fit.**—A screw and hole threaded to Class 2 specifications may vary from a fit with no play when pitch diameters of the mating parts are at their common basic value, to a fit with some play in the case of a minimum screw and a maximum hole, when the total pitch diameter tolerance has been applied to both parts. This fit represents a high quality of screw thread product. It is recommended for the majority of interchangeable

**Table 4.—Allowances and Tolerances for Screws and Nuts**
A.S.A. Standard B1.1—1935

| Threads per Inch | **Class 1 Fit** Coarse Thread Series Only | | | | **Class 2 Fit §** Coarse- and Fine-Thread Series Only | | | **Class 3 Fit §** Coarse- and Fine-Thread Series Only | | | **Class 4 Fit** Coarse- and Fine-Thread Series Only | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Allowance, in.* | Pitch Diameter Tolerance, in. | Lead Variation, in.† | Variation in 1/2 Angle‡ | Pitch Diameter Tolerance, in. | Lead Variation, in.† | Variation in 1/2 Angle‡ | Pitch Diameter Tolerance, in. | Lead Variation, in.† | Variation in 1/2 Angle‡ | Interference or Negative Allowance, in.¶ | Pitch Diameter Tolerance, in. | Lead Variation, in.† | Variation in 1/2 Angle‡ |
| 80 | ...... | ...... | ...... | ....... | 0.0017 | 0.0005 | 2° 36′ | 0.0013 | 0.0004 | 1° 59′ | ...... | ...... | ...... | ....... |
| 72 | ...... | ...... | ...... | ....... | .0018 | .0005 | 2° 28′ | .0013 | .0004 | 1° 47′ | ...... | ...... | ...... | ....... |
| 64 | 0.0007 | 0.0026 | 0.0008 | 3° 10′ | .0019 | .0005 | 2° 19′ | .0014 | .0004 | 1° 43′ | ...... | ...... | ...... | ....... |
| 56 | .0008 | .0028 | .0008 | 3° 0′ | .0020 | .0006 | 2° 8′ | .0015 | .0004 | 1° 36′ | ...... | ...... | ...... | ....... |
| 48 | .0009 | .0031 | .0009 | 2° 50′ | .0022 | .0006 | 2° 1′ | .0016 | .0005 | 1° 28′ | ...... | ...... | ...... | ....... |
| 44 | ...... | ...... | ...... | ....... | .0023 | .0007 | 1° 56′ | .0016 | .0005 | 1° 21′ | ...... | ...... | ...... | ....... |
| 40 | .0010 | .0034 | .0010 | 2° 36′ | .0024 | .0007 | 1° 50′ | .0017 | .0005 | 1° 18′ | ...... | ...... | ...... | ....... |
| 36 | ...... | ...... | ...... | ....... | .0025 | .0007 | 1° 43′ | .0018 | .0005 | 1° 14′ | ...... | ...... | ...... | ....... |
| 32 | .0011 | .0038 | .0011 | 2° 19′ | .0027 | .0008 | 1° 39′ | .0019 | .0005 | 1° 10′ | ...... | ...... | ...... | ....... |
| 28 | ...... | ...... | ...... | ....... | .0031 | .0009 | 1° 39′ | .0022 | .0006 | 1° 11′ | 0.0002 | 0.0011 | 0.0003 | 0° 35′ |
| 24 | .0013 | .0046 | .0013 | 2° 6′ | .0033 | .0010 | 1° 31′ | .0024 | .0007 | 1° 6′ | .0003 | .0012 | .0003 | 0° 33′ |
| 20 | .0015 | .0051 | .0015 | 1° 57′ | .0036 | .0010 | 1° 22′ | .0026 | .0008 | 1° 0′ | .0003 | .0013 | .0004 | 0° 30′ |
| 18 | .0016 | .0057 | .0016 | 1° 58′ | .0041 | .0012 | 1° 25′ | .0030 | .0009 | 1° 2′ | .0003 | .0015 | .0004 | 0° 31′ |
| 16 | .0018 | .0063 | .0018 | 1° 55′ | .0045 | .0013 | 1° 22′ | .0032 | .0009 | 0° 59′ | .0004 | .0016 | .0005 | 0° 29′ |
| 14 | .0021 | .0070 | .0020 | 1° 52′ | .0049 | .0014 | 1° 19′ | .0036 | .0010 | 0° 58′ | .0004 | .0018 | .0005 | 0° 29′ |
| 13 | .0022 | .0074 | .0021 | 1° 50′ | .0052 | .0015 | 1° 17′ | .0037 | .0011 | 0° 55′ | .0004 | .0019 | .0005 | 0° 28′ |
| 12 | .0024 | .0079 | .0023 | 1° 49′ | .0056 | .0016 | 1° 17′ | .0040 | .0012 | 0° 55′ | .0005 | .0020 | .0006 | 0° 28′ |
| 11 | .0026 | .0085 | .0025 | 1° 47′ | .0059 | .0017 | 1° 14′ | .0042 | .0012 | 0° 53′ | .0005 | .0021 | .0006 | 0° 26′ |
| 10 | .0028 | .0092 | .0027 | 1° 45′ | .0064 | .0018 | 1° 13′ | .0045 | .0013 | 0° 52′ | .0006 | .0023 | .0007 | 0° 26′ |
| 9 | .0031 | .0100 | .0029 | 1° 43′ | .0070 | .0020 | 1° 12′ | .0049 | .0014 | 0° 51′ | .0006 | .0024 | .0007 | 0° 25′ |
| 8 | .0034 | .0111 | .0032 | 1° 42′ | .0076 | .0022 | 1° 10′ | .0054 | .0016 | 0° 50′ | .0007 | .0027 | .0008 | 0° 25′ |
| 7 | .0039 | .0124 | .0036 | 1° 39′ | .0085 | .0025 | 1° 8′ | .0059 | .0017 | 0° 47′ | .0008 | .0030 | .0009 | 0° 24′ |
| 6 | .0044 | .0145 | .0042 | 1° 40′ | .0101 | .0029 | 1° 9′ | .0071 | .0020 | 0° 49′ | .0009 | .0036 | .0010 | 0° 25′ |
| 5 | .0052 | .0169 | .0049 | 1° 37′ | .0116 | .0033 | 1° 6′ | .0082 | .0024 | 0° 47′ | .0010 | .0041 | .0012 | 0° 23′ |
| 4½ | .0057 | .0184 | .0053 | 1° 35′ | .0127 | .0037 | 1° 5′ | .0089 | .0026 | 0° 46′ | .0011 | .0044 | .0013 | 0° 23′ |
| 4 | .0064 | .0204 | .0059 | 1° 33′ | .0140 | .0040 | 1° 4′ | .0097 | .0028 | 0° 44′ | .0013 | .0048 | .0014 | 0° 22′ |

* Allowances apply to screw only; the nut is basic.

† Variation in lead between any two threads not farther apart than the length of engagement which would absorb one-half the pitch diameter tolerance.

‡ Variation in half-angle which would absorb one-half the pitch diameter tolerance.

§ Allowance is zero.

¶ Negative allowance applies to screw only; the nut is basic.

screw thread work. In the Class 2 fit the minimum nut and the maximum screw are basic. The tolerance on the nut is *plus* and on the screw *minus*. Allowance between pitch diameters of maximum screw and minimum nuts is zero for all pitches and diameters. Pitch diameter tolerances for screw or nut of a given pitch are specified in Table 4. Limiting dimensions are given in Tables 6 and 10.

**Class 3 Fit.**—A screw and hole threaded to Class 3 specifications may vary from a fit with no play, when pitch diameter of both mating parts is at their common basic value, to a fit with a play of about 70% of that allowed under Class 2. This is obtained with a minimum screw and a maximum hole when the total Class 3 pitch diameter tolerance has been applied to both parts. Class 3 fits are recommended only where the high cost of precision tools, and continual checking of tools and products, is warranted. In the Class 3 fit, the minimum nut and the maximum screw are basic. The tolerance on the nut is *plus* and on the screw is *minus*. The allowance between pitch diameters of maximum screw and minimum nut is zero for all pitches and diameters. Pitch diameter tolerances for screw or nut of a given pitch in the coarse- and fine-thread series are specified in Table 4. Limiting dimensions for the coarse- and fine-thread series are given in Table 7

(*Continued on p.* 9–13)

**Table 5.—Limiting Dimensions, Coarse-thread Series Screw Threads, Class 1 Fits**

A.S.A. Standard B1.1—1935

| Size | Threads per Inch | Screw | | | | | Nut | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Diameter, in. | | Pitch Diameter, in.¶ | | Minor Diameter, in., Max.† | Major Diameter, in., Min.‡§ | Pitch Diameter, in.¶ | | Minor Diameter, in. | |
| | | Max.* | Min. | Max.* | Min. | | | Min. | Max. | Min. | Max. |
| No. 1 | 64 | 0.0723 | 0.0671 | 0.0622 | 0.0596 | 0.0531 | 0.0730 | 0.0629 | 0.0655 | 0.0561 | 0.0623 |
| 2 | 56 | .0852 | .0796 | .0736 | .0708 | .0633 | .0860 | .0744 | .0772 | .0667 | .0737 |
| 3 | 48 | .0981 | .0919 | .0846 | .0815 | .0725 | .0990 | .0855 | .0886 | .0764 | .0841 |
| 4 | 40 | .1110 | .1042 | .0948 | .0914 | .0803 | .1120 | .0958 | .0992 | .0849 | .0938 |
| 5 | 40 | .1240 | .1172 | .1078 | .1044 | .0933 | .1250 | .1088 | .1122 | .0979 | .1062 |
| 6 | 32 | .1369 | .1293 | .1166 | .1128 | .0986 | .1380 | .1177 | .1215 | .1042 | .1145 |
| 8 | 32 | .1629 | .1553 | .1426 | .1388 | .1246 | .1640 | .1437 | .1475 | .1302 | .1384 |
| 10 | 24 | .1887 | .1795 | .1616 | .1570 | .1376 | .1900 | .1629 | .1675 | .1449 | .1559 |
| 12 | 24 | .2147 | .2055 | .1876 | .1830 | .1636 | .2160 | .1889 | .1935 | .1709 | .1801 |
| 1/4 | 20 | .2485 | .2383 | .2160 | .2109 | .1872 | .2500 | .2175 | .2226 | .1959 | .2060 |
| 5/16 | 18 | .3109 | .2995 | .2748 | .2691 | .2427 | .3125 | .2764 | .2821 | .2524 | .2630 |
| 3/8 | 16 | .3732 | .3606 | .3326 | .3263 | .2965 | .3750 | .3344 | .3407 | .3073 | .3184 |
| 7/16 | 14 | .4354 | .4214 | .3890 | .3820 | .3478 | .4375 | .3911 | .3981 | .3602 | .3721 |
| 1/2 | 13 | .4978 | .4830 | .4478 | .4404 | .4034 | .5000 | .4500 | .4574 | .4167 | .4290 |
| 9/16 | 12 | .5601 | .5443 | .5060 | .4981 | .4579 | .5625 | .5084 | .5163 | .4723 | .4850 |
| 5/8 | 11 | .6224 | .6054 | .5634 | .5549 | .5109 | .6250 | .5660 | .5745 | .5266 | .5397 |
| 3/4 | 10 | .7472 | .7288 | .6822 | .6730 | .6245 | .7500 | .6850 | .6942 | .6417 | .6553 |
| 7/8 | 9 | .8719 | .8519 | .7997 | .7897 | .7356 | .8750 | .8028 | .8128 | .7547 | .7689 |
| 1 | 8 | .9966 | .9744 | .9154 | .9043 | .8432 | 1.0000 | .9188 | .9299 | .8647 | .8795 |
| 1 1/8 | 7 | 1.1211 | 1.0963 | 1.0283 | 1.0159 | .9458 | 1.1250 | 1.0322 | 1.0446 | .9704 | .9858 |
| 1 1/4 | 7 | 1.2461 | 1.2213 | 1.1533 | 1.1409 | 1.0708 | 1.2500 | 1.1572 | 1.1696 | 1.0954 | 1.1108 |
| 1 3/8 | 6 | 1.3706 | 1.3416 | 1.2623 | 1.2478 | 1.1661 | 1.3750 | 1.2667 | 1.2812 | 1.1946 | 1.2126 |
| 1 1/2 | 6 | 1.4956 | 1.4666 | 1.3873 | 1.3728 | 1.2911 | 1.5000 | 1.3917 | 1.4062 | 1.3196 | 1.3376 |
| 1 3/4 | 5 | 1.7448 | 1.7110 | 1.6149 | 1.5980 | 1.4994 | 1.7500 | 1.6201 | 1.6370 | 1.5335 | 1.5551 |
| 2 | 4 1/2 | 1.9943 | 1.9575 | 1.8500 | 1.8316 | 1.7217 | 2.0000 | 1.8557 | 1.8741 | 1.7594 | 1.7835 |
| 2 1/4 | 4 1/2 | 2.2443 | 2.2075 | 2.1000 | 2.0816 | 1.9717 | 2.2500 | 2.1057 | 2.1241 | 2.0094 | 2.0335 |
| 2 1/2 | 4 | 2.4936 | 2.4528 | 2.3312 | 2.3108 | 2.1869 | 2.5000 | 2.3376 | 2.3580 | 2.2294 | 2.2564 |
| 2 3/4 | 4 | 2.7436 | 2.7028 | 2.5812 | 2.5608 | 2.4369 | 2.7500 | 2.5876 | 2.6080 | 2.4794 | 2.5064 |
| 3 | 4 | 2.9936 | 2.9528 | 2.8312 | 2.8108 | 2.6869 | 3.0000 | 2.8376 | 2.8580 | 2.7294 | 2.7564 |
| 3 1/4 | 4 | 3.2436 | 3.2028 | 3.0812 | 3.0608 | 2.9369 | 3.2500 | 3.0876 | 3.1080 | 2.9794 | 3.0064 |
| 3 1/2 | 4 | 3.4936 | 3.4528 | 3.3312 | 3.3108 | 3.1869 | 3.5000 | 3.3376 | 3.3580 | 3.2294 | 3.2564 |
| 3 3/4 | 4 | 3.7436 | 3.7028 | 3.5812 | 3.5608 | 3.4369 | 3.7500 | 3.5876 | 3.6080 | 3.4794 | 3.5064 |
| 4 | 4 | 3.9936 | 3.9528 | 3.8312 | 3.8108 | 3.6869 | 4.0000 | 3.8376 | 3.8580 | 3.7294 | 3.7564 |

* Below basic; see Class 1 Fits, p. 9–07.

† Dimensions are figured to the intersection of the worn tool arc with a center line through crest and root. Minimum flat at root equals 1/8 *p*.

‡ Basic diameter.

§ Dimensions given are produced only with tap having theoretically sharp corners. Threaded holes must not reject correct basic "Go" gage through interference between gage and rounded root of thread in hole due to worn tap. Minimum flat at root equals 1/24 *p*.

¶ The tolerances specified, *i.e.* (max. pitch diam. – min. pitch diam.), are cumulative, and include variations in lead and angle.

**Table 6.—Limiting Dimensions of Screw Threads, Class 2 Fits**
A.S.A. Standard B1.1—1935. All dimensions in inches.

| Size | Threads per Inch | Screw | | | | | Nut | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Diam. | | Pitch Diam. ¶ | | Minor Diam., | Major Diam., | Pitch Diam. ¶ | | Minor Diam. | |
| | | Max.* | Min.† | Max.* | Min. | Max.‡ | Min.*§ | Min.* | Max. | Min. | Max. |
| **Coarse-thread Series** | | | | | | | | | | | |
| No. 1 | 64 | 0.0730 | 0.0678 | 0.0629 | 0.0610 | 0.0538 | 0.0730 | 0.0629 | 0.0648 | 0.0561 | 0.0623 |
| 2 | 56 | .0860 | .0804 | .0744 | .0724 | .0641 | .0860 | .0744 | .0764 | .0667 | .0737 |
| 3 | 48 | .0990 | .0928 | .0855 | .0833 | .0734 | .0990 | .0855 | .0877 | .0764 | .0841 |
| 4 | 40 | .1120 | .1052 | .0958 | .0934 | .0813 | .1120 | .0958 | .0982 | .0849 | .0938 |
| 5 | 40 | .1250 | .1182 | .1088 | .1064 | .0943 | .1250 | .1088 | .1112 | .0979 | .1062 |
| 6 | 32 | .1380 | .1304 | .1177 | .1150 | .0997 | .1380 | .1177 | .1204 | .1042 | .1145 |
| 8 | 32 | .1640 | .1564 | .1437 | .1410 | .1257 | .1640 | .1437 | .1464 | .1302 | .1384 |
| 10 | 24 | .1900 | .1808 | .1629 | .1596 | .1389 | .1900 | .1629 | .1662 | .1449 | .1559 |
| 12 | 24 | .2160 | .2068 | .1889 | .1856 | .1649 | .2160 | .1889 | .1922 | .1709 | .1801 |
| 1/4 | 20 | .2500 | .2398 | .2175 | .2139 | .1887 | .2500 | .2175 | .2211 | .1959 | .2060 |
| 5/16 | 18 | .3125 | .3011 | .2764 | .2723 | .2443 | .3125 | .2764 | .2805 | .2524 | .2630 |
| 3/8 | 16 | .3750 | .3624 | .3344 | .3299 | .2983 | .3750 | .3344 | .3389 | .3073 | .3184 |
| 7/16 | 14 | .4375 | .4235 | .3911 | .3862 | .3499 | .4375 | .3911 | .3960 | .3602 | .3721 |
| 1/2 | 13 | .5000 | .4852 | .4500 | .4448 | .4056 | .5000 | .4500 | .4552 | .4167 | .4290 |
| 9/16 | 12 | .5625 | .5467 | .5084 | .5028 | .4603 | .5625 | .5084 | .5140 | .4723 | .4850 |
| 5/8 | 11 | .6250 | .6080 | .5660 | .5601 | .5135 | .6250 | .5660 | .5719 | .5266 | .5397 |
| 3/4 | 10 | .7500 | .7316 | .6850 | .6786 | .6273 | .7500 | .6850 | .6914 | .6417 | .6553 |
| 7/8 | 9 | .8750 | .8550 | .8028 | .7958 | .7387 | .8750 | .8028 | .8098 | .7547 | .7689 |
| 1 | 8 | 1.0000 | .9778 | .9188 | .9112 | .8466 | 1.0000 | .9188 | .9264 | .8647 | .8795 |
| 1 1/8 | 7 | 1.1250 | 1.1002 | 1.0322 | 1.0237 | .9497 | 1.1250 | 1.0322 | 1.0407 | .9704 | .9858 |
| 1 1/4 | 7 | 1.2500 | 1.2252 | 1.1572 | 1.1487 | 1.0747 | 1.2500 | 1.1572 | 1.1657 | 1.0954 | 1.1108 |
| 1 3/8 | 6 | 1.3750 | 1.3460 | 1.2667 | 1.2566 | 1.1705 | 1.3750 | 1.2667 | 1.2768 | 1.1946 | 1.2126 |
| 1 1/2 | 6 | 1.5000 | 1.4710 | 1.3917 | 1.3816 | 1.2955 | 1.5000 | 1.3917 | 1.4018 | 1.3196 | 1.3376 |
| 1 3/4 | 5 | 1.7500 | 1.7162 | 1.6201 | 1.6085 | 1.5046 | 1.7500 | 1.6201 | 1.6317 | 1.5335 | 1.5551 |
| 2 | 4 1/2 | 2.0000 | 1.9632 | 1.8557 | 1.8430 | 1.7274 | 2.0000 | 1.8557 | 1.8684 | 1.7594 | 1.7835 |
| 2 1/4 | 4 1/2 | 2.2500 | 2.2132 | 2.1057 | 2.0930 | 1.9774 | 2.2500 | 2.1057 | 2.1184 | 2.0094 | 2.0335 |
| 2 1/2 | 4 | 2.5000 | 2.4592 | 2.3376 | 2.3236 | 2.1933 | 2.5000 | 2.3376 | 2.3516 | 2.2294 | 2.2564 |
| 2 3/4 | 4 | 2.7500 | 2.7092 | 2.5876 | 2.5736 | 2.4433 | 2.7500 | 2.5876 | 2.6016 | 2.4794 | 2.5064 |
| 3 | 4 | 3.0000 | 2.9592 | 2.8376 | 2.8236 | 2.6933 | 3.0000 | 2.8376 | 2.8516 | 2.7294 | 2.7564 |
| 3 1/4 | 4 | 3.2500 | 3.2092 | 3.0876 | 3.0736 | 2.9433 | 3.2500 | 3.0876 | 3.1016 | 2.9794 | 3.0064 |
| 3 1/2 | 4 | 3.5000 | 3.4592 | 3.3376 | 3.3236 | 3.1933 | 3.5000 | 3.3376 | 3.3516 | 3.2294 | 3.2564 |
| 3 3/4 | 4 | 3.7500 | 3.7092 | 3.5876 | 3.5736 | 3.4433 | 3.7500 | 3.5876 | 3.6016 | 3.4794 | 3.5064 |
| 4 | 4 | 4.0000 | 3.9592 | 3.8376 | 3.8236 | 3.6933 | 4.0000 | 3.8376 | 3.8516 | 3.7294 | 3.7564 |
| **Fine-thread Series** | | | | | | | | | | | |
| No. 0 | 80 | 0.0600 | 0.0566 | 0.0519 | 0.0502 | 0.0447 | 0.0600 | 0.0519 | 0.0536 | 0.0465 | 0.0514 |
| 1 | 72 | .0730 | .0694 | .0640 | .0622 | .0560 | .0730 | .0640 | .0658 | .0580 | .0634 |
| 2 | 64 | .0860 | .0822 | .0759 | .0740 | .0668 | .0860 | .0759 | .0778 | .0691 | .0746 |
| 3 | 56 | .0990 | .0950 | .0874 | .0854 | .0771 | .0990 | .0874 | .0894 | .0797 | .0856 |
| 4 | 48 | .1120 | .1076 | .0985 | .0963 | .0864 | .1120 | .0985 | .1007 | .0894 | .0960 |
| 5 | 44 | .1250 | .1204 | .1102 | .1079 | .0971 | .1250 | .1102 | .1125 | .1004 | .1068 |
| 6 | 40 | .1380 | .1332 | .1218 | .1194 | .1073 | .1380 | .1218 | .1242 | .1109 | .1179 |
| 8 | 36 | .1640 | .1590 | .1460 | .1435 | .1299 | .1640 | .1460 | .1485 | .1339 | .1402 |
| 10 | 32 | .1900 | .1846 | .1697 | .1670 | .1517 | .1900 | .1697 | .1724 | .1562 | .1624 |
| 12 | 28 | .2160 | .2098 | .1928 | .1897 | .1722 | .2160 | .1928 | .1959 | .1773 | .1835 |
| 1/4 | 28 | .2500 | .2438 | .2268 | .2237 | .2062 | .2500 | .2268 | .2299 | .2113 | .2173 |
| 5/16 | 24 | .3125 | .3059 | .2854 | .2821 | .2614 | .3125 | .2854 | .2887 | .2674 | .2739 |
| 3/8 | 24 | .3750 | .3684 | .3479 | .3446 | .3239 | .3750 | .3479 | .3512 | .3299 | .3364 |
| 7/16 | 20 | .4375 | .4303 | .4050 | .4014 | .3762 | .4375 | .4050 | .4086 | .3834 | .3906 |
| 1/2 | 20 | .5000 | .4928 | .4675 | .4639 | .4387 | .5000 | .4675 | .4711 | .4459 | .4531 |
| 9/16 | 18 | .5625 | .5543 | .5264 | .5223 | .4943 | .5625 | .5264 | .5305 | .5024 | .5100 |
| 5/8 | 18 | .6250 | .6168 | .5889 | .5848 | .5568 | .6250 | .5889 | .5930 | .5649 | .5725 |
| 3/4 | 16 | .7500 | .7410 | .7094 | .7049 | .6733 | .7500 | .7094 | .7139 | .6823 | .6903 |
| 7/8 | 14 | .8750 | .8652 | .8286 | .8237 | .7874 | .8750 | .8286 | .8335 | .7977 | .8062 |
| 1 | 14 | 1.0000 | .9902 | .9536 | .9487 | .9124 | 1.0000 | .9536 | .9585 | .9227 | .9312 |
| 1 1/8 | 12 | 1.1250 | 1.1138 | 1.0709 | 1.0653 | 1.0228 | 1.1250 | 1.0709 | 1.0765 | 1.0348 | 1.0438 |
| 1 1/4 | 12 | 1.2500 | 1.2388 | 1.1959 | 1.1903 | 1.1478 | 1.2500 | 1.1959 | 1.2015 | 1.1598 | 1.1688 |
| 1 3/8 | 12 | 1.3750 | 1.3638 | 1.3209 | 1.3153 | 1.2728 | 1.3750 | 1.3209 | 1.3265 | 1.2848 | 1.2938 |
| 1 1/2 | 12 | 1.5000 | 1.4888 | 1.4459 | 1.4403 | 1.3978 | 1.5000 | 1.4459 | 1.4515 | 1.4098 | 1.4188 |

* Basic diameter. † Threaded portion only of unfinished hot-rolled material.

‡ Dimensions are figured to the intersection of the worn tool arc with a center line through crest and root. Minimum flat at root equals $1/8\ p$.

§ Dimensions given are produced only with tap having theoretically sharp corners. Threaded hole must not reject correct basic "Go" gage through interference between gage and rounded root of thread in hole due to worn tap. Minimum flat at root equals $1/24\ p$.

¶ The tolerances specified, *i.e.* (max. pitch diam. − min. pitch diam.), are cumulative and include variations in lead and angle.

## Table 7.—Limiting Dimensions of Screw Threads, Class 3 Fits

A.S.A. Standard B1.1—1935. All dimensions in inches.

| Size | Threads per Inch | Screw | | | | | Nut | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Diameter | | Pitch Diameter § | | Minor Diam., Max.† | Major Diam., Min.*‡ | Pitch Diameter, § | | Minor Diameter | |
| | | Max.* | Min. | Max.* | Min. | | | Min.* | Max. | Min. | Max. |
| **Coarse-thread Series** | | | | | | | | | | | |
| No. 1 | 64 | 0.0730 | 0.0692 | 0.0629 | 0.0615 | 0.0538 | 0.0730 | 0.0629 | 0.0643 | 0.0561 | 0.0623 |
| 2 | 56 | .0860 | .0820 | .0744 | .0729 | .0641 | .0860 | .0744 | .0759 | .0667 | .0737 |
| 3 | 48 | .0990 | .0946 | .0855 | .0839 | .0734 | .0990 | .0855 | .0871 | .0764 | .0841 |
| 4 | 40 | .1120 | .1072 | .0958 | .0941 | .0813 | .1120 | .0958 | .0975 | .0849 | .0938 |
| 5 | 40 | .1250 | .1202 | .1088 | .1071 | .0943 | .1250 | .1088 | .1105 | .0979 | .1062 |
| 6 | 32 | .1380 | .1326 | .1177 | .1158 | .0997 | .1380 | .1177 | .1196 | .1042 | .1145 |
| 8 | 32 | .1640 | .1586 | .1437 | .1418 | .1257 | .1640 | .1437 | .1456 | .1302 | .1384 |
| 10 | 24 | .1900 | .1834 | .1629 | .1605 | .1389 | .1900 | .1629 | .1653 | .1449 | .1559 |
| 12 | 24 | .2160 | .2094 | .1889 | .1865 | .1649 | .2160 | .1889 | .1913 | .1709 | .1801 |
| 1/4 | 20 | .2500 | .2428 | .2175 | .2149 | .1887 | .2500 | .2175 | .2201 | .1959 | .2060 |
| 5/16 | 18 | .3125 | .3043 | .2764 | .2734 | .2443 | .3125 | .2764 | .2794 | .2524 | .2630 |
| 3/8 | 16 | .3750 | .3660 | .3344 | .3312 | .2983 | .3750 | .3344 | .3376 | .3073 | .3184 |
| 7/16 | 14 | .4375 | .4277 | .3911 | .3875 | .3499 | .4375 | .3911 | .3947 | .3602 | .3721 |
| 1/2 | 13 | .5000 | .4896 | .4500 | .4463 | .4056 | .5000 | .4500 | .4537 | .4167 | .4290 |
| 9/16 | 12 | .5626 | .5513 | .5084 | .5044 | .4603 | .5625 | .5084 | .5124 | .4723 | .4850 |
| 5/8 | 11 | .6250 | .6132 | .5660 | .5618 | .5135 | .6250 | .5660 | .5702 | .5266 | .5397 |
| 3/4 | 10 | .7500 | .7372 | .6850 | .6805 | .6273 | .7500 | .6850 | .6895 | .6417 | .6553 |
| 7/8 | 9 | .8750 | .8610 | .8028 | .7979 | .7387 | .8750 | .8028 | .8077 | .7547 | .7689 |
| 1 | 8 | 1.0000 | .9848 | .9188 | .9134 | .8466 | 1.0000 | .9188 | .9242 | .8647 | .8795 |
| 1 1/8 | 7 | 1.1250 | 1.1080 | 1.0322 | 1.0263 | .9497 | 1.1250 | 1.0322 | 1.0381 | .9704 | .9858 |
| 1 1/4 | 7 | 1.2500 | 1.2330 | 1.1572 | 1.1513 | 1.0747 | 1.2500 | 1.1572 | 1.1631 | 1.0954 | 1.1108 |
| 1 3/8 | 6 | 1.3750 | 1.3548 | 1.2667 | 1.2596 | 1.1705 | 1.3750 | 1.2667 | 1.2738 | 1.1946 | 1.2126 |
| 1 1/2 | 6 | 1.5000 | 1.4798 | 1.3917 | 1.3846 | 1.2955 | 1.5000 | 1.3917 | 1.3988 | 1.3196 | 1.3376 |
| 1 3/4 | 5 | 1.7500 | 1.7268 | 1.6201 | 1.6119 | 1.5046 | 1.7500 | 1.6201 | 1.6283 | 1.5335 | 1.5551 |
| 2 | 4 1/2 | 2.0000 | 1.9746 | 1.8557 | 1.8468 | 1.7274 | 2.0000 | 1.8557 | 1.8646 | 1.7594 | 1.7835 |
| 2 1/4 | 4 1/2 | 2.2500 | 2.2246 | 2.1057 | 2.0968 | 1.9774 | 2.2500 | 2.1057 | 2.1146 | 2.0094 | 2.0335 |
| 2 1/2 | 4 | 2.5000 | 2.4720 | 2.3376 | 2.3279 | 2.1933 | 2.5000 | 2.3376 | 2.3473 | 2.2294 | 2.2564 |
| 2 3/4 | 4 | 2.7500 | 2.7220 | 2.5876 | 2.5779 | 2.4433 | 2.7500 | 2.5876 | 2.5973 | 2.4794 | 2.5064 |
| 3 | 4 | 3.0000 | 2.9720 | 2.8376 | 2.8279 | 2.6933 | 3.0000 | 2.8376 | 2.8473 | 2.7294 | 2.7564 |
| 3 1/4 | 4 | 3.2500 | 3.2220 | 3.0876 | 3.0779 | 2.9433 | 3.2500 | 3.0876 | 3.0973 | 2.9794 | 3.0064 |
| 3 1/2 | 4 | 3.5000 | 3.4720 | 3.3376 | 3.3279 | 3.1933 | 3.5000 | 3.3376 | 3.3473 | 3.2294 | 3.2564 |
| 3 3/4 | 4 | 3.7500 | 3.7220 | 3.5876 | 3.5779 | 3.4433 | 3.7500 | 3.5876 | 3.5973 | 3.4794 | 3.5064 |
| 4 | 4 | 4.0000 | 3.9720 | 3.8376 | 3.8279 | 3.6933 | 4.0000 | 3.8376 | 3.8473 | 3.7294 | 3.7564 |
| **Fine-thread Series** | | | | | | | | | | | |
| No. 0 | 80 | 0.0600 | 0.0566 | 0.0519 | 0.0506 | 0.0447 | 0.0600 | 0.0519 | 0.0532 | 0.0465 | 0.0514 |
| 1 | 72 | .0730 | .0694 | .0640 | .0627 | .0560 | .0730 | .0640 | .0653 | .0580 | .0634 |
| 2 | 64 | .0860 | .0822 | .0759 | .0745 | .0668 | .0860 | .0759 | .0773 | .0691 | .0746 |
| 3 | 56 | .0990 | .0950 | .0874 | .0859 | .0771 | .0990 | .0874 | .0889 | .0797 | .0856 |
| 4 | 48 | .1120 | .1076 | .0985 | .0969 | .0864 | .1120 | .0985 | .1001 | .0894 | .0960 |
| 5 | 44 | .1250 | .1204 | .1102 | .1086 | .0971 | .1250 | .1102 | .1118 | .1004 | .1068 |
| 6 | 40 | .1380 | .1332 | .1218 | .1201 | .1073 | .1380 | .1218 | .1235 | .1109 | .1179 |
| 8 | 36 | .1640 | .1590 | .1460 | .1442 | .1299 | .1640 | .1460 | .1478 | .1339 | .1402 |
| 10 | 32 | .1900 | .1846 | .1697 | .1678 | .1517 | .1900 | .1697 | .1716 | .1562 | .1624 |
| 12 | 28 | .2160 | .2098 | .1928 | .1906 | .1722 | .2160 | .1928 | .1950 | .1773 | .1835 |
| 1/4 | 28 | .2500 | .2438 | .2268 | .2246 | .2062 | .2500 | .2268 | .2290 | .2113 | .2173 |
| 5/16 | 24 | .3125 | .3059 | .2854 | .2830 | .2614 | .3125 | .2854 | .2878 | .2674 | .2739 |
| 3/8 | 24 | .3750 | .3684 | .3479 | .3455 | .3239 | .3750 | .3479 | .3503 | .3299 | .3364 |
| 7/16 | 20 | .4375 | .4303 | .4050 | .4024 | .3762 | .4375 | .4050 | .4076 | .3834 | .3906 |
| 1/2 | 20 | .5000 | .4928 | .4675 | .4649 | .4387 | .5000 | .4675 | .4701 | .4459 | .4531 |
| 9/16 | 18 | .5625 | .5543 | .5264 | .5234 | .4943 | .5625 | .5264 | .5294 | .5024 | .5100 |
| 5/8 | 18 | .6250 | .6168 | .5889 | .5859 | .5568 | .6250 | .5889 | .5919 | .5649 | .5725 |
| 3/4 | 16 | .7500 | .7410 | .7094 | .7062 | .6733 | .7500 | .7094 | .7126 | .6823 | .6903 |
| 7/8 | 14 | .8750 | .8652 | .8286 | .8250 | .7874 | .8750 | .8286 | .8322 | .7977 | .8062 |
| 1 | 14 | 1.0000 | .9902 | .9536 | .9500 | .9124 | 1.0000 | .9536 | .9572 | .9227 | .9312 |
| 1 1/8 | 12 | 1.1250 | 1.1138 | 1.0709 | 1.0669 | 1.0228 | 1.1250 | 1.0709 | 1.0749 | 1.0348 | 1.0438 |
| 1 1/4 | 12 | 1.2500 | 1.2388 | 1.1959 | 1.1919 | 1.1478 | 1.2500 | 1.1959 | 1.1999 | 1.1598 | 1.1688 |
| 1 3/8 | 12 | 1.3750 | 1.3638 | 1.3209 | 1.3169 | 1.2728 | 1.3750 | 1.3209 | 1.3249 | 1.2848 | 1.2938 |
| 1 1/2 | 12 | 1.5000 | 1.4888 | 1.4459 | 1.4419 | 1.3978 | 1.5000 | 1.4459 | 1.4499 | 1.4098 | 1.4188 |

* Basic diameter.

† Dimensions are figured to the intersection of the worn tool arc with a center line through crest and root. Minimum flat at root of coarse threads equals 1/8 $p$; of fine threads equals 1/24 $p$.

‡ Dimensions given are produced only with tap having theoretically sharp corners. Threaded hole must not reject correct basic "Go" gage through interference between gage and rounded root of thread due to worn tap. Minimum flat at root equals 1/24 $p$.

§ The tolerances specified, *i.e.* (max. pitch diam. − min. pitch diam.), are cumulative, and include variations in lead and angle.

### Table 8.—Limiting Dimensions, Coarse-thread Series Screw Threads, Class 4 Fits

A.S.A. Standard B1.1—1935

| Size, in. | Threads per Inch | Screw: Major Diameter, in. | | Screw: Pitch Diameter, in.¶ | | Screw: Minor Diameter, in., | Nut: Major Diameter, in., | Nut: Pitch Diameter, in.¶ | | Nut: Minor Diameter, in. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Max.* | Min. | Max.† | Min. | Max.‡ | Min.*§ | Min.* | Max. | Min. | Max. |
| 1/4 | 20 | 0.2500 | 0.2428 | 0.2178 | 0.2165 | 0.1887 | 0.2500 | 0.2175 | 0.2188 | 0.1959 | 0.2060 |
| 5/16 | 18 | .3125 | .3043 | .2767 | .2752 | .2443 | .3125 | .2764 | .2779 | .2524 | .2630 |
| 3/8 | 16 | .3750 | .3660 | .3348 | .3332 | .2983 | .3750 | .3344 | .3360 | .3073 | .3184 |
| 7/16 | 14 | .4375 | .4277 | .3915 | .3897 | .3499 | .4375 | .3911 | .3929 | .3602 | .3721 |
| 1/2 | 13 | .5000 | .4896 | .4504 | .4485 | .4056 | .5000 | .4500 | .4519 | .4167 | .4290 |
| 9/16 | 12 | .5625 | .5513 | .5089 | .5069 | .4603 | .5625 | .5084 | .5104 | .4723 | .4850 |
| 5/8 | 11 | .6250 | .6132 | .5665 | .5644 | .5135 | .6250 | .5660 | .5681 | .5266 | .5397 |
| 3/4 | 10 | .7500 | .7372 | .6856 | .6833 | .6273 | .7500 | .6850 | .6873 | .6417 | .6553 |
| 7/8 | 9 | .8750 | .8610 | .8034 | .8010 | .7387 | .8750 | .8028 | .8052 | .7547 | .7689 |
| 1 | 8 | 1.0000 | .9848 | .9195 | .9168 | .8466 | 1.0000 | .9188 | .9215 | .8647 | .8795 |
| 1 1/8 | 7 | 1.1250 | 1.1080 | 1.0330 | 1.0300 | .9497 | 1.1250 | 1.0322 | 1.0352 | .9704 | .9858 |
| 1 1/4 | 7 | 1.2500 | 1.2330 | 1.1580 | 1.1550 | 1.0747 | 1.2500 | 1.1572 | 1.1602 | 1.0954 | 1.1108 |
| 1 3/8 | 6 | 1.3750 | 1.3548 | 1.2676 | 1.2640 | 1.1705 | 1.3750 | 1.2667 | 1.2703 | 1.1946 | 1.2126 |
| 1 1/2 | 6 | 1.5000 | 1.4798 | 1.3926 | 1.3890 | 1.2955 | 1.5000 | 1.3917 | 1.3953 | 1.3196 | 1.3376 |
| 1 3/4 | 5 | 1.7500 | 1.7268 | 1.6211 | 1.6170 | 1.5046 | 1.7500 | 1.6201 | 1.6242 | 1.5335 | 1.5551 |
| 2 | 4 1/2 | 2.0000 | 1.9746 | 1.8568 | 1.8524 | 1.7274 | 2.0000 | 1.8557 | 1.8601 | 1.7594 | 1.7835 |
| 2 1/4 | 4 1/2 | 2.2500 | 2.2246 | 2.1068 | 2.1024 | 1.9774 | 2.2500 | 2.1057 | 2.1101 | 2.0094 | 2.0335 |
| 2 1/2 | 4 | 2.5000 | 2.4720 | 2.3389 | 2.3341 | 2.1933 | 2.5000 | 2.3376 | 2.3424 | 2.2294 | 2.2564 |
| 2 3/4 | 4 | 2.7500 | 2.7220 | 2.5889 | 2.5841 | 2.4433 | 2.7500 | 2.5876 | 2.5924 | 2.4794 | 2.5064 |
| 3 | 4 | 3.0000 | 2.9720 | 2.8389 | 2.8341 | 2.6933 | 3.0000 | 2.8376 | 2.8424 | 2.7294 | 2.7564 |
| 3 1/4 | 4 | 3.2500 | 3.2220 | 3.0889 | 3.0841 | 2.9433 | 3.2500 | 3.0876 | 3.0924 | 2.9794 | 3.0064 |
| 3 1/2 | 4 | 3.5000 | 3.4720 | 3.3389 | 3.3341 | 3.1933 | 3.5000 | 3.3376 | 3.3424 | 3.2294 | 3.2564 |
| 3 3/4 | 4 | 3.7500 | 3.7220 | 3.5889 | 3.5841 | 3.4433 | 3.7500 | 3.5876 | 3.5924 | 3.4794 | 3.5064 |
| 4 | 4 | 4.0000 | 3.9720 | 3.8389 | 3.8341 | 3.6933 | 4.0000 | 3.8376 | 3.8424 | 3.7294 | 3.7564 |

### Table 9.—Limiting Dimensions of Fine-thread Series Screw Threads, Class 4 Fits

A.S.A. Standard B1.1—1935

| Size, in. | Threads per Inch | Screw: Major Diameter, in. | | Screw: Pitch Diameter, in.¶ | | Screw: Minor Diameter, in., | Nut: Major Diameter, in., | Nut: Pitch Diameter, in.¶ | | Nut: Minor Diameter, in. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Max.* | Min. | Max.† | Min. | Max.‡ | Min.*§ | Min. | Max. | Min. | Max. |
| 1/4 | 28 | 0.2500 | 0.2438 | 0.2270 | 0.2259 | 0.2062 | 0.2500 | 0.2268 | 0.2279 | 0.2113 | 0.2173 |
| 5/16 | 24 | .3125 | .3059 | .2857 | .2845 | .2614 | .3125 | .2854 | .2866 | .2674 | .2739 |
| 3/8 | 24 | .3750 | .3684 | .3482 | .3470 | .3239 | .3750 | .3479 | .3491 | .3299 | .3364 |
| 7/16 | 20 | .4375 | .4303 | .4053 | .4040 | .3762 | .4375 | .4050 | .4063 | .3834 | .3906 |
| 1/2 | 20 | .5000 | .4928 | .4678 | .4665 | .4387 | .5000 | .4675 | .4688 | .4459 | .4531 |
| 9/16 | 18 | .5625 | .5543 | .5267 | .5252 | .4943 | .5625 | .5264 | .5279 | .5024 | .5100 |
| 5/8 | 18 | .6250 | .6168 | .5892 | .5877 | .5568 | .6250 | .5889 | .5904 | .5649 | .5725 |
| 3/4 | 16 | .7500 | .7410 | .7098 | .7082 | .6733 | .7500 | .7094 | .7110 | .6823 | .6903 |
| 7/8 | 14 | .8750 | .8652 | .8290 | .8272 | .7874 | .8750 | .8286 | .8304 | .7977 | .8062 |
| 1 | 14 | 1.0000 | .9902 | .9540 | .9522 | .9124 | 1.0000 | .9536 | .9554 | .9227 | .9312 |
| 1 1/8 | 12 | 1.1250 | 1.1138 | 1.0714 | 1.0694 | 1.0228 | 1.1250 | 1.0709 | 1.0729 | 1.0348 | 1.0438 |
| 1 1/4 | 12 | 1.2500 | 1.2388 | 1.1964 | 1.1944 | 1.1478 | 1.2500 | 1.1959 | 1.1979 | 1.1598 | 1.1688 |
| 1 3/8 | 12 | 1.3750 | 1.3638 | 1.3214 | 1.3194 | 1.2728 | 1.3750 | 1.3209 | 1.3229 | 1.2848 | 1.2938 |
| 1 1/2 | 12 | 1.5000 | 1.4888 | 1.4464 | 1.4444 | 1.3978 | 1.5000 | 1.4459 | 1.4479 | 1.4098 | 1.4188 |

* Basic diameter.

† Above basic; see Class 4 Fit, p. 9-13.

‡ Dimensions are figured to the intersection of the worn tool arc with a center line through crest and root. Minimum flat at root = 1/8 $p$.

§ Dimensions given are produced only with tap having theoretically sharp corners. Threaded hole must not reject correct basic "Go" gage through interference between gage and rounded root of thread in hole due to worn tap. Minimum flat at root = 1/24 $p$.

¶ The tolerances specified, *i.e.* (max. pitch diam. − min. pitch diam.), are cumulative, and include variations in lead and angle.

## Table 10.—Limiting Dimensions of 8-pitch-thread Series Screw Threads

A.S.A. Standard B1.1—1935

| Size, in. | Threads per Inch* | Screws | | | | | Nuts | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Diameter, in. | | Pitch Diameter, in.¶ | | Minor Diameter, in., | Major Diameter, in., | Pitch Diameter, in.¶ | | Minor Diameter, in. | |
| | | Max.† | Min. | Max.† | Min. | Max.‡ | Min.†§ | Min. | Max. | Min. | Max. |
| **Class 2 Fits** | | | | | | | | | | | |
| 1 | 8 | 1.0000 | 0.9848 | 0.9188 | 0.9112 | 0.8466 | 1.0000 | 0.9188 | 0.9264 | 0.8647 | 0.8795 |
| 1 1/8 | 8 | 1.1250 | 1.1098 | 1.0438 | 1.0359 | 0.9716 | 1.1250 | 1.0438 | 1.0517 | 0.9897 | 1.0045 |
| 1 1/4 | 8 | 1.2500 | 1.2348 | 1.1688 | 1.1605 | 1.0966 | 1.2500 | 1.1688 | 1.1771 | 1.1147 | 1.1295 |
| 1 3/8 | 8 | 1.3750 | 1.3598 | 1.2938 | 1.2852 | 1.2216 | 1.3750 | 1.2938 | 1.3024 | 1.2397 | 1.2545 |
| 1 1/2 | 8 | 1.5000 | 1.4848 | 1.4188 | 1.4098 | 1.3466 | 1.5000 | 1.4188 | 1.4278 | 1.3647 | 1.3795 |
| 1 5/8 | 8 | 1.6250 | 1.6098 | 1.5438 | 1.5345 | 1.4716 | 1.6250 | 1.5438 | 1.5531 | 1.4897 | 1.5045 |
| 1 3/4 | 8 | 1.7500 | 1.7348 | 1.6688 | 1.6591 | 1.5966 | 1.7500 | 1.6688 | 1.6785 | 1.6147 | 1.6295 |
| 1 7/8 | 8 | 1.8750 | 1.8598 | 1.7938 | 1.7838 | 1.7216 | 1.8750 | 1.7938 | 1.8038 | 1.7397 | 1.7545 |
| 2 | 8 | 2.0000 | 1.9848 | 1.9188 | 1.9084 | 1.8466 | 2.0000 | 1.9188 | 1.9292 | 1.8647 | 1.8795 |
| 2 1/8 | 8 | 2.1250 | 2.1098 | 2.0438 | 2.0331 | 1.9716 | 2.1250 | 2.0438 | 2.0545 | 1.9897 | 2.0045 |
| 2 1/4 | 8 | 2.2500 | 2.2348 | 2.1688 | 2.1578 | 2.0966 | 2.2500 | 2.1688 | 2.1798 | 2.1147 | 2.1295 |
| 2 1/2 | 8 | 2.5000 | 2.4848 | 2.4188 | 2.4071 | 2.3466 | 2.5000 | 2.4188 | 2.4305 | 2.3647 | 2.3795 |
| 2 3/4 | 8 | 2.7500 | 2.7348 | 2.6688 | 2.6564 | 2.5966 | 2.7500 | 2.6688 | 2.6812 | 2.6147 | 2.6295 |
| 3 | 8 | 3.0000 | 2.9848 | 2.9188 | 2.9058 | 2.8466 | 3.0000 | 2.9188 | 2.9318 | 2.8647 | 2.8795 |
| 3 1/4 | 8 | 3.2500 | 3.2348 | 3.1688 | 3.1556 | 3.0966 | 3.2500 | 3.1688 | 3.1820 | 3.1147 | 3.1295 |
| 3 1/2 | 8 | 3.5000 | 3.4848 | 3.4188 | 3.4055 | 3.3466 | 3.5000 | 3.4188 | 3.4321 | 3.3647 | 3.3795 |
| 3 3/4 | 8 | 3.7500 | 3.7348 | 3.6688 | 3.6554 | 3.5966 | 3.7500 | 3.6688 | 3.6822 | 3.6147 | 3.6295 |
| 4 | 8 | 4.0000 | 3.9848 | 3.9188 | 3.9053 | 3.8466 | 4.0000 | 3.9188 | 3.9323 | 3.8647 | 3.8795 |
| 4 1/4 | 8 | 4.2500 | 4.2348 | 4.1688 | 4.1551 | 4.0966 | 4.2500 | 4.1688 | 4.1825 | 4.1147 | 4.1295 |
| 4 1/2 | 8 | 4.5000 | 4.4848 | 4.4188 | 4.4050 | 4.3466 | 4.5000 | 4.4188 | 4.4326 | 4.3647 | 4.3795 |
| 4 3/4 | 8 | 4.7500 | 4.7348 | 4.6688 | 4.6549 | 4.5966 | 4.7500 | 4.6688 | 4.6827 | 4.6147 | 4.6295 |
| 5 | 8 | 5.0000 | 4.9848 | 4.9188 | 4.9048 | 4.8466 | 5.0000 | 4.9188 | 4.9328 | 4.8647 | 4.8795 |
| 5 1/4 | 8 | 5.2500 | 5.2348 | 5.1688 | 5.1547 | 5.0966 | 5.2500 | 5.1688 | 5.1829 | 5.1147 | 5.1295 |
| 5 1/2 | 8 | 5.5000 | 5.4848 | 5.4188 | 5.4046 | 5.3466 | 5.5000 | 5.4188 | 5.4330 | 5.3647 | 5.3795 |
| 5 3/4 | 8 | 5.7500 | 5.7348 | 5.6688 | 5.6545 | 5.5966 | 5.7500 | 5.6688 | 5.6831 | 5.6147 | 5.6295 |
| 6 | 8 | 6.0000 | 5.9848 | 5.9188 | 5.9044 | 5.8466 | 6.0000 | 5.9188 | 5.9332 | 5.8647 | 5.8795 |
| **Class 3 Fits** | | | | | | | | | | | |
| 1 | 8 | 1.0000 | 0.9848 | 0.9188 | 0.9134 | 0.8466 | 1.0000 | 0.9188 | 0.9242 | 0.8647 | 0.8795 |
| 1 1/8 | 8 | 1.1250 | 1.1098 | 1.0438 | 1.0383 | 0.9716 | 1.1250 | 1.0438 | 1.0493 | 0.9897 | 1.0045 |
| 1 1/4 | 8 | 1.2500 | 1.2348 | 1.1688 | 1.1630 | 1.0966 | 1.2500 | 1.1688 | 1.1746 | 1.1147 | 1.1295 |
| 1 3/8 | 8 | 1.3750 | 1.3598 | 1.2938 | 1.2877 | 1.2216 | 1.3750 | 1.2938 | 1.2999 | 1.2397 | 1.2545 |
| 1 1/2 | 8 | 1.5000 | 1.4848 | 1.4188 | 1.4125 | 1.3466 | 1.5000 | 1.4188 | 1.4251 | 1.3647 | 1.3795 |
| 1 5/8 | 8 | 1.6250 | 1.6098 | 1.5438 | 1.5373 | 1.4716 | 1.6250 | 1.5438 | 1.5503 | 1.4897 | 1.5045 |
| 1 3/4 | 8 | 1.7500 | 1.7348 | 1.6688 | 1.6620 | 1.5966 | 1.7500 | 1.6688 | 1.6756 | 1.6147 | 1.6295 |
| 1 7/8 | 8 | 1.8750 | 1.8598 | 1.7938 | 1.7868 | 1.7216 | 1.8750 | 1.7938 | 1.8008 | 1.7397 | 1.7545 |
| 2 | 8 | 2.0000 | 1.9848 | 1.9188 | 1.9115 | 1.8466 | 2.0000 | 1.9188 | 1.9261 | 1.8647 | 1.8795 |
| 2 1/8 | 8 | 2.1250 | 2.1098 | 2.0438 | 2.0363 | 1.9716 | 2.1250 | 2.0438 | 2.0513 | 1.9897 | 2.0045 |
| 2 1/4 | 8 | 2.2500 | 2.2348 | 2.1688 | 2.1611 | 2.0966 | 2.2500 | 2.1688 | 2.1765 | 2.1147 | 2.1295 |
| 2 1/2 | 8 | 2.5000 | 2.4848 | 2.4188 | 2.4106 | 2.3466 | 2.5000 | 2.4188 | 2.4270 | 2.3647 | 2.3795 |
| 2 3/4 | 8 | 2.7500 | 2.7348 | 2.6688 | 2.6601 | 2.5966 | 2.7500 | 2.6688 | 2.6775 | 2.6147 | 2.6295 |
| 3 | 8 | 3.0000 | 2.9848 | 2.9188 | 2.9096 | 2.8466 | 3.0000 | 2.9188 | 2.9280 | 2.8647 | 2.8795 |
| 3 1/4 | 8 | 3.2500 | 3.2348 | 3.1688 | 3.1595 | 3.0966 | 3.2500 | 3.1688 | 3.1781 | 3.1147 | 3.1295 |
| 3 1/2 | 8 | 3.5000 | 3.4848 | 3.4188 | 3.4095 | 3.3466 | 3.5000 | 3.4188 | 3.4281 | 3.3647 | 3.3795 |
| 3 3/4 | 8 | 3.7500 | 3.7348 | 3.6688 | 3.6594 | 3.5966 | 3.7500 | 3.6688 | 3.6782 | 3.6147 | 3.6295 |
| 4 | 8 | 4.0000 | 3.9848 | 3.9188 | 3.9093 | 3.8466 | 4.0000 | 3.9188 | 3.9283 | 3.8647 | 3.8795 |
| 4 1/4 | 8 | 4.2500 | 4.2348 | 4.1688 | 4.1592 | 4.0966 | 4.2500 | 4.1688 | 4.1784 | 4.1147 | 4.1295 |
| 4 1/2 | 8 | 4.5000 | 4.4848 | 4.4188 | 4.4091 | 4.3466 | 4.5000 | 4.4188 | 4.4285 | 4.3647 | 4.3795 |
| 4 3/4 | 8 | 4.7500 | 4.7348 | 4.6688 | 4.6590 | 4.5966 | 4.7500 | 4.6688 | 4.6786 | 4.6147 | 4.6295 |
| 5 | 8 | 5.0000 | 4.9848 | 4.9188 | 4.9089 | 4.8466 | 5.0000 | 4.9188 | 4.9287 | 4.8647 | 4.8795 |
| 5 1/4 | 8 | 5.2500 | 5.2348 | 5.1688 | 5.1589 | 5.0966 | 5.2500 | 5.1688 | 5.1787 | 5.1147 | 5.1295 |
| 5 1/2 | 8 | 5.5000 | 5.4848 | 5.4188 | 5.4088 | 5.3466 | 5.5000 | 5.4188 | 5.4288 | 5.3647 | 5.3795 |
| 5 3/4 | 8 | 5.7500 | 5.7348 | 5.6688 | 5.6587 | 5.5966 | 5.7500 | 5.6688 | 5.6789 | 5.6147 | 5.6295 |
| 6 | 8 | 6.0000 | 5.9848 | 5.9188 | 5.9086 | 5.8466 | 6.0000 | 5.9188 | 5.9290 | 5.8647 | 5.8795 |

* Pitch $p$ = 0.1250 in.; basic depth of thread $h$ = 0.08119 in.

† Basic diameter.

‡ Dimensions are figured to intersection of worn tool arc with center line through crest and root. Minimum flat at root = $1/8\, p$ = 0.01562 in.

§ Dimensions given are allowable only with tap having theoretically sharp corners. Threaded hole must not reject correct basic "Go" gage through interference between gage and rounded root of thread in hole due to worn tap. Minimum flat at root = $1/24\, p$ = 0.00521 in.

¶ The tolerances specified, i.e. (max. pitch diam. − min. pitch diam.), are cumulative, and include variations in lead and angle.

*(Continued from p. 9-08)*

Limiting dimensions for 8-pitch and 16-pitch thread series are given in Tables 10 and 11.

**Class 4 Fit.** Screws and holes threaded to Class 4 specifications are intended to meet unusual requirements, more exacting than those for which Class 3 is intended. It is a selective fit if initial assembly by hand is required and is not as yet (1935) adaptable to quantity production. There is an actual interference between a maximum screw and a minimum hole inasmuch as the pitch diameter of the maximum screw is greater than the basic value by a definite allowance, and the pitch diameter of the minimum hole is basic. With a minimum screw and maximum hole a very slight amount of play is present when the total Class 4 pitch diameter tolerance has been applied to both parts. In Class 4 fits, the minimum nut is basic. The pitch diameter of the minimum screw is above basic by the amount of the interference specified in Table 3. The tolerance on the nut is plus and on the screw is minus. Pitch diameter tolerances for a screw or nut of a given pitch in the coarse- and fine-thread series are specified in Table 4. Limiting dimensions are given in Tables 8 and 9.

Table 11.—Limiting Dimensions of 16-pitch Thread Series Screw Threads, Class 3 Fits

A. S. A. Standard B1.1—1935

| Size, in. | Threads per Inch* | Screws: Major Diameter, in., Max.† | Screws: Major Diameter, in., Min. | Screws: Pitch Diameter, in.¶, Max. | Screws: Pitch Diameter, in.¶, Min. | Screws: Minor Diameter, in., Max.‡ | Nuts: Major Diameter, in., Min.†§ | Nuts: Pitch Diameter, in.¶, Min.† | Nuts: Pitch Diameter, in.¶, Max. | Nuts: Minor Diameter, in., Min. | Nuts: Minor Diameter, in., Max. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3/4 | 16 | 0.7500 | 0.7410 | 0.7094 | 0.7062 | 0.6733 | 0.7500 | 0.7094 | 0.7126 | 0.6823 | 0.6903 |
| 13/16 | 16 | .8125 | .8035 | .7719 | .7684 | .7358 | .8125 | .7719 | .7754 | .7448 | .7528 |
| 7/8 | 16 | .8750 | .8660 | .8344 | .8308 | .7983 | .8750 | .8344 | .8380 | .8073 | .8153 |
| 15/16 | 16 | .9375 | .9285 | .8969 | .8933 | .8608 | .9375 | .8969 | .9005 | .8698 | .8778 |
| 1 | 16 | 1.0000 | .9910 | .9594 | .9557 | .9233 | 1.0000 | .9594 | .9631 | .9323 | .9403 |
| 1 1/16 | 16 | 1.0625 | 1.0535 | 1.0219 | 1.0182 | .9858 | 1.0625 | 1.0219 | 1.0256 | .9948 | 1.0028 |
| 1 1/8 | 16 | 1.1250 | 1.1160 | 1.0844 | 1.0806 | 1.0483 | 1.1250 | 1.0844 | 1.0882 | 1.0573 | 1.0653 |
| 1 3/16 | 16 | 1.1875 | 1.1785 | 1.1469 | 1.1431 | 1.1108 | 1.1875 | 1.1469 | 1.1507 | 1.1198 | 1.1278 |
| 1 1/4 | 16 | 1.2500 | 1.2410 | 1.2094 | 1.2056 | 1.1733 | 1.2500 | 1.2094 | 1.2132 | 1.1823 | 1.1903 |
| 1 5/16 | 16 | 1.3125 | 1.3035 | 1.2719 | 1.2680 | 1.2358 | 1.3125 | 1.2719 | 1.2758 | 1.2448 | 1.2528 |
| 1 3/8 | 16 | 1.3750 | 1.3660 | 1.3344 | 1.3305 | 1.2983 | 1.3750 | 1.3344 | 1.3383 | 1.3073 | 1.3153 |
| 1 7/16 | 16 | 1.4375 | 1.4285 | 1.3969 | 1.3929 | 1.3608 | 1.4375 | 1.3969 | 1.4009 | 1.3698 | 1.3778 |
| 1 1/2 | 16 | 1.5000 | 1.4910 | 1.4594 | 1.4554 | 1.4233 | 1.5000 | 1.4594 | 1.4634 | 1.4323 | 1.4403 |
| 1 9/16 | 16 | 1.5625 | 1.5535 | 1.5219 | 1.5179 | 1.4858 | 1.5625 | 1.5219 | 1.5259 | 1.4948 | 1.5028 |
| 1 5/8 | 16 | 1.6250 | 1.6160 | 1.5844 | 1.5803 | 1.5483 | 1.6250 | 1.5844 | 1.5885 | 1.5573 | 1.5653 |
| 1 11/16 | 16 | 1.6875 | 1.6785 | 1.6469 | 1.6428 | 1.6108 | 1.6875 | 1.6469 | 1.6510 | 1.6198 | 1.6278 |
| 1 3/4 | 16 | 1.7500 | 1.7410 | 1.7094 | 1.7053 | 1.6733 | 1.7500 | 1.7094 | 1.7135 | 1.6823 | 1.6903 |
| 1 13/16 | 16 | 1.8125 | 1.8035 | 1.7719 | 1.7677 | 1.7358 | 1.8125 | 1.7719 | 1.7761 | 1.7448 | 1.7528 |
| 1 7/8 | 16 | 1.8750 | 1.8660 | 1.8344 | 1.8302 | 1.7983 | 1.8750 | 1.8344 | 1.8386 | 1.8073 | 1.8153 |
| 1 15/16 | 16 | 1.9375 | 1.9285 | 1.8969 | 1.8927 | 1.8608 | 1.9375 | 1.8969 | 1.9011 | 1.8698 | 1.8778 |
| 2 | 16 | 2.0000 | 1.9910 | 1.9594 | 1.9551 | 1.9233 | 2.0000 | 1.9594 | 1.9637 | 1.9323 | 1.9403 |
| 2 1/16 | 16 | 2.0625 | 2.0535 | 2.0219 | 2.0176 | 1.9858 | 2.0625 | 2.0219 | 2.0262 | 1.9948 | 2.0028 |
| 2 1/8 | 16 | 2.1250 | 2.1160 | 2.0844 | 2.0801 | 2.0483 | 2.1250 | 2.0844 | 2.0887 | 2.0573 | 2.0653 |
| 2 3/16 | 16 | 2.1875 | 2.1785 | 2.1469 | 2.1426 | 2.1108 | 2.1875 | 2.1469 | 2.1512 | 2.1198 | 2.1278 |
| 2 1/4 | 16 | 2.2500 | 2.2410 | 2.2094 | 2.2050 | 2.1733 | 2.2500 | 2.2094 | 2.2138 | 2.1823 | 2.1903 |
| 2 5/16 | 16 | 2.3125 | 2.3035 | 2.2719 | 2.2675 | 2.2358 | 2.3125 | 2.2719 | 2.2763 | 2.2448 | 2.2528 |
| 2 3/8 | 16 | 2.3750 | 2.3660 | 2.3344 | 2.3300 | 2.2983 | 2.3750 | 2.3344 | 2.3388 | 2.3073 | 2.3153 |
| 2 7/16 | 16 | 2.4375 | 2.4285 | 2.3969 | 2.3924 | 2.3608 | 2.4375 | 2.3969 | 2.4014 | 2.3698 | 2.3778 |
| 2 1/2 | 16 | 2.5000 | 2.4910 | 2.4594 | 2.4549 | 2.4233 | 2.5000 | 2.4594 | 2.4639 | 2.4323 | 2.4403 |
| 2 5/8 | 16 | 2.6250 | 2.6160 | 2.5844 | 2.5799 | 2.5483 | 2.6250 | 2.5844 | 2.5889 | 2.5573 | 2.5653 |
| 2 3/4 | 16 | 2.7500 | 2.7410 | 2.7094 | 2.7048 | 2.6733 | 2.7500 | 2.7094 | 2.7140 | 2.6823 | 2.6903 |
| 2 7/8 | 16 | 2.8750 | 2.8660 | 2.8344 | 2.8298 | 2.7983 | 2.8750 | 2.8344 | 2.8390 | 2.8073 | 2.8153 |
| 3 | 16 | 3.0000 | 2.9910 | 2.9594 | 2.9547 | 2.9233 | 3.0000 | 2.9594 | 2.9641 | 2.9323 | 2.9403 |
| 3 1/8 | 16 | 3.1250 | 3.1160 | 3.0844 | 3.0797 | 3.0483 | 3.1250 | 3.0844 | 3.0891 | 3.0573 | 3.0653 |
| 3 1/4 | 16 | 3.2500 | 3.2410 | 3.2094 | 3.2046 | 3.1733 | 3.2500 | 3.2094 | 3.2142 | 3.1823 | 3.1903 |
| 3 3/8 | 16 | 3.3750 | 3.3660 | 3.3344 | 3.3296 | 3.2983 | 3.3750 | 3.3344 | 3.3392 | 3.3073 | 3.3153 |
| 3 1/2 | 16 | 3.5000 | 3.4910 | 3.4594 | 3.4545 | 3.4233 | 3.5000 | 3.4594 | 3.4643 | 3.4323 | 3.4403 |
| 3 5/8 | 16 | 3.6250 | 3.6160 | 3.5844 | 3.5795 | 3.5483 | 3.6250 | 3.5844 | 3.5893 | 3.5573 | 3.5653 |
| 3 3/4 | 16 | 3.7500 | 3.7410 | 3.7094 | 3.7044 | 3.6733 | 3.7500 | 3.7094 | 3.7144 | 3.6823 | 3.6903 |
| 3 7/8 | 16 | 3.8750 | 3.8660 | 3.8344 | 3.8294 | 3.7983 | 3.8750 | 3.8344 | 3.8394 | 3.8073 | 3.8153 |
| 4 | 16 | 4.0000 | 3.9910 | 3.9594 | 3.9543 | 3.9233 | 4.0000 | 3.9594 | 3.9645 | 3.9323 | 3.9403 |

* Pitch $p = 0.06250$ in.; basic depth of thread $h = 0.04059$ in. † Basic diameter.

‡ Dimensions are figured to intersection of worn tool arc with center line through crest and root; minimum flat at root = $1/8\ p = 0.00781$ in.

§ Dimensions given are allowable only with tap having theoretically sharp corners. Threaded hole must not reject correct basic 'Go' gage through interference between gage and rounded root of thread in hole due to worn tap. Minimum flat at root = $1/24\ p = 0.00260$ in.

¶ The tolerances specified, i.e. (max. pitch diam. − min. pitch diam.), are cumulative, and include variations in lead and angle.

**All Classes.**—Tolerances for pitch diameter as specified in Table 4 are cumulative and include all variations of lead and angle. The full tolerance on pitch diameter, therefore, is not available unless lead and angle of the thread are perfect. Variations in lead (per length of thread engaged) and in angle, respectively, can each be compensated by half the tolerance on pitch diameter shown. If lead and angle variations both exist to the amount tabulated, the pitch diameter must be reduced by the full tolerance or the thread will not enter a basic nut or gage, nor is the full tolerance available unless the gage is perfect. If no lead variation existed, the angle variation could be twice that given and *vice versa*. These extreme conditions, however, are not desirable.

Specified tolerances are based on the length of engagement not to exceed the nominal or major diameter of the screw, except in the 8-pitch, 12-pitch and 16-pitch thread series. In the 8-pitch series, they are based on a length of engagement of 1 × major diameter for sizes from 1 to 3 in. inclusive, and a length of engagement of 3 in. for sizes over 3 in. In the 12-pitch and 16-pitch series, they are based on a length of engagement of six threads. Where greater lengths of engagement are required, a corresponding increase in the accuracy of lead and thread form is necessary. This may be obtained by the use of more accurate threading tools and longer "Go" gages.

Crest clearances allowed will permit rounded root forms in both nut and screw which may be formed by tools perfectly rounded, or rounded as the result of wear. The maximum minor diameter is that which would result from using a threading tool, set at the maximum pitch diameter, whose point has been worn or ground back from the basic outline by an amount equal to 1/18 of the basic depth of engagement, or 0.03608 × pitch.

**THE DARDELET THREAD** was designed primarily as a thread on which the nut would be self locking. Fig. 6 is a section of the thread through the nut and bolt. The thread is similar in form to the Acme thread (see Fig. 2), the sides being inclined at an angle of 14 1/2 deg. The bases of the several threads on the bolt are joined by a surface inclined at 6 deg. to the axis of the bolt. The crests of the nut thread are parallel to the surfaces joining the bases of the bolt threads. The distance between the successive corresponding thread faces is about 1 3/4 times the thickness of the thread at the mean thread diameter. A space exists between the threads of the nut and the bolt when the two are assembled together. These relations are shown in Fig. 6.

**Table 12.—S.A.E. Standard Screw Threads, Extra Fine Series**

S.A.E. Handbook, 1936 Edition. All dimensions in inches

| Size | Threads per in. | Screws | | | | | Nuts | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Diam. | | Pitch Diam. | | Minor Diam., | Major Diam., | Pitch Diam. | | Minor Diam. | |
| | | Max. | Min. | Max. | Min. | Max. | Min. | Min. | Max. | Min. | Max. |
| Class 2 Fit | | | | | | | | | | | |
| 1/4 | 32 | 0.2500 | 0.2446 | 0.2297 | 0.2265 | 0.2117 | 0.2500 | 0.2297 | 0.2329 | 0.2162 | 0.2208 |
| 5/16 | 32 | .3125 | .3071 | .2922 | .2889 | .2742 | .3125 | .2922 | .2955 | .2787 | .2833 |
| 3/8 | 32 | .3750 | .3696 | .3547 | .3513 | .3367 | .3750 | .3547 | .3581 | .3412 | .3458 |
| 7/16 | 28 | .4375 | .4313 | .4143 | .4107 | .3937 | .4375 | .4143 | .4179 | .3988 | .4041 |
| 1/2 | 28 | .5000 | .4938 | .4768 | .4731 | .4562 | .5000 | .4768 | .4805 | .4613 | .4666 |
| 9/16 | 24 | .5625 | .5559 | .5354 | .5314 | .5114 | .5625 | .5354 | .5394 | .5174 | .5235 |
| 5/8 | 24 | .6250 | .6184 | .5979 | .5938 | .5739 | .6250 | .5979 | .6020 | .5799 | .5860 |
| 3/4 | 20 | .7500 | .7428 | .7175 | .7129 | .6887 | .7500 | .7175 | .7221 | .6959 | .7027 |
| 7/8 | 20 | .8750 | .8678 | .8425 | .8378 | .8137 | .8750 | .8425 | .8472 | .8209 | .8277 |
| 1 | 20 | 1.0000 | .9928 | .9675 | .9627 | .9387 | 1.0000 | .9675 | .9723 | .9459 | .9527 |
| 1 1/8 | 18 | 1.1250 | 1.1168 | 1.0889 | 1.0837 | 1.0568 | 1.1250 | 1.0889 | 1.0941 | 1.0649 | 1.0724 |
| 1 1/4 | 18 | 1.2500 | 1.2418 | 1.2139 | 1.2086 | 1.1818 | 1.2500 | 1.2139 | 1.2192 | 1.1899 | 1.1974 |
| 1 1/2 | 18 | 1.5000 | 1.4918 | 1.4639 | 1.4584 | 1.4318 | 1.5000 | 1.4639 | 1.4694 | 1.4399 | 1.4474 |
| Class 3 Fit | | | | | | | | | | | |
| 1/4 | 32 | 0.2500 | 0.2446 | 0.2297 | 0.2275 | 0.2117 | 0.2500 | 0.2297 | 0.2319 | 0.2162 | 0.2208 |
| 5/16 | 32 | .3125 | .3071 | .2922 | .2899 | .2742 | .3125 | .2922 | .2345 | .2787 | .2833 |
| 3/8 | 32 | .3750 | .3696 | .3547 | .3523 | .3367 | .3750 | .3547 | .3571 | .3412 | .3458 |
| 7/16 | 28 | .4375 | .4313 | .4143 | .4118 | .3937 | .4375 | .4143 | .4168 | .3988 | .4041 |
| 1/2 | 28 | .5000 | .4938 | .4768 | .4742 | .4562 | .5000 | .4768 | .4794 | .4613 | .4666 |
| 9/16 | 24 | .5625 | .5559 | .5354 | .5326 | .5114 | .5625 | .5354 | .5382 | .5174 | .5235 |
| 5/8 | 24 | .6250 | .6184 | .5979 | .5950 | .5739 | .6250 | .5979 | .6008 | .5799 | .5860 |
| 3/4 | 20 | .7500 | .7428 | .7175 | .7143 | .6887 | .7500 | .7175 | .7207 | .6959 | .7027 |
| 7/8 | 20 | .8750 | .8678 | .8425 | .8392 | .8137 | .8750 | .8425 | .8458 | .8209 | .8277 |
| 1 | 20 | 1.0000 | .9928 | .9675 | .9641 | .9387 | 1.0000 | .9675 | .9709 | .9459 | .9527 |
| 1 1/8 | 18 | 1.1250 | 1.1168 | 1.0889 | 1.0853 | 1.0568 | 1.1250 | 1.0889 | 1.0925 | 1.0649 | 1.0724 |
| 1 1/4 | 18 | 1.2500 | 1.2418 | 1.2139 | 1.2102 | 1.1818 | 1.2500 | 1.2139 | 1.2176 | 1.1899 | 1.1974 |
| 1 1/2 | 18 | 1.5000 | 1.4918 | 1.4639 | 1.4601 | 1.4318 | 1.5000 | 1.4639 | 1.4677 | 1.4399 | 1.4474 |

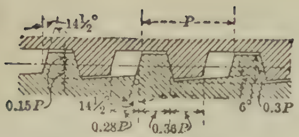


FIG. 6. Dardelet Thread

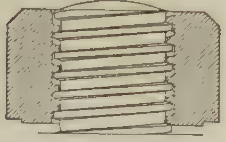


FIG. 7. Dardelet Thread, Unlocked

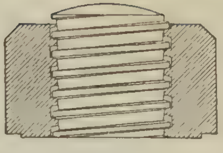


FIG. 8. Dardelet Thread, Locked

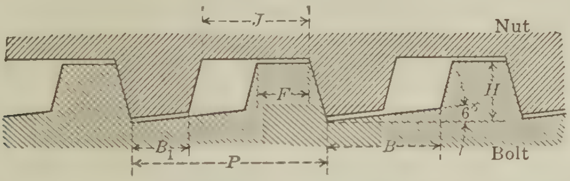
The action of the Dardelet thread as a locking device is shown in Figs. 7 and 8. In the unlocked position, the lower faces of the nut threads rest on the upper faces of the bolt threads as shown in Fig. 7. When the nut is screwed down into contact with the work, the bolt is drawn up until the lower faces of the bolt threads bear on the upper faces of the nut threads, as shown in Fig. 8. Prior to this contact, the inclined crests of the nut threads have come into contact with the tapered root section of the bolt thread. As the rotation of the nut is continued the inclined crests are compressed against the tapered roots of the thread with continually increasing pressure, until the load finally is taken by the sides of the thread coming in contact. Since vibration varies with the stress in the bolt, it affects only the load on the sides of the threads. It has no influence on the binding stress between the tapered root-sections and inclined crests. This form of thread, therefore, forms a positive threadlock.

Dardelet threads are made in a standard series, and in a fine-thread series which correspond in major diameters and pitch with the American National Standard thread. Elements of the Dardelet threads are given in Table 13, and limiting dimensions in Tables 14 and 15. Dimensions correspond to American National Standard Nuts.

Dardelet threads can be made by any of the processes commonly used for forming threads in machine tools. They also can be cut with thread chasers or taps, or they may be cold or hot rolled. Tests made at Columbia University and the U. S. Bureau of Standards show them to be stronger than American National Standard threads. See p. 9–43. Complete data for the forming of these threads, together with design data, records of tests, heat treatment, etc., are given in a 200-page handbook issued by the Dardelet Threadlock Corp., New York.

**Table 13.—Elements of the Dardelet Thread**

All dimensions in inches



| Number of threads, per inch, $N$ | Pitch, $P$ | External Thread | | | | | | | | Internal Thread | | | Axial movement between free and locked position |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Depth of thread, $H$ | Double depth of thread, $2H$ | Depth of abutment side | Double depth of abutment side | Projected width of binding surface, $B$ | Width of thread crest, $F$ | Width of ridge base | Clearance at crest | Projected width of binding surface, $B_1$ | Width of thread base | Width of surface between thread bases, $J$ | |
| 24 | 0.0417 | 0.0104 | 0.0208 | 0.0076 | 0.0152 | 0.0272 | 0.0098 | 0.0145 | 0.0010 | 0.0107 | 0.0153 | 0.0264 | 0.0166 |
| 20 | .0500 | .0135 | .0270 | .0103 | .0206 | .0308 | .0130 | .0192 | .0010 | .0133 | .0200 | .0300 | .0170 |
| 18 | .0556 | .0156 | .0312 | .0122 | .0244 | .0332 | .0151 | .0224 | .0010 | .0155 | .0233 | .0323 | .0172 |
| 16 | .0625 | .0188 | .0375 | .0149 | .0298 | .0361 | .0176 | .0264 | .0010 | .0181 | .0274 | .0351 | .0175 |
| 14 | .0714 | .0214 | .0428 | .0171 | .0342 | .0413 | .0201 | .0301 | .0015 | .0207 | .0312 | .0402 | .0200 |
| 13 | .0769 | .0231 | .0461 | .0184 | .0368 | .0444 | .0217 | .0324 | .0015 | .0223 | .0338 | .0432 | .0215 |
| 12 | .0833 | .0250 | .0500 | .0199 | .0398 | .0481 | .0235 | .0352 | .0015 | .0242 | .0365 | .0468 | .0233 |
| 11 | .0909 | .0273 | .0545 | .0217 | .0434 | .0526 | .0256 | .0384 | .0015 | .0264 | .0398 | .0511 | .0255 |
| 10 | .1000 | .0300 | .0600 | .0239 | .0478 | .0578 | .0282 | .0422 | .0015 | .0290 | .0438 | .0562 | .0280 |
| 9 | .1111 | .0333 | .0666 | .0266 | .0532 | .0642 | .0313 | .0469 | .0015 | .0322 | .0486 | .0624 | .0311 |
| 8 | .1250 | .0375 | .0750 | .0299 | .0598 | .0723 | .0353 | .0528 | .0015 | .0363 | .0547 | .0703 | .0350 |
| 7 | .1429 | .0429 | .0857 | .0342 | .0684 | .0826 | .0403 | .0603 | .0015 | .0414 | .0625 | .0804 | .0400 |
| 6 | .1667 | .0500 | .1000 | .0398 | .0796 | .0964 | .0470 | .0704 | .0015 | .0483 | .0730 | .0937 | .0466 |
| 5 | .2000 | .0600 | .1200 | .0478 | .0956 | .1156 | .0564 | .0844 | .0015 | .0580 | .0875 | .1125 | .0560 |
| 4 | .2500 | .0750 | .1500 | .0598 | .1196 | .1445 | .0705 | .1055 | .0015 | .0725 | .1094 | .1406 | .0700 |

## Table 14.—Dimensions and Limits of Dardelet External or Bolt Threads

All dimensions in inches

| Size | Threads per Inch | Standard Thread Series | | | | | | | | | | | | Threads per Inch | Fine Thread Series | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Major Thread Diameter | | Mean Thread Diam. | Helix Angle at Mean Thread Diam. | Taper Diameter | | | | | Average Taper Diam. Helix Angle | Width of Thread Base | | | Major Thread Diameter | | Mean Thread Diam. | Helix Angle at Mean Thread Diam. | Taper Diameter | | | | | Average Taper Diam. Helix Angle | Width of Thread Base | |
| | | Max. | Min. | | | Major | | Minor | | Average | | Max. | Min. | | Max. | Min. | | | Major | | Minor | | Average | | Max. | Min. |
| | | | | | | Max. | Min. | Max. | Min. | | | | | | | | | | Max. | Min. | Max. | Min. | | | | |
| No. 8 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24 | 0.1640 | 0.1625 | 0.1536 | 4° 56′ | 0.1489 | 0.1474 | 0.1432 | 0.1417 | 0.1461 | 5° 11′ | 0.0145 | 0.0135 |
| 10 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24 | .1875 | .1860 | .1771 | 4° 17′ | .1724 | .1709 | .1667 | .1652 | .1696 | 4° 28′ | .0145 | .0135 |
| 12 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24 | .2160 | .2145 | .2056 | 3° 41′ | .2009 | .1994 | .1952 | .1937 | .1981 | 3° 50′ | .0145 | .0135 |
| 14 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 20 | .2420 | .2405 | .2285 | 3° 59′ | .2214 | .2199 | .2150 | .2135 | .2182 | 4° 11′ | .0192 | .0182 |
| 1/4 | 16 | 0.2500 | 0.2480 | 0.2313 | 4° 55′ | 0.2201 | 0.2186 | 0.2125 | 0.2110 | 0.2163 | 5° 15′ | 0.0264 | 0.0254 | 20 | .2500 | .2485 | .2365 | 3° 51′ | .2294 | .2279 | .2230 | .2215 | .2262 | 4° 02′ | .0192 | .0182 |
| 5/16 | 14 | .3125 | .3095 | .2911 | 4° 28′ | .2784 | .2764 | .2697 | .2677 | .2741 | 4° 44′ | .0301 | .0291 | 18 | .3125 | .3110 | .2969 | 3° 25′ | .2882 | .2867 | .2813 | .2798 | .2848 | 3° 33′ | .0224 | .0214 |
| 3/8 | 12 | .3750 | .3720 | .3500 | 4° 20′ | .3352 | .3332 | .3250 | .3230 | .3301 | 4° 35′ | .0352 | .0342 | 16 | .3750 | .3730 | .3563 | 3° 12′ | .3452 | .3437 | .3375 | .3360 | .3414 | 3° 20′ | .0264 | .0254 |
| 7/16 | 11 | .4375 | .4345 | .4103 | 4° 02′ | .3941 | .3921 | .3830 | .3810 | .3885 | 4° 16′ | .0384 | .0369 | 16 | .4375 | .4355 | .4188 | 2° 43′ | .4077 | .4062 | .4000 | .3985 | .4038 | 2° 49′ | .0264 | .0254 |
| 1/2 | 10 | .5000 | .4960 | .4700 | 3° 53′ | .4522 | .4502 | .4400 | .4380 | .4461 | 4° 04′ | .0422 | .0402 | 16 | .5000 | .4980 | .4813 | 2° 22′ | .4702 | .4687 | .4625 | .4610 | .4664 | 2° 27′ | .0264 | .0254 |
| 9/16 | 9 | .5625 | .5585 | .5292 | 3° 49′ | .5093 | .5073 | .4959 | .4939 | .5026 | 4° 02′ | .0469 | .0449 | 14 | .5625 | .5595 | .5411 | 2° 24′ | .5283 | .5263 | .5197 | .5177 | .5240 | 2° 29′ | .0301 | .0291 |
| 5/8 | 8 | .6250 | .6210 | .5875 | 3° 53′ | .5651 | .5621 | .5500 | .5470 | .5576 | 4° 05′ | .0528 | .0508 | 13 | .6250 | .6220 | .6019 | 2° 19′ | .5882 | .5862 | .5788 | .5768 | .5835 | 2° 24′ | .0325 | .0315 |
| 11/16 | 8 | .6875 | .6835 | .6500 | 3° 30′ | .6276 | .6246 | .6125 | .6095 | .6201 | 3° 40′ | .0528 | .0508 | 12 | .6875 | .6845 | .6625 | 2° 17′ | .6477 | .6457 | .6375 | .6355 | .6426 | 2° 22′ | .0352 | .0342 |
| 3/4 | 8 | .7500 | .7460 | .7125 | 3° 12′ | .6901 | .6871 | .6750 | .6720 | .6826 | 3° 20′ | .0528 | .0508 | 12 | .7500 | .7470 | .7250 | 2° 06′ | .7102 | .7082 | .7000 | .6980 | .7051 | 2° 09′ | .0352 | .0342 |
| 13/16 | 7 | .8125 | .8085 | .7697 | 3° 23′ | .7441 | .7306 | .7268 | .7233 | .7355 | 3° 32′ | .0603 | .0583 | 12 | .8125 | .8095 | .7875 | 1° 56′ | .7727 | .7707 | .7625 | .7605 | .7676 | 1° 59′ | .0352 | .0342 |
| 7/8 | 7 | .8750 | .8710 | .8322 | 3° 08′ | .8066 | .8031 | .7893 | .7858 | .7980 | 3° 16′ | .0603 | .0583 | 11 | .8750 | .8720 | .8478 | 1° 57′ | .8316 | .8296 | .8205 | .8185 | .8261 | 2° 01′ | .0384 | .0369 |
| 15/16 | 6 | .9375 | .9325 | .8875 | 3° 25′ | .8579 | .8539 | .8375 | .8335 | .8477 | 3° 35′ | .0704 | .0684 | 10 | .9375 | .9335 | .9075 | 2° 00′ | .8897 | .8877 | .8775 | .8755 | .8836 | 2° 04′ | .0422 | .0402 |
| 1 | 6 | 1.0000 | .9950 | .9500 | 3° 12′ | .9204 | .9164 | .9000 | .8960 | .9102 | 3° 20′ | .0704 | .0684 | 10 | 1.0000 | .9960 | .9700 | 1° 52′ | .9522 | .9502 | .9400 | .9380 | .9461 | 1° 55′ | .0422 | .0402 |
| 1 1/16 | 6 | 1.0625 | 1.0575 | 1.0125 | 3° 00′ | .9829 | .9789 | .9625 | .9585 | .9727 | 3° 07′ | .0704 | .0684 | 9 | 1.0625 | 1.0585 | 1.0292 | 1° 58′ | 1.0093 | 1.0073 | .9959 | .9939 | 1.0026 | 2° 01′ | .0469 | .0449 |
| 1 1/8 | 5 | 1.1250 | 1.1200 | 1.0650 | 3° 25′ | 1.0294 | 1.0254 | 1.0050 | 1.0010 | 1.0172 | 3° 35′ | .0844 | .0824 | 8 | 1.1250 | 1.1210 | 1.0875 | 2° 06′ | 1.0652 | 1.0622 | 1.0500 | 1.0470 | 1.0576 | 2° 09′ | .0528 | .0508 |
| 1 3/16 | 5 | 1.1875 | 1.1825 | 1.1275 | 3° 14′ | 1.0919 | 1.0879 | 1.0675 | 1.0635 | 1.0797 | 3° 23′ | .0844 | .0824 | 8 | 1.1875 | 1.1835 | 1.1500 | 1° 59′ | 1.1277 | 1.1247 | 1.1125 | 1.1095 | 1.1201 | 2° 02′ | .0528 | .0508 |
| 1 1/4 | 5 | 1.2500 | 1.2450 | 1.1900 | 3° 04′ | 1.1544 | 1.1504 | 1.1300 | 1.1260 | 1.1422 | 3° 12′ | .0844 | .0824 | 8 | 1.2500 | 1.2460 | 1.2125 | 1° 53′ | 1.1902 | 1.1872 | 1.1750 | 1.1720 | 1.1826 | 1° 55′ | .0528 | .0508 |
| 1 5/16 | 5 | 1.3125 | 1.3075 | 1.2525 | 2° 55′ | 1.2169 | 1.2129 | 1.1925 | 1.1885 | 1.2047 | 3° 01′ | .0844 | .0824 | 8 | 1.3125 | 1.3085 | 1.2750 | 1° 47′ | 1.2527 | 1.2497 | 1.2375 | 1.2345 | 1.2451 | 1° 50′ | .0528 | .0508 |
| 1 3/8 | 5 | 1.3750 | 1.3700 | 1.3150 | 2° 46′ | 1.2794 | 1.2754 | 1.2550 | 1.2510 | 1.2672 | 2° 54′ | .0844 | .0824 | 7 | 1.3750 | 1.3710 | 1.3322 | 1° 57′ | 1.3066 | 1.3046 | 1.2893 | 1.2858 | 1.2980 | 2° 00′ | .0603 | .0583 |
| 1 7/16 | 5 | 1.4375 | 1.4325 | 1.3775 | 2° 39′ | 1.3419 | 1.3379 | 1.3175 | 1.3135 | 1.3297 | 2° 45′ | .0844 | .0824 | 7 | 1.4375 | 1.4335 | 1.3947 | 1° 52′ | 1.3691 | 1.3671 | 1.3518 | 1.3483 | 1.3605 | 1° 55′ | .0603 | .0583 |
| 1 1/2 | 4 | 1.5000 | 1.4950 | 1.4250 | 3° 12′ | 1.3804 | 1.3764 | 1.3500 | 1.3460 | 1.3652 | 3° 20′ | .1055 | .1035 | 6 | 1.5000 | 1.4950 | 1.4500 | 2° 06′ | 1.4204 | 1.4184 | 1.4000 | 1.3960 | 1.4102 | 2° 10′ | .0704 | .0684 |
| 1 5/8 | 4 | 1.6250 | 1.6200 | 1.5500 | 2° 56′ | 1.5054 | 1.5014 | 1.4750 | 1.4710 | 1.4902 | 3° 03′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 1 3/4 | 4 | 1.7500 | 1.7450 | 1.6750 | 2° 43′ | 1.6304 | 1.6264 | 1.6000 | 1.5960 | 1.6152 | 2° 49′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 1 7/8 | 4 | 1.8750 | 1.8700 | 1.8000 | 2° 32′ | 1.7554 | 1.7514 | 1.7250 | 1.7210 | 1.7402 | 2° 37′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 | 4 | 2.0000 | 1.9950 | 1.9250 | 2° 22′ | 1.8804 | 1.8764 | 1.8500 | 1.8460 | 1.8652 | 2° 27′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 1/4 | 4 | 2.2500 | 2.2450 | 2.1750 | 2° 06′ | 2.1304 | 2.1264 | 2.1000 | 2.0960 | 2.1152 | 2° 09′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 1/2 | 4 | 2.5000 | 2.4950 | 2.4250 | 1° 53′ | 2.3804 | 2.3764 | 2.3500 | 2.3460 | 2.3652 | 1° 56′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 3/4 | 4 | 2.7500 | 2.7450 | 2.6750 | 1° 42′ | 2.6304 | 2.6264 | 2.6000 | 2.5960 | 2.6152 | 1° 45′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 3 | 4 | 3.0000 | 2.9950 | 2.9250 | 1° 33′ | 2.8804 | 2.8764 | 2.8500 | 2.8460 | 2.8652 | 1° 36′ | .1055 | .1035 | ... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

### Table 15.—Dimensions and Limits of Dardelet Internal or Nut Threads

All dimensions in inches

| Size | Standard Thread Series | | | | | | | | | | | Fine Thread Series | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Threads per Inch | Major Thread Diameter | | Mean Thread Diameter | Helix Angle of Thread at Mean Diam. | Taper Diameter | | | | Width of Thread Base | | Threads per Inch | Major Thread Diameter | | Mean Thread Diameter | Helix Angle of Thread at Mean Diam. | Taper Diameter | | | | Width of Thread Base | |
| | | | | | | Major | | Minor | | | | | | | | | Major | | Minor | | | |
| | | Max. | Min. | | | Max. | Min. | Max. | Min. | Max. | Min. | | Max. | Min. | | | Max. | Min. | Max. | Min. | Max. | Min. |
| No. 8 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24 | 0.1680 | 0.1660 | 0.1546 | 4° 54′ | 0.1464 | 0.1454 | 0.1442 | 0.1432 | 0.0153 | 0.0143 |
| 10 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24 | .1915 | .1895 | .1781 | 4° 16′ | .1699 | .1689 | .1677 | .1667 | .0153 | .0143 |
| 12 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24 | .2200 | .2180 | .2066 | 3° 40′ | .1984 | .1974 | .1962 | .1952 | .0153 | .0143 |
| 14 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 20 | .2460 | .2440 | .2295 | 3° 58′ | .2188 | .2178 | .2160 | .2150 | .0200 | .0190 |
| 1/4 | 16 | 0.2540 | 0.2520 | 0.2323 | 4° 54′ | 0.2172 | 0.2162 | 0.2135 | 0.2125 | 0.0274 | 0.0264 | 20 | .2540 | .2520 | .2375 | 3° 50′ | .2268 | .2258 | .2240 | .2230 | .0200 | .0190 |
| 5/16 | 14 | .3175 | .3155 | .2926 | 4° 27′ | .2749 | .2739 | .2707 | .2697 | .0312 | .0302 | 18 | .3165 | .3145 | .2979 | 3° 24′ | .2854 | .2844 | .2823 | .2813 | .0233 | .0223 |
| 3/8 | 12 | .3800 | .3780 | .3515 | 4° 19′ | .3309 | .3299 | .3260 | .3250 | .0365 | .0355 | 16 | .3790 | .3770 | .3573 | 3° 11′ | .3422 | .3412 | .3385 | .3375 | .0274 | .0264 |
| 7/16 | 11 | .4435 | .4405 | .4117 | 4° 01′ | .3894 | .3884 | .3840 | .3830 | .0398 | .0383 | 16 | .4415 | .4395 | .4198 | 2° 43′ | .4047 | .4037 | .4010 | .4000 | .0274 | .0264 |
| 1/2 | 10 | .5070 | .5030 | .4715 | 3° 51′ | .4469 | .4459 | .4410 | .4400 | .0438 | .0418 | 16 | .5040 | .5020 | .4823 | 2° 21′ | .4672 | .4662 | .4635 | .4625 | .0274 | .0264 |
| 9/16 | 9 | .5695 | .5655 | .5307 | 3° 49′ | .5035 | .5025 | .4969 | .4959 | .0486 | 0466 | 14 | .5675 | .5655 | .5426 | 2° 24′ | .5249 | .5239 | .5207 | .5197 | .0312 | .0302 |
| 5/8 | 8 | .6320 | .6280 | .5890 | 3° 52′ | .5589 | .5574 | .5515 | .5500 | .0547 | .0527 | 13 | .6300 | .6280 | .6034 | 2° 19′ | .5844 | .5834 | .5798 | .5788 | .0338 | .0328 |
| 11/16 | 8 | .6945 | .6905 | .6515 | 3° 30′ | .6214 | .6199 | .6140 | .6125 | .0547 | .0527 | 12 | .6925 | .6905 | .6640 | 2° 17′ | .6434 | .6424 | .6385 | .6375 | .0365 | .0355 |
| 3/4 | 8 | .7570 | .7530 | .7140 | 3° 11′ | .6839 | .6824 | .6765 | .6750 | .0547 | .0527 | 12 | .7550 | .7530 | .7265 | 2° 05′ | .7059 | .7049 | .7010 | .7000 | .0365 | .0355 |
| 13/16 | 7 | .8195 | .8155 | .7712 | 3° 23′ | .7368 | .7353 | .7283 | .7268 | .0625 | .0605 | 12 | .8175 | .8155 | .7890 | 1° 55′ | .7684 | .7674 | .7635 | .7625 | .0365 | .0355 |
| 7/8 | 7 | .8820 | .8780 | .8337 | 3° 07′ | .7993 | .7978 | .7908 | .7893 | .0625 | .0605 | 11 | .8810 | .8780 | .8493 | 1° 57′ | .8269 | .8259 | .8215 | .8205 | .0398 | .0383 |
| 15/16 | 6 | .9445 | .9405 | .8890 | 3° 25′ | .8494 | .8474 | .8395 | .8375 | .0730 | .0710 | 10 | .9445 | .9405 | .9090 | 2° 00′ | .8844 | .8834 | .8785 | .8775 | .0438 | .0418 |
| 1 | 6 | 1.0070 | 1.0030 | .9515 | 3° 12′ | .9119 | .9099 | .9020 | .9000 | .0730 | .0710 | 10 | 1.0070 | 1.0030 | .9715 | 1° 52′ | .9469 | .9459 | .9410 | .9400 | .0438 | .0418 |
| 1 1/16 | 6 | 1.0695 | 1.0655 | 1.0140 | 3° 00′ | .9744 | .9724 | .9645 | .9625 | .0730 | .0710 | 9 | 1.0695 | 1.0665 | 1.0307 | 1° 58′ | 1.0035 | 1.0025 | .9969 | .9959 | .0486 | .0466 |
| 1 1/8 | 5 | 1.1320 | 1.1280 | 1.0665 | 3° 25′ | 1.0189 | 1.0169 | 1.0070 | 1.0050 | .0875 | .0855 | 8 | 1.1320 | 1.1280 | 1.0890 | 2° 05′ | 1.0589 | 1.0574 | 1.0515 | 1.0500 | .0547 | .0527 |
| 1 3/16 | 5 | 1.1945 | 1.1905 | 1.1290 | 3° 14′ | 1.0814 | 1.0794 | 1.0695 | 1.0675 | .0875 | .0855 | 8 | 1.1945 | 1.1905 | 1.1515 | 1° 59′ | 1.1214 | 1.1199 | 1.1140 | 1.1125 | .0547 | .0527 |
| 1 1/4 | 5 | 1.2570 | 1.2530 | 1.1915 | 3° 03′ | 1.1439 | 1.1419 | 1.1320 | 1.1300 | .0875 | .0855 | 8 | 1.2570 | 1.2530 | 1.2140 | 1° 53′ | 1.1839 | 1.1824 | 1.1765 | 1.1750 | .0547 | .0527 |
| 1 5/16 | 5 | 1.3195 | 1.3155 | 1.2540 | 2° 54′ | 1.2064 | 1.2044 | 1.1945 | 1.1925 | .0875 | .0855 | 8 | 1.3195 | 1.3155 | 1.2765 | 1° 47′ | 1.2464 | 1.2449 | 1.2390 | 1.2375 | .0547 | .0527 |
| 1 3/8 | 5 | 1.3820 | 1.3780 | 1.3165 | 2° 46′ | 1.2689 | 1.2669 | 1.2570 | 1.2550 | .0875 | .0855 | 7 | 1.3820 | 1.3780 | 1.3337 | 1° 57′ | 1.2993 | 1.2978 | 1.2908 | 1.2893 | .0625 | .0605 |
| 1 7/16 | 5 | 1.4445 | 1.4405 | 1.3790 | 2° 39′ | 1.3314 | 1.3294 | 1.3195 | 1.3175 | 0875 | .0855 | 7 | 1.4445 | 1.4405 | 1.3962 | 1° 52′ | 1.3618 | 1.3603 | 1.3533 | 1.3518 | .0625 | .0605 |
| 1 1/2 | 4 | 1.5070 | 1.5030 | 1.4265 | 3° 11′ | 1.3669 | 1.3649 | 1.3520 | 1.3500 | .1094 | .1074 | 6 | 1.5070 | 1.5030 | 1.4515 | 2° 06′ | 1.4119 | 1.4099 | 1.4020 | 1.4000 | .0730 | .0710 |
| 1 5/8 | 4 | 1.6320 | 1.6280 | 1.5515 | 2° 56′ | 1.4919 | 1.4899 | 1.4770 | 1.4750 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 1 3/4 | 4 | 1.7570 | 1.7530 | 1.6765 | 2° 43′ | 1.6169 | 1.6149 | 1.6020 | 1.6000 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 1 7/8 | 4 | 1.8820 | 1.8780 | 1.8015 | 2° 32′ | 1.7419 | 1.7399 | 1.7270 | 1.7250 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 | 4 | 2.0070 | 2.0030 | 1.9265 | 2° 22′ | 1.8669 | 1.8649 | 1.8520 | 1.8500 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 1/4 | 4 | 2.2570 | 2.2530 | 2.1765 | 2° 06′ | 2.1169 | 2.1149 | 2.1020 | 2.1000 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 1/2 | 4 | 2.5070 | 2.5030 | 2.4265 | 1° 53′ | 2.3669 | 2.3649 | 2.3520 | 2.3500 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 3/4 | 4 | 2.7570 | 2.7530 | 2.6765 | 1° 42′ | 2.6169 | 2.6149 | 2.6020 | 2.6000 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 3 | 4 | 3.0070 | 3.0030 | 2.9265 | 1° 33′ | 2.8669 | 2.8649 | 2.8520 | 2.8500 | .1094 | .1074 | ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

**WHITWORTH STANDARD THREADS.**—In the Whitworth or English system the angle of the thread is 55 deg. The point and the root of the thread are rounded to a radius of $r = 0.1373 \times$ pitch. The depth of the thread is $d = 0.6403 \times$ pitch. See Fig. 9 and Table 16.

**Table 16.—Proportions of Whitworth Triangular Screw Threads, Bolts, and Nuts**

| Diam., in. | Threads per in. | Pitch, in. | Diam. at Root of Thread, in. | Area of Root of Thread, in. | Thickness of Head, in. | Thickness, of Nut, in. | Width | |
|---|---|---|---|---|---|---|---|---|
| | | | | | | | Across Flats, in. | Across Corners, in. |
| 1/4 | 20 | 0.05 | 0.186 | 0.027 | 7/32 | 1/4 | 17/32 | 39/64 |
| 5/16 | 18 | .0555 | .241 | .046 | 17/64 | 5/16 | 39/64 | 11/16 |
| 3/8 | 16 | .0625 | .295 | .068 | 21/64 | 3/8 | 45/64 | 13/16 |
| 7/16 | 14 | .0714 | .346 | .094 | 3/8 | 7/16 | 53/64 | 61/64 |
| 1/2 | 12 | .0833 | .393 | .121 | 7/16 | 1/2 | 59/64 | 1 1/16 |
| 5/8 | 11 | .0909 | .509 | .204 | 35/64 | 5/8 | 1 3/32 | 1 17/64 |
| 3/4 | 10 | .100 | .622 | .304 | 21/32 | 3/4 | 1 19/64 | 1 1/2 |
| 7/8 | 9 | .111 | .733 | .422 | 49/64 | 7/8 | 1 31/64 | 1 45/64 |
| 1 | 8 | .125 | .840 | .554 | 7/8 | 1 | 1 43/64 | 1 61/64 |
| 1 1/8 | 7 | .143 | .942 | .697 | 63/64 | 1 1/8 | 1 55/64 | 2 5/32 |
| 1 1/4 | 7 | .143 | 1.067 | .894 | 1 3/32 | 1 1/4 | 2 3/64 | 2 23/64 |
| 1 3/8 | 6 | .167 | 1.161 | 1.058 | 1 13/64 | 1 3/8 | 2 7/32 | 2 35/64 |
| 1 1/2 | 6 | .167 | 1.286 | 1.299 | 1 5/16 | 1 1/2 | 2 13/32 | 2 25/32 |
| 1 5/8 | 5 | .200 | 1.369 | 1.472 | 1 27/64 | 1 5/8 | 2 37/64 | 2 31/32 |
| 1 3/4 | 5 | .200 | 1.494 | 1.753 | 1 17/32 | 1 3/4 | 2 3/4 | 3 3/16 |
| 1 7/8 | 4 1/2 | .222 | 1.590 | 1.986 | 1 41/64 | 1 7/8 | 3 1/64 | 3 31/64 |
| 2 | 4 1/2 | .222 | 1.715 | 2.311 | 1 3/4 | 2 | 3 5/32 | 3 5/8 |
| 2 1/8 | 4 1/2 | .222 | 1.840 | 2.659 | 1 55/64 | 2 1/8 | 3 11/32 | 3 7/8 |
| 2 1/4 | 4 | .250 | 1.930 | 2.926 | 1 31/32 | 2 1/4 | 3 35/64 | 4 3/32 |
| 2 1/2 | 4 | .250 | 2.180 | 3.733 | 2 3/16 | 2 1/2 | 3 57/64 | 4 31/64 |
| 2 3/4 | 3 1/2 | .286 | 2.384 | 4.464 | 2 13/32 | 2 3/4 | 4 3/16 | 4 53/64 |
| 3 | 3 1/2 | .286 | 2.634 | 5.450 | 2 5/8 | 3 | 4 17/32 | 5 15/64 |
| 3 1/4 | 3 1/4 | .308 | 2.855 | 6.402 | | | | |
| 3 1/2 | 3 1/4 | .308 | 3.105 | 7.563 | | | | |
| 3 3/4 | 3 | .333 | 3.323 | 8.673 | | | | |
| 4 | 3 | .333 | 3.573 | 10.027 | | | | |
| 4 1/4 | 2 7/8 | .348 | 3.804 | 11.365 | | | | |
| 4 1/2 | 2 7/8 | .348 | 4.054 | 12.908 | | | | |
| 4 3/4 | 2 3/4 | .364 | 4.284 | 14.404 | | | | |
| 5 | 2 3/4 | .364 | 4.534 | 16.146 | | | | |
| 5 1/4 | 2 5/8 | .381 | 4.762 | 17.810 | | | | |
| 5 1/2 | 2 5/8 | .381 | 5.012 | 19.72 | | | | |
| 5 3/4 | 2 1/2 | .400 | 5.239 | 21.57 | | | | |
| 6 | 2 1/2 | .400 | 5.489 | 23.64 | | | | |

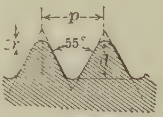


FIG. 9. Whitworth Thread

**Table 17.—Metric Screw Threads.** (French and International Standard)

Dimensions in millimeters

| Outside Diam. | Pitch | | Pitch Diam. | Root Diam. | Tap Drill | Outside Diam. | Pitch | | Pitch Diam. | Root Diam. | Tap Drill |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | French | International | | | | | French | International | | | |
| 6 | 1.0 | 1.0 | 5.35 | 4.70 | 4.80 | 33 | ....... | 3.5 | 30.73 | 28.45 | 28.5 |
| 7 | 1.0 | 1.0 | 6.35 | 5.70 | 5.80 | 34 | 3.5 | ....... | 31.73 | 29.45 | 29.5 |
| 8 | 1.0 | ....... | 7.35 | 6.70 | 6.80 | 36 | 4.0 | 4.0 | 33.40 | 30.80 | 31 |
| 8 | ....... | 1.25 | 7.19 | 6.38 | 6.40 | 38 | 4.0 | ....... | 35.40 | 32.80 | 33 |
| 9 | 1.0 | ....... | 8.35 | 7.70 | 7.80 | 39 | ....... | 4.0 | 36.40 | 33.80 | 34 |
| 9 | ....... | 1.25 | 8.19 | 7.38 | 7.40 | 40 | 4.0 | ....... | 37.40 | 34.80 | 35 |
| 10 | 1.5 | 1.5 | 9.02 | 8.05 | 8.10 | 42 | 4.5 | 4.5 | 39.08 | 36.15 | 36.5 |
| 11 | ....... | 1.5 | 10.02 | 9.05 | 9.10 | 44 | 4.5 | ....... | 41.08 | 38.15 | 38.5 |
| 12 | 1.5 | ....... | 11.03 | 10.05 | 10.50 | 45 | ....... | 4.5 | 42.08 | 39.15 | 39.5 |
| 12 | ....... | 1.75 | 10.86 | 9.73 | 9.80 | 46 | 4.5 | ....... | 43.08 | 40.15 | 40.5 |
| 14 | 2.0 | 2.0 | 12.70 | 11.40 | 11.50 | 48 | 5.0 | 5.0 | 44.75 | 41.50 | 42 |
| 16 | 2.0 | 2.0 | 14.70 | 13.40 | 13.50 | 50 | 5.0 | ....... | 46.75 | 43.50 | 44 |
| 18 | 2.5 | 2.5 | 16.38 | 14.75 | 15.00 | 52 | ....... | 5.0 | 48.75 | 45.50 | 46 |
| 20 | 2.5 | 2.5 | 18.38 | 16.75 | 17.00 | 56 | ....... | 5.5 | 52.43 | 48.85 | 49 |
| 22 | 2.5 | 2.5 | 20.38 | 18.75 | 19.00 | 60 | ....... | 5.5 | 56.43 | 52.85 | 53 |
| 24 | 3.0 | 3.0 | 22.05 | 20.10 | 20.5 | 64 | ....... | 6.0 | 60.10 | 56.20 | 56.5 |
| 26 | 3.0 | ....... | 24.05 | 22.10 | 22.5 | 68 | ....... | 6.0 | 64.10 | 60.20 | 60.5 |
| 27 | ....... | 3.0 | 25.05 | 23.10 | 23.5 | 72 | ....... | 6.5 | 67.78 | 63.56 | 64 |
| 28 | 3.0 | ....... | 26.05 | 24.10 | 24.5 | 76 | ....... | 6.5 | 71.78 | 67.56 | 68 |
| 30 | 3.5 | 3.5 | 27.73 | 25.45 | 25.5 | 80 | ....... | 7.0 | 75.45 | 70.91 | 71 |
| 32 | 3.5 | ....... | 29.73 | 27.45 | 27.5 | | | | | | |

**FRENCH AND INTERNATIONAL STANDARD THREADS.**—The formulas for French, International, and American standard threads are all the same, the only difference being that the pitches of the first two are stated in metric measure and in application give somewhat finer values for corresponding diameters, than does the latter system. While the clearance at the bottom of the thread shall not exceed $^1/_{16}$ of the height of the original triangle according to both the International and the French standards, the former leaves the shape of the bottom of the thread to the manufacturer, whereas the French standard makes the bottom of the thread flat. See Table 17.

**BRITISH ASSOCIATION STANDARD THREAD.**—The angle between the threads is 47 $^1/_2$ deg. The depth of the thread $d = 0.6 \times$ pitch. The tops and bottoms of the threads are rounded with a radius of $r = {^2/_{11}} \times p$. See Fig. 10 and Table 18.

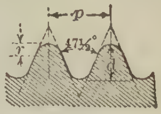


Fig. 10. British Association Thread

**THREADLOCKS.**—The most common form of threadlock is a jam nut, screwed tightly against the first nut. This is not a particularly satisfactory lock, as the two nuts frequently act as a single nut and both are loosened by vibration. More positive means of locking are shown in Fig. 11. The Dardelet thread (see p. 9–14) is practically vibration proof.

Spring or lock washers are quite commonly used to prevent nuts from loosening, and

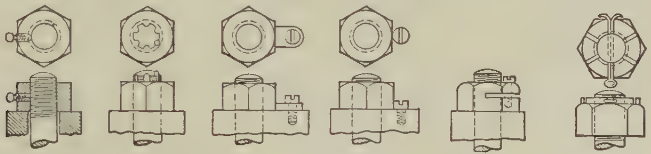


Fig. 11. Methods of Locking Nuts

**Table 18.—British Association Screw Threads**

| No. | Outside Diam., in. | Pitch, in. | Diam. at Root of Thread, in. | No. | Outside Diam., in. | Pitch, in. | Diam. at Root of Thread, in. |
|---|---|---|---|---|---|---|---|
| 0 | 0.236 | 0.039 | 0.189 | 13 | 0.047 | 0.0098 | 0.0354 |
| 1 | .209 | .035 | .166 | 14 | .039 | .0091 | .0283 |
| 2 | .185 | .032 | .147 | 15 | .035 | .0083 | .0256 |
| 3 | .161 | .029 | .127 | 16 | .031 | .0075 | .0220 |
| 4 | .142 | .026 | .111 | 17 | .028 | .0067 | .0197 |
| 5 | .126 | .023 | .098 | 18 | .024 | .0059 | .0173 |
| 6 | .110 | .021 | .085 | 19 | .021 | .0055 | .0146 |
| 7 | .098 | .0189 | .076 | 20 | .019 | .0047 | .0134 |
| 8 | .087 | .0169 | .066 | 21 | .017 | .0043 | .0114 |
| 9 | .075 | .0154 | .056 | 22 | .015 | .0039 | .0098 |
| 10 | .067 | .0138 | .050 | 23 | .013 | .0035 | .0087 |
| 11 | .059 | .0122 | .044 | 24 | .011 | .0031 | .0075 |
| 12 | .051 | .0110 | .038 | 25 | .010 | .0028 | .0067 |

**Table 19.—British Standard Fine Screw Threads. (B.S.F.)**

| Diam., in. | Threads per in. | Pitch, in. | Diam., Root of Thread, in. | Area, of Root Thread, sq. in. | Diam., in. | Threads per in. | Pitch, in. | Diam., Root of Thread, in. | Area, Root of Thread, sq. in. |
|---|---|---|---|---|---|---|---|---|---|
| 1/4 | 25 | 0.0400 | 0.1988 | 0.0310 | 1 1/2 | 8 | 0.1250 | 1.3399 | 1.4100 |
| 0.270 | 25 | .0400 | .2188 | .0376 | 1 5/8 | 8 | .1250 | 1.4649 | 1.6854 |
| 5/16 | 22 | .0455 | .2543 | .0508 | 1 3/4 | 7 | .1429 | 1.5670 | 1.9285 |
| 3/8 | 20 | .0500 | .3110 | .0760 | 2 | 7 | .1429 | 1.8170 | 2.5930 |
| 7/16 | 18 | .0556 | .3664 | .1054 | 2 1/4 | 6 | .1667 | 2.0366 | 3.2576 |
| 1/2 | 16 | .0625 | .4200 | .1385 | 2 1/2 | 6 | .1667 | 2.2866 | 4.1065 |
| 9/16 | 16 | .0625 | .4825 | .1828 | 2 3/4 | 6 | .1667 | 2.5366 | 5.0535 |
| 5/8 | 14 | .0714 | .5335 | .2235 | 3 | 5 | .2000 | 2.7439 | 5.9133 |
| 11/16 | 14 | .0714 | .5960 | .2790 | 3 1/4 | 5 | .2000 | 2.9939 | 7.0399 |
| 3/4 | 12 | .0833 | .6433 | .3250 | 3 1/2 | 4.5 | .2222 | 3.2154 | 8.1201 |
| 13/16 | 12 | .0833 | .7058 | .3913 | 3 3/4 | 4.5 | .2222 | 3.4654 | 9.4319 |
| 7/8 | 11 | .0909 | .7586 | .4520 | 4 | 4.5 | .2222 | 3.7154 | 10.8418 |
| 1 | 10 | .1000 | .8719 | .5971 | 4 1/2 | 4 | .2500 | 4.1798 | 13.7215 |
| 1 1/8 | 9 | .1111 | .9827 | .7585 | 5 | 4 | .2500 | 4.6798 | 17.2006 |
| 1 1/4 | 9 | .1111 | 1.1077 | .9637 | 5 1/2 | 3.5 | .2857 | 5.1341 | 20.7023 |
| 1 3/8 | 8 | .1250 | 1.2149 | 1.1593 | 6 | 3.5 | .2857 | 5.6341 | 24.9310 |

for ordinary service are quite satisfactory. The S.A.E. standard lock washer is shown in Fig. 12. Dimensions are given in Table 20. Another form of lock washer that has proved successful is that shown in Fig. 13, made by the Shakeproof Lock Washer Co., Chicago. The teeth on the internal or external circumference are twisted out of the plane of the washer. The sharp edges of the teeth bite into both the surface of the nut and of the work and resist any tendency of the nut to unscrew under the influence of vibration. Dimensions of stock sizes are given in Table 21. These washers are made in steel and bronze, and also are made with both internal and external teeth and with countersunk teeth for use with countersunk-head screws.

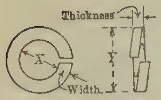


Fig. 12. S.A.E. Standard Lock Washer

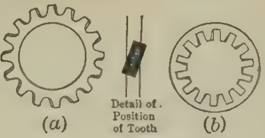


Fig. 13. Shakeproof Lock Washer

**TAPS.**—Tables 22 and 23 show the form and dimensions of machine screw taps, hand taps, tapper taps and nut taps with both cut and ground thread. Cut thread taps should produce holes within Class 2 screw thread tolerances. Ground taps should produce holes within Class 3 tolerances. See Table 4. The form of thread and the dimensions of thread are given in Table 3. The tolerances for the various classes and sizes of taps are given in Table 24.

**Table 20.—Dimensions of S.A.E. Standard Lock Washers**

All dimensions in inches. See Fig. 12.

| Nominal Size | Inside Diameter | | Outside Diameter, Y | | | Washer Sections | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Regular | | Light | | Heavy | |
| | Clearance, min. | X, min. | Regular | Light | Heavy | Width | Thick | Width | Thick | Width | Thick |
| No. 2 (0.086) | 0.003 | 0.089 | 0.151 | 0.133 | 0.151 | 1/32 | 0.022 | 0.022 | 0.022 | 1/32 | 1/32 |
| 4 (0.112) | .003 | .115 | .177 | .177 | .209 | 1/32 | 1/32 | 1/32 | 0.022 | 3/64 | 1/32 |
| 6 (0.138) | .005 | .143 | .237 | .205 | .237 | 3/64 | 1/32 | 1/32 | 1/32 | 3/64 | 3/64 |
| 8 (0.164) | .005 | .169 | .263 | .263 | .294 | 3/64 | 3/64 | 3/64 | 1/32 | 1/16 | 3/64 |
| 10 (0.190) | .010 | .200 | .325 | .294 | .325 | 1/16 | 3/64 | 3/64 | 3/64 | 1/16 | 1/16 |
| 12 (0.216) | .010 | .226 | .351 | .351 | .382 | 1/16 | 1/16 | 1/16 | 3/64 | 5/64 | 1/16 |
| 1/4 | .015 | .265 | ....... | .453 | ....... | 3/32 | 1/16 | 3/32 | 3/64 | 3/32 | 5/64 |
| 5/16 | .015 | .328 | ....... | .578 | ....... | 1/8 | 1/16 | 1/8 | 3/64 | 1/8 | 3/32 |
| 3/8 | .015 | .390 | ....... | .640 | ....... | 1/8 | 3/32 | 1/8 | 1/16 | 1/8 | 1/8 |
| 7/16 | .015 | .453 | ....... | .767 | ....... | 5/32 | 1/8 | 5/32 | 1/16 | 5/32 | 5/32 |
| 1/2 | .015 | .515 | ....... | .859 | ....... | 11/64 | 1/8 | 11/64 | 1/16 | 11/64 | 11/64 |
| 9/16 | .030 | .593 | ....... | .968 | ....... | 3/16 | 1/8 | 3/16 | 3/32 | 3/16 | 3/16 |
| 5/8 | .030 | .655 | ....... | 1.062 | ....... | 13/64 | 5/32 | 13/64 | 3/32 | 13/64 | 13/64 |
| 11/16 | .030 | .718 | ....... | 1.155 | ....... | 7/32 | 3/16 | 7/32 | 1/8 | 7/32 | 7/32 |
| 3/4 | .030 | .780 | ....... | 1.280 | ....... | 1/4 | 3/16 | 1/4 | 1/8 | 1/4 | 1/4 |
| 7/8 | .030 | .905 | ....... | 1.437 | ....... | 17/64 | 3/16 | 17/64 | 5/32 | 17/64 | 17/64 |
| 1 | .030 | 1.030 | ....... | 1.655 | ....... | 5/16 | 1/4 | 5/16 | 3/16 | 5/16 | 5/16 |
| 1 1/8 | .030 | 1.155 | ....... | 1.905 | ....... | 3/8 | 1/4 | 3/8 | 3/16 | 3/8 | 3/8 |
| 1 1/4 | .030 | 1.280 | ....... | 2.155 | ....... | 7/16 | 1/4 | 7/16 | 3/16 | 7/16 | 5/16 |
| 1 3/8 | .030 | 1.405 | ....... | 2.280 | ....... | 7/16 | 5/16 | 7/16 | 1/4 | 7/16 | 3/8 |
| 1 1/2 | .030 | 1.530 | ....... | 2.530 | ....... | 1/2 | 5/16 | 1/2 | 1/4 | 1/2 | 3/8 |

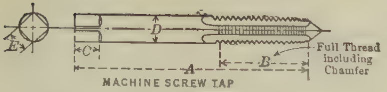
**Table 21.—Dimensions of Shakeproof Lock Washers**

All dimensions in inches. See Fig. 13.

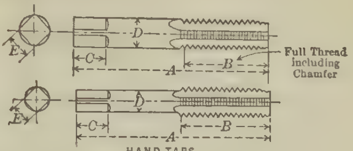
| Size of Screw | External Teeth | | Internal Teeth | | Extra-heavy, Internal Teeth | | Size of Screw | External Teeth | | Internal Teeth | | Extra-heavy, Internal Teeth | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Outside Diam. | Thickness | Outside Diam. | Thickness | Outside Diam. | Thickness | | Outside Diam. | Thickness | Outside Diam. | Thickness | Outside Diam. | Thickness |
| No. 2 | ...... | ...... | 3/16 | 0 010 | ...... | ...... | 7/16 | 25/32 | 0.035 | 25/32 | 0.035 | 27/32 | 0.062 |
| 3 | ...... | ...... | 7/32 | .012 | ...... | ...... | 1/2 | 7/8 | .040 | 7/8 | .040 | 59/64 | .062 |
| 4 | 9/32 | 0.018 | 17/64 | .018 | ...... | ...... | 9/16 | 31/32 | .040 | 31/32 | .040 | 1 1/32 | .062 |
| 6 (1/8″) | 5/16 | .018 | 9/32 | .018 | ...... | ...... | 5/8 | 1 1/16 | .045 | 1 1/16 | .045 | 1 1/8 | .062 |
| 8 (5/32″) | 3/8 | .018 | 21/64 | .018 | ...... | ...... | 11/16 | ...... | ...... | 1 5/32 | .045 | ...... | ...... |
| 10 (3/16″) | 13/32 | .021 | 3/8 | .021 | ...... | ...... | 3/4 | 1 1/4 | .050 | 1 1/4 | .050 | 1 1/4 | .072 |
| 12 | ...... | ...... | 13/32 | .021 | ...... | ...... | 7/8 | 1 7/16 | .062 | 1 3/8 | .050 | 1 7/16 | .078 |
| 1/4 | 1/2 | .025 | 15/32 | .025 | 17/32 | 0.040 | 1 | 1 5/8 | .062 | 1 5/8 | .062 | ...... | ...... |
| 5/16 | 19/32 | .030 | 19/32 | .030 | 19/32 | .045 | 1 1/8 | ...... | ...... | 1 13/16 | .062 | ...... | ...... |
| 3/8 | 11/16 | .035 | 11/16 | .035 | 47/64 | .045 | 1 1/4 | ...... | ...... | 1 15/16 | .062 | ...... | ...... |

## Table 22.—Dimensions of American Standard Machine Screw and Hand Taps

A.S.A. B5e—1930. All dimensions in inches.



MACHINE SCREW TAP



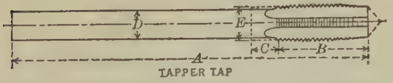
HAND TAPS

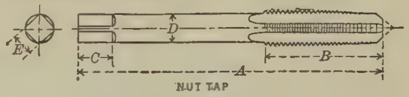
| Size | Threads per in.* | | Pitch Diameter | | | | Major Diameter | | | | | Dimensions | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Basic | Min. | Cut Thread | Ground Thread | Basic | Cut Thread | | Ground Thread | | | | | | |
| | N.C. | N.F. | | | Max. | Max. | | Min. | Max. | Min. | Max. | *A* | *B* | *C* | *D* | *E* |
| **Machine Screw Taps** | | | | | | | | | | | | | | | | |
| 0 | .... | 80 | 0.0519 | 0.0524 | 0.0534 | ...... | 0.0600 | 0.0610 | 0.0625 | ...... | ....... | 1 5/8 | 5/16 | 3/16 | 0.141 | 0.110 |
| 1 | 64 | .... | .0629 | .0634 | .0644 | ...... | .0730 | .0740 | .0755 | ...... | ....... | 1 11/16 | 3/8 | 3/16 | .141 | .110 |
| 1 | .... | 72 | .0640 | .0645 | .0655 | ...... | .0730 | .0740 | .0755 | ...... | ....... | 1 11/16 | 3/8 | 3/16 | .141 | .110 |
| 2 | 56 | .... | .0744 | .0749 | .0759 | ...... | .0860 | .0875 | .0890 | ...... | ....... | 1 3/4 | 7/16 | 3/16 | .141 | .110 |
| 2 | .... | 64 | .0759 | .0764 | .0774 | ...... | .0860 | .0875 | .0890 | ...... | ....... | 1 3/4 | 7/16 | 3/16 | .141 | .110 |
| 3 | 48 | .... | .0855 | .0860 | .0870 | ...... | .0990 | .1005 | .1020 | ...... | ....... | 1 13/16 | 1/2 | 3/16 | .141 | .110 |
| 3 | .... | 56 | .0874 | .0879 | .0889 | ...... | .0990 | .1005 | .1020 | ...... | ....... | 1 13/16 | 1/2 | 3/16 | .141 | .110 |
| 4 | 40 | .... | .0958 | .0963 | .0978 | ...... | .1120 | .1135 | .1155 | ...... | ....... | 1 7/8 | 9/16 | 3/16 | .141 | .110 |
| 4 | .... | 48 | .0985 | .0990 | .1005 | ...... | .1120 | .1135 | .1155 | ...... | ....... | 1 7/8 | 9/16 | 3/16 | .141 | .110 |
| 5 | 40 | .... | .1088 | .1093 | .1108 | ...... | .1250 | .1265 | .1285 | ...... | ....... | 1 15/16 | 5/8 | 3/16 | .141 | .110 |
| 5 | .... | 44 | .1102 | .1107 | .1122 | ...... | .1250 | .1265 | .1285 | ...... | ....... | 1 15/16 | 5/8 | 3/16 | .141 | .110 |
| 6 | 32 | .... | .1177 | .1182 | .1197 | 0.1187 | .1380 | .1395 | .1415 | 0.1395 | 0.1410 | 2 | 11/16 | 3/16 | .141 | .110 |
| 6 | .... | 40 | .1218 | .1223 | .1238 | ...... | .1380 | .1395 | .1415 | ...... | ....... | 2 | 11/16 | 3/16 | .141 | .110 |
| 8 | 32 | .... | .1437 | .1442 | .1457 | .1447 | .1640 | .1660 | .1680 | .1655 | .1670 | 2 1/8 | 3/4 | 1/4 | .168 | .131 |
| 8 | .... | 36 | .1460 | .1465 | .1480 | ...... | .1640 | .1660 | .1680 | ...... | ....... | 2 1/8 | 3/4 | 1/4 | .168 | .131 |
| 10 | 24 | .... | .1629 | .1634 | .1649 | .1639 | .1900 | .1925 | .1945 | .1925 | .1940 | 2 3/8 | 7/8 | 1/4 | .194 | .152 |
| 10 | .... | 32 | .1697 | .1702 | .1717 | .1707 | .1900 | .1925 | .1945 | .1915 | .1930 | 2 3/8 | 7/8 | 1/4 | .194 | .152 |
| 12 | 24 | .... | .1889 | .1894 | .1909 | .1899 | .2160 | .2185 | .2205 | .2185 | .2200 | 2 3/8 | 15/16 | 9/32 | .220 | .165 |
| 12 | .... | 28 | .1928 | .1933 | .1948 | .1938 | .2160 | .2185 | .2205 | .2180 | .2195 | 2 3/8 | 15/16 | 9/32 | .220 | .165 |
| **Hand Taps** | | | | | | | | | | | | | | | | |
| 1/4 | 20 | .... | 0.2175 | 0.2180 | 0.2200 | 0.2185 | 0.2500 | 0.2525 | 0.2550 | 0.2535 | 0.2550 | 2 1/2 | 1 | 9/32 | 0.255 | 0.191 |
| 1/4 | .... | 28 | .2268 | .2273 | .2288 | .2278 | .2500 | .2525 | .2550 | .2520 | .2535 | 2 1/2 | 1 | 9/32 | .255 | .191 |
| 5/16 | 18 | .... | .2764 | .2769 | .2789 | .2774 | .3125 | .3155 | .3180 | .3165 | .3180 | 2 23/32 | 1 1/8 | 5/16 | .318 | .238 |
| 5/16 | .... | 24 | .2854 | .2859 | .2874 | .2864 | .3125 | .3155 | .3180 | .3150 | .3165 | 2 23/32 | 1 1/8 | 5/16 | .318 | .238 |
| 3/8 | 16 | .... | .3344 | .3349 | .3369 | .3354 | .3750 | .3785 | .3810 | .3795 | .3810 | 2 15/16 | 1 1/4 | 3/8 | .381 | .286 |
| 3/8 | .... | 24 | .3479 | .3484 | .3499 | .3489 | .3750 | .3785 | .3810 | .3775 | .3790 | 2 15/16 | 1 1/4 | 3/8 | .381 | .286 |
| 3/8 | 16 | .... | .3344 | .3349 | .3369 | .3354 | .3750 | .3785 | .3810 | .3795 | .3810 | 2 15/16 | 1 1/4 | 3/8 | .275 | .206 |
| 3/8 | .... | 24 | .3479 | .3484 | .3499 | .3489 | .3750 | .3785 | .3810 | .3775 | .3790 | 2 15/16 | 1 1/4 | 3/8 | .275 | .206 |
| 7/16 | 14 | .... | .3911 | .3916 | .3941 | .3921 | .4375 | .4410 | .4440 | .4425 | .4440 | 3 5/32 | 1 7/16 | 13/32 | .323 | .242 |
| 7/16 | .... | 20 | .4050 | .4055 | .4075 | .4060 | .4375 | .4410 | .4440 | .4410 | .4425 | 3 5/32 | 1 7/16 | 13/32 | .323 | .242 |
| 1/2 | 13 | .... | .4500 | .4505 | .4530 | .4510 | .5000 | .5040 | .5070 | .5055 | .5070 | 3 3/8 | 1 21/32 | 7/16 | .367 | .275 |
| 1/2 | .... | 20 | .4675 | .4680 | .4700 | .4685 | .5000 | .5040 | .5070 | .5035 | .5050 | 3 3/8 | 1 21/32 | 7/16 | .367 | .275 |
| 9/16 | 12 | .... | .5084 | .5089 | .5114 | .5094 | .5625 | .5670 | .5700 | .5685 | .5700 | 3 19/32 | 1 21/32 | 1/2 | .429 | .322 |
| 9/16 | .... | 18 | .5264 | .5269 | .5289 | .5274 | .5625 | .5670 | .5700 | .5665 | .5680 | 3 19/32 | 1 21/32 | 1/2 | .429 | .322 |
| 5/8 | 11 | .... | .5660 | .5665 | .5690 | .5676 | .6250 | .6300 | .6330 | .6315 | .6330 | 3 13/16 | 1 13/16 | 9/16 | .480 | .360 |
| 5/8 | .... | 18 | .5889 | .5894 | .5914 | .5902 | .6250 | .6300 | .6330 | .6290 | .6305 | 3 13/16 | 1 13/16 | 9/16 | .480 | .360 |
| 3/4 | 10 | .... | .6850 | .6855 | .6685 | .6866 | .7500 | .7550 | .7590 | .7570 | .7590 | 4 1/4 | 2 | 11/16 | .590 | .442 |
| 3/4 | .... | 16 | .7094 | .7099 | .7124 | .7107 | .7500 | .7550 | .7590 | .7540 | .7560 | 4 1/4 | 2 | 11/16 | .590 | .442 |
| 7/8 | 9 | .... | .8028 | .8038 | .8068 | .8050 | .8750 | .8805 | .8845 | .8830 | .8850 | 4 11/16 | 2 7/32 | 3/4 | .697 | .523 |
| 7/8 | .... | 14 | .8286 | .8296 | .8321 | .8305 | .8750 | .8805 | .8845 | .8795 | .8815 | 4 11/16 | 2 7/32 | 3/4 | .697 | .523 |
| 1 | 8 | .... | .9188 | .9198 | .9228 | .9212 | 1.0000 | 1.0060 | 1.0100 | 1.0090 | 1.0110 | 5 1/8 | 2 1/2 | 13/16 | .800 | .600 |
| 1 | .... | 14 | .9536 | .9546 | .9571 | .9555 | 1.0000 | 1.0060 | 1.0100 | 1.0045 | 1.0065 | 5 1/8 | 2 1/2 | 13/16 | .800 | .600 |
| 1 1/8 | 7 | .... | 1.0322 | 1.0332 | 1.0367 | 1.0347 | 1.1250 | 1.1310 | 1.1355 | 1.1345 | 1.1370 | 5 7/16 | 2 9/16 | 7/8 | .896 | .672 |
| 1 1/8 | .... | 12 | 1.0709 | 1.0719 | 1.0749 | 1.0729 | 1.1250 | 1.1310 | 1.1355 | 1.1300 | 1.1325 | 5 7/16 | 2 9/16 | 7/8 | .896 | .672 |
| 1 1/4 | 7 | .... | 1.1572 | 1.1582 | 1.1617 | 1.1597 | 1.2500 | 1.2565 | 1.2610 | 1.2595 | 1.2620 | 5 3/4 | 2 9/16 | 1 | 1.021 | .766 |
| 1 1/4 | .... | 12 | 1.1959 | 1.1969 | 1.1999 | 1.1979 | 1.2500 | 1.2565 | 1.2610 | 1.2550 | 1.2575 | 5 3/4 | 2 9/16 | 1 | 1.021 | .766 |
| 1 1/2 | 6 | .... | 1.3917 | 1.3927 | 1.3962 | 1.3945 | 1.5000 | 1.5075 | 1.5120 | 1.5115 | 1.5140 | 6 3/8 | 3 | 1 1/8 | 1.233 | .925 |
| 1 1/2 | .... | 12 | 1.4459 | 1.4469 | 1.4499 | 1.4479 | 1.5000 | 1.5075 | 1.5120 | 1.5050 | 1.5075 | 6 3/8 | 3 | 1 1/8 | 1.233 | .925 |
| 1 3/4 | 5 | .... | 1.6201 | 1.6216 | 1.6256 | 1.6236 | 1.7500 | 1.7575 | 1.7630 | 1.7620 | 1.7650 | 7 | 3 3/16 | 1 1/4 | 1.430 | 1.072 |
| 2 | 4 1/2 | .... | 1.8557 | 1.8572 | 1.8612 | 1.8592 | 2.0000 | 2.0085 | 2.0140 | 2.0130 | 2.0160 | 7 5/8 | 3 9/16 | 1 3/8 | 1.644 | 1.233 |
| 2 1/4 | 4 1/2 | .... | 2.1057 | 2.1072 | 2.1117 | 2.1092 | 2.2500 | 2.2590 | 2.2650 | 2.2630 | 2.2660 | 8 1/4 | 3 9/16 | 1 7/16 | 1.894 | 1.420 |
| 2 1/2 | 4 | .... | 2.3376 | 2.3396 | 2.3441 | 2.3416 | 2.5000 | 2.5100 | 2.5160 | 2.5140 | 2.5170 | 8 3/4 | 4 | 1 1/2 | 2.100 | 1.575 |
| 2 3/4 | 4 | .... | 2.5876 | 2.5896 | 2.5946 | 2.5916 | 2.7500 | 2.7600 | 2.7670 | 2.7640 | 2.7670 | 9 1/4 | 4 | 1 9/16 | 2.350 | 1.762 |
| 3 | 3 1/2 | .... | 2.8144 | 2.8164 | 2.8214 | 2.8184 | 3.0000 | 3.0105 | 3.0175 | 3.0150 | 3.0180 | 9 3/4 | 4 9/16 | 1 5/8 | 2.543 | 1.907 |

* N.C. = American National Standard coarse thread; N.F. = American National Standard fine thread.

### Table 23.—Dimensions of American Standard Tapper Taps and Nut Taps

A.S.A. B5e—1930. All dimensions in inches.





| Size | Threads per in.* | | Pitch Diameter | | | | Major Diameter | | | | | Dimensions | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Cut Thread | Ground Thread | | Cut Thread | | Ground Thread | | | | | | |
| | N.C. | N.F. | Basic | Min. | Max. | Max. | Basic | Min. | Max. | Min. | Max. | A | B | C | D | E |
| Tapper Taps | | | | | | | | | | | | | | | | |
| 1/4 | 20 | .... | 0.2175 | 0.2180 | 0.2200 | 0.2185 | 0.2500 | 0.2525 | 0.2550 | 0.2535 | 0.2550 | 12, 15 | 1 1/4 | .... | 0.185 | ...... |
| 1/4 | .... | 28 | .2268 | .2273 | .2288 | .2278 | .2500 | .2525 | .2550 | .2520 | .2535 | 12, 15 | 1 | 1/4 | .185 | 0.2036 |
| 5/16 | 18 | .... | .2764 | .2769 | .2789 | .2774 | .3125 | .3155 | .3180 | .3165 | .3180 | 12, 15 | 1 3/8 | .... | .240 | ...... |
| 5/16 | .... | 24 | .2854 | .2859 | .2874 | .2864 | .3125 | .3155 | .3180 | .3150 | .3165 | 12, 15 | 1 1/16 | 5/16 | .240 | .2584 |
| 3/8 | 16 | .... | .3344 | .3349 | .3369 | .3354 | .3750 | .3785 | .3810 | .3795 | .3810 | 12, 15 | 1 9/16 | .... | .294 | ...... |
| 3/8 | .... | 24 | .3479 | .3484 | .3499 | .3489 | .3750 | .3785 | .3810 | .3775 | .3790 | 12, 15 | 1 3/16 | 3/8 | .294 | .3209 |
| 7/16 | 14 | .... | .3911 | .3916 | .3941 | .3921 | .4375 | .4410 | .4440 | .4425 | .4440 | 12, 15 | 1 13/16 | .... | .345 | ...... |
| 7/16 | .... | 20 | .4050 | .4055 | .4075 | .4060 | .4375 | .4410 | .4440 | .4410 | .4425 | 12, 15 | 1 3/8 | 7/16 | .345 | .3726 |
| 1/2 | 13 | .... | .4500 | .4505 | .4530 | .4510 | .5000 | .5040 | .5070 | .5055 | .5070 | 12, 15 | 1 7/8 | .... | .400 | ...... |
| 1/2 | .... | 20 | .4675 | .4680 | .4700 | .4685 | .5000 | .5040 | .5070 | .5035 | .5050 | 12, 15 | 1 3/8 | 1/2 | .400 | .4351 |
| 9/16 | 12 | .... | .5084 | .5089 | .5114 | .5094 | .5625 | .5670 | .5700 | .5685 | .5700 | 12, 15 | 2 1/8 | .... | .450 | ...... |
| 9/16 | .... | 18 | .5264 | .5269 | .5289 | .5274 | .5625 | .5670 | .5700 | .5665 | .5680 | 12, 15 | 1 9/16 | 9/16 | .450 | .4903 |
| 5/8 | 11 | .... | .5660 | .5665 | .5690 | .5676 | .6250 | .6300 | .6330 | .6315 | .6330 | 12, 15 | 2 5/16 | .... | .503 | ...... |
| 5/8 | .... | 18 | .5889 | .5894 | .5914 | .5902 | .6250 | .6300 | .6330 | .6290 | .6305 | 12, 15 | 1 11/16 | 5/8 | .503 | .5528 |
| 3/4 | 10 | .... | .6850 | .6855 | .6885 | .6866 | .7500 | .7550 | .7590 | .7570 | .7590 | 12, 15 | 2 1/2 | .... | .616 | ...... |
| 3/4 | .... | 16 | .7094 | .7099 | .7124 | .7107 | .7500 | .7550 | .7590 | .7540 | .7560 | 12, 15 | 1 3/4 | 3/4 | .616 | .6688 |
| 7/8 | 9 | .... | .8028 | .8038 | .8068 | .8050 | .8750 | .8805 | .8845 | .8830 | .8850 | 12, 15 | 2 3/4 | .... | .727 | ...... |
| 7/8 | .... | 14 | .8286 | .8296 | .8321 | .8305 | .8750 | .8805 | .8845 | .8795 | .8815 | 12, 15 | 1 7/8 | 7/8 | .727 | .7822 |
| 1 | 8 | .... | .9188 | .9198 | .9228 | .9212 | 1.0000 | 1.0060 | 1.0100 | 1.0090 | 1.0110 | 12, 15 | 3 1/8 | .... | .834 | ...... |
| 1 | .... | 14 | .9536 | .9546 | .9571 | .9555 | 1.0000 | 1.0060 | 1.0100 | 1.0045 | 1.0065 | 12, 15 | 2 1/8 | 1 | .834 | .9072 |
| 1 1/8 | 7 | .... | 1.0322 | 1.0332 | 1.0367 | 1.0347 | 1.1250 | 1.1310 | 1.1355 | 1.1345 | 1.1370 | 15 | 3 1/2 | .... | .933 | ...... |
| 1 1/8 | .... | 12 | 1.0709 | 1.0719 | 1.0749 | 1.0729 | 1.1250 | 1.1310 | 1.1355 | 1.1300 | 1.1325 | 15 | 2 3/8 | 1 1/8 | .933 | 1.0163 |
| 1 1/4 | 7 | .... | 1.1572 | 1.1582 | 1.1617 | 1.1597 | 1.2500 | 1.2565 | 1.2610 | 1.2595 | 1.2620 | 15 | 3 1/2 | .... | 1.058 | ...... |
| 1 1/4 | .... | 12 | 1.1959 | 1.1969 | 1.1999 | 1.1979 | 1.2500 | 1.2565 | 1.2610 | 1.2550 | 1.2575 | 15 | 2 3/8 | 1 1/8 | 1.058 | 1.1418 |
| 1 1/2 | 6 | .... | 1.3917 | 1.3927 | 1.3962 | 1.3945 | 1.5000 | 1.5075 | 1.5120 | 1.5115 | 1.5140 | 15 | 4 | .... | 1.278 | ...... |
| 1 1/2 | .... | 12 | 1.4459 | 1.4469 | 1.4499 | 1.4479 | 1.5000 | 1.5075 | 1.5120 | 1.5050 | 1.5075 | 15 | 2 5/8 | 1 3/8 | 1.278 | 1.3918 |
| 1 3/4 | 5 | .... | 1.6201 | 1.6216 | 1.6256 | 1.6236 | 1.7500 | 1.7575 | 1.7630 | 1.7620 | 1.7650 | 15 | 4 1/2 | .... | 1.484 | ...... |
| 2 | 4 1/2 | .... | 1.8557 | 1.8572 | 1.8612 | 1.8592 | 2.0000 | 2.0085 | 2.0140 | 2.0130 | 2.0160 | 15 | 4 1/2 | .... | 1.705 | ...... |
| Nut Taps | | | | | | | | | | | | | | | | |
| 1/4 | 20 | .... | 0.2175 | 0.2180 | 0.2200 | 0.2185 | 0.2500 | 0.2525 | 0.2550 | 0.2535 | 0.2550 | 5 | 1 5/8 | 9/16 | 0.185 | 0.139 |
| 1/4 | .... | 28 | .2268 | .2273 | .2288 | .2278 | .2500 | .2525 | .2550 | .2520 | .2535 | 5 | 1 1/4 | 9/16 | .185 | .139 |
| 5/16 | 18 | .... | .2764 | .2769 | .2789 | .2774 | .3125 | .3155 | .3180 | .3165 | .3180 | 5 1/2 | 1 13/16 | 5/8 | .240 | .180 |
| 5/16 | .... | 24 | .2854 | .2859 | .2874 | .2864 | .3125 | .3155 | .3180 | .3150 | .3165 | 5 1/2 | 1 3/8 | 5/8 | .240 | .180 |
| 3/8 | 16 | .... | .3344 | .3349 | .3369 | .3354 | .3750 | .3785 | .3810 | .3795 | .3810 | 6 | 2 | 11/16 | .294 | .220 |
| 3/8 | ... | 24 | .3479 | .3484 | .3499 | .3489 | .3750 | .3785 | .3810 | .3775 | .3790 | 6 | 1 1/2 | 11/16 | .294 | .220 |
| 7/16 | 14 | .... | .3911 | .3916 | .3941 | .3921 | .4375 | .4410 | .4440 | .4425 | .4440 | 6 1/2 | 2 3/8 | 3/4 | .345 | .259 |
| 7/16 | ... | 20 | .4050 | .4055 | .4075 | .4060 | .4375 | .4410 | .4440 | .4410 | .4425 | 6 1/2 | 1 3/4 | 3/4 | .345 | .259 |
| 1/2 | 13 | ... | .4500 | .4505 | .4530 | .4510 | .5000 | .5040 | .5070 | .5055 | .5070 | 7 | 2 1/2 | 7/8 | .400 | .300 |
| 1/2 | ... | 20 | .4675 | .4680 | .4700 | .4685 | .5000 | .5040 | .5070 | .5035 | .5050 | 7 | 1 7/8 | 7/8 | .400 | .300 |
| 9/16 | 12 | .... | .5084 | .5089 | .5114 | .5094 | .5625 | .5670 | .5700 | .5685 | .5700 | 7 1/2 | 2 3/4 | 7/8 | .450 | .337 |
| 9/16 | ... | 18 | .5264 | .5269 | .5289 | .5274 | .5625 | .5670 | .5700 | .5665 | .5680 | 7 1/2 | 2 | 7/8 | .450 | .337 |
| 5/8 | 11 | .... | .5660 | .5665 | .5690 | .5676 | .6250 | .6300 | .6330 | .6315 | .6330 | 8 | 3 | 15/16 | .503 | .377 |
| 5/8 | ... | 18 | .5889 | .5894 | .5914 | .5902 | .6250 | .6300 | .6330 | .6290 | .6305 | 8 | 2 1/4 | 15/16 | .503 | .377 |
| 3/4 | 10 | .... | .6850 | .6855 | .6885 | .6866 | .7500 | .7550 | .7590 | .7570 | .7590 | 9 | 3 1/4 | 1 | .616 | .462 |
| 3/4 | .... | 16 | .7094 | .7099 | .7124 | .7107 | .7500 | .7550 | .7590 | .7540 | .7560 | 9 | 2 1/2 | 1 | .616 | .462 |
| 7/8 | 9 | ... | .8028 | .8038 | .8068 | .8050 | .8750 | .8805 | .8845 | .8830 | .8850 | 10 | 3 5/8 | 1 1/16 | .727 | .545 |
| 7/8 | .... | 14 | .8286 | .8296 | .8321 | .8305 | .8750 | .8805 | .8845 | .8795 | .8815 | 10 | 2 3/4 | 1 1/16 | .727 | .545 |
| 1 | 8 | ... | .9188 | .9198 | .9228 | .9212 | 1.0000 | 1.0060 | 1.0100 | 1.0090 | 1.0110 | 11 | 4 | 1 1/8 | .834 | .625 |
| 1 | .... | 14 | .9536 | .9546 | .9571 | .9555 | 1.0000 | 1.0060 | 1.0100 | 1.0045 | 1.0065 | 11 | 3 | 1 1/8 | .834 | .625 |
| 1 1/8 | 7 | .... | 1.0322 | 1.0332 | 1.0367 | 1.0347 | 1.1250 | 1.1310 | 1.1355 | 1.1345 | 1.1370 | 11 1/2 | 4 3/4 | 1 1/4 | .933 | .700 |
| 1 1/8 | .... | 12 | 1.0709 | 1.0719 | 1.0749 | 1.0729 | 1.1250 | 1.1310 | 1.1355 | 1.1300 | 1.1325 | 11 1/2 | 3 1/2 | 1 1/4 | .933 | .700 |
| 1 1/4 | 7 | .... | 1.1572 | 1.1582 | 1.1617 | 1.1597 | 1.2500 | 1.2565 | 1.2610 | 1.2595 | 1.2620 | 12 | 4 3/4 | 1 5/16 | 1.058 | .793 |
| 1 1/4 | ... | 12 | 1.1959 | 1.1969 | 1.1999 | 1.1979 | 1.2500 | 1.2565 | 1.2610 | 1.2550 | 1.2575 | 12 | 3 1/2 | 1 5/16 | 1.058 | .793 |
| 1 1/2 | 6 | .... | 1.3917 | 1.3927 | 1.3962 | 1.3945 | 1.5000 | 1.5075 | 1.5120 | 1.5115 | 1.5140 | 13 | 5 3/8 | 1 1/2 | 1.278 | .958 |
| 1 1/2 | .... | 12 | 1.4459 | 1.4469 | 1.4499 | 1.4479 | 1.5000 | 1.5075 | 1.5120 | 1.5050 | 1.5075 | 13 | 4 | 1 1/2 | 1.278 | .958 |
| 1 3/4 | 5 | .... | 1.6201 | 1.6216 | 1.6256 | 1.6236 | 1.7500 | 1.7575 | 1.7630 | 1.7620 | 1.7650 | 14 | 5 1/2 | 1 5/8 | 1.484 | 1.113 |
| 2 | 4 1/2 | .... | 1.8557 | 1.8572 | 1.8612 | 1.8592 | 2.0000 | 2.0085 | 2.0140 | 2.0130 | 2.0160 | 15 | 6 1/8 | 1 3/4 | 1.705 | 1.279 |
| 2 1/4 | 4 1/2 | .... | 2.1057 | 2.1072 | 2.1117 | 2.1092 | 2.2500 | 2.2590 | 2.2650 | 2.2630 | 2.2660 | 16 | 6 1/8 | 1 7/8 | 1.953 | 1.465 |
| 2 1/2 | 4 | .... | 2.3376 | 2.3396 | 2.3441 | 2.3416 | 2.5000 | 2.5100 | 2.5160 | 2.5140 | 2.5170 | 17 | 6 7/8 | 2 | 2.167 | 1.625 |

* N.C. = American National Standard coarse thread; N.F. = American National Standard fine thread.

**Table 24.—Tolerances in Tap Dimensions.** (A.S.A., B5e—1930)

| Tolerance in | Machine Screw Tap | | Hand Tap | | Tapper Tap | | Nut Tap | |
|---|---|---|---|---|---|---|---|---|
| | Cut | Ground | Cut | Ground | Cut | Ground | Cut | Ground |
| Lead, in. per in. of thread | ±0.003 | ±0.0005 | ±0.003 | ±0.0005 | ±0.003 | ±0.0005 | ±0.003 | ±0.0005 |
| Length overall, in. | ±1/32 | ±1/32 | | | | | | |
| 1/4–1 in. incl., in. | | | ±1/32 | ±1/32 | ±1/8 | ±1/8 | ±1/16 | ±1/16 |
| 1 1/8–3 " " " | | | ±1/16 | ±1/16 | ±3/16 | ±3/16 | ±3/32 | ±3/32 |
| Length of thread, in. | ±3/64 | ±3/64 | | | | | | |
| 1/4–1/2 in. incl., in. | | | ±1/16 | ±1/16 | ±1/16 | ±1/16 | ±1/16 | ±1/16 |
| 9/16–1 1/2 " " " | | | ±3/32 | ±3/32 | ±3/32 | ±3/32 | ±3/32 | ±3/32 |
| 1 3/4–3 " " " | | | ±1/8 | ±1/8 | ±1/8 | ±1/8 | ±1/8 | ±1/8 |
| Length of square, in. | ±1/32 | ±1/32 | | | | | | |
| 1/4–5/8 in. incl., in. | | | | | | | ±1/32 | ±1/32 |
| 1/4–1 " " " | | | ±1/32 | ±1/32 | | | | |
| 3/4–1 1/2 " " " | | | | | | | ±3/64 | ±3/64 |
| 1 1/8–3 " " " | | | ±1/16 | ±1/16 | | | | |
| 1 3/4–2 1/2 " " " | | | | | | | ±1/16 | ±1/16 |
| Diam. of shank, in. | −0.004 | −0.0005 | | | | | | |
| 1/4–1/2 in. incl., in. | | | | | −0.005 | −0.005 | −0.005 | −0.005 |
| 1/4–5/8 " " " | | | | −0.0015 | | | | |
| 1/4–1 " " " | | | −0.005 | | | | | |
| 9/16–1 " " " | | | | | −0.006 | −0.006 | −0.006 | −0.006 |
| 3/4–2 " " " | | | | −0.0020 | | | | |
| 1 1/8–2 " " " | | | −0.007 | | −0.008 | −0.008 | −0.008 | −0.008 |
| 1 3/4–3 " " " | | | | −0.0030 | | | | |
| 2 1/4–3 " " " | | | −0.009 | | | | −0.010 | −0.010 |
| Size of square, in. | −0.004 | −0.004 | | | | | | |
| 1/4–1/2 in. incl., in. | | | −0.004 | −0.004 | | | −0.004 | −0.004 |
| 9/16–1 " " " | | | −0.006 | −0.006 | | | −0.006 | −0.006 |
| 1 1/8–2 " " " | | | −0.008 | −0.008 | | | −0.008 | −0.008 |
| 2 1/4–3 " " " | | | −0.010 | −0.010 | | | −0.010 | −0.010 |
| Half Angle, minutes | | | | | | | | |
| 24–28 thds. per in. | ±45 | | | | | | | |
| 24–32 " " " | | ±30 | | | | | | |
| 30–80 " " " | ±60 | | | | | | | |
| 3 1/2–4 " " " | | | ±30 | | | | | |
| 3 1/2–5 " " " | | | | ±20 | | | | |
| 4 " " " | | | | | | | ±30 | |
| 4–5 " " " | | | | | | | | ±20 |
| 4 1/2–5 " " " | | | ±35 | | ±35 | ±20 | ±35 | |
| 6–9 " " " | | | ±40 | ±25 | ±40 | ±25 | ±40 | ±25 |
| 10–28 " " " | | | ±45 | ±30 | ±45 | ±30 | ±45 | ±30 |
| Full Angle, minutes | | | | | | | | |
| 24–28 thds. per in. | 68 | | | | | | | |
| 30–80 " " " | 90 | | | | | | | |
| 3 1/2–4 " " " | | | 45 | | | | | |
| 4 " " " | | | | | | | 45 | |
| 4 1/2–5 " " " | | | 53 | | 53 | | 53 | |
| 6–9 " " " | | | 60 | | 60 | | 60 | |
| 10–28 " " " | | | 68 | | 68 | | 68 | |
| Pilot Diameter, in. | | | | | | | | |
| 1/4–1 1/2 N.F.* | | | | | −0.005 | −0.005 | | |
| Major Diam., max.† | | | | | | | | |
| No. 0–12 incl. | $D_b + 0.01\sqrt{D_b}$ | | | | | | | |
| 24–32 thds. per in. | | $D_b + 0.45t$ | | | | | | |
| 3 1/2–5 " " " | | | | $D_b + 0.35t$ | | | | |
| 4–5 " " " | | | | | | | | $D_b + 0.35t$ |
| 4 1/2–5 " " " | | | | | | $D_b + 0.35t$ | | |
| 6–12 " " " | | | | $D_b + 0.40t$ | | $D_b + 0.40t$ | | $D_b + 0.40t$ |
| 13–28 " " " | | | | $D_b + 0.45t$ | | $D_b + 0.45t$ | | $D_b + 0.4t$ |
| 1/4–3 in. | | | $D_b + 0.01\sqrt{D_b}$ | | $D_b + 0.01\sqrt{D_b}$ | | $D_b + 0.01\sqrt{D_b}$ | |
| Major Diam., min. | | | | | | | | |
| No. 0–3 incl. | $D_x - 0.0015$ | | | | | | | |
| No. 4–12 incl. | $D_x - 0.002$ | | | | | | | |
| No. 6–12 incl. | | $D_x - 0.0015$ | | | | | | |
| 1/4–3/8 | | | $D_x - 0.0025$ | | $D_x - 0.0025$ | | $D_x - 0.0025$ | |
| 1/4–5/8 | | | | $D_x - 0.0015$ | | $D_x - 0.0015$ | | $D_x - 0.0015$ |
| 7/16–5/8 | | | $D_x - 0.003$ | | $D_x - 0.003$ | | $D_x - 0.003$ | |
| 3/4–1 | | | $D_x - 0.004$ | $D_x - 0.002$ | $D_x - 0.004$ | $D_x - 0.002$ | $D_x - 0.004$ | $D_x - 0.002$ |
| 1 1/8–1 1/2 | | | $D_x - 0.0045$ | $D_x - 0.0025$ | $D_x - 0.0045$ | $D_x - 0.0025$ | $D_x - 0.0045$ | $D_x - 0.0025$ |
| 1 3/4–2 | | | $D_x - 0.0055$ | | $D_x - 0.0055$ | $D_x - 0.003$ | $D_x - 0.0055$ | |
| 1 3/4–2 1/2 | | | | | | | | $D_x - 0.003$ |
| 1 3/4–3 | | | | $D_x - 0.003$ | | | | |
| 2 1/4–2 1/2 | | | $D_x - 0.006$ | | | | $D_x - 0.006$ | |
| 2 3/4–3 | | | $D_x - 0.007$ | | | | | |

* National Fine Thread Series.

† $D_b$ = basic major diameter; $D_x$ = maximum diameter; $t$ = truncation.

**Table 25.—Tap Drill Sizes for American National Standard Screw Threads**

Cleveland Twist Drill Co., Cleveland, Ohio

| Size | Coarse Thread Series | | | Fine Thread Series | | | Size | Coarse Thread Series | | | Fine Thread Series | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Threads per in. | Commercial Tap Drill to Produce Approx. 75% Full Thread | Diam. of Drill, in. | Threads per in. | Commercial Tap Drill to Produce Approx. 75% Full Thread | Diam. of Drill, in. | | Threads per in. | Commercial Tap Drill to Produce Approx. 75% Full Thread | Diam. of Drill, in. | Threads per in. | Commercial Tap Drill to Produce Approx. 75% Full Thread | Diam. of Drill, in. |
| 0 | ...... | ........ | ...... | 80 | 3/64 | 0.0469 | 3/4 | 10 | 21/32 | 0.6562 | 16 | 11/16 | 0.6875 |
| 1 | 64 | No. 53 | 0.0595 | 72 | No. 53 | .0595 | 7/8 | 9 | 49/64 | .7656 | 14 | 13/16 | .8125 |
| 2 | 56 | No. 50 | .0700 | 64 | No. 50 | .0700 | 1 | 8 | 7/8 | .8750 | 14 | 15/16 | .9375 |
| 3 | 48 | No. 47 | .0785 | 56 | No. 45 | .0820 | 1 1/8 | 7 | 63/64 | .9844 | 12 | 1 3/64 | 1.0469 |
| 4 | 40 | No. 43 | .0890 | 48 | No. 42 | .0935 | 1 1/4 | 7 | 1 7/64 | 1.1094 | 12 | 1 11/64 | 1.1719 |
| 5 | 40 | No. 38 | .1015 | 44 | No. 37 | .1040 | 1 3/8 | 6 | 1 7/32 | 1.2187 | 12 | 1 19/64 | 1.2969 |
| 6 | 32 | No. 36 | .1065 | 40 | No. 33 | .1130 | 1 1/2 | 6 | 1 21/64 | 1.3281 | 12 | 1 27/64 | 1.4219 |
| 8 | 32 | No. 29 | .1360 | 36 | No. 29 | .1360 | 1 3/4 | 5 | 1 35/64 | 1.5469 | ....... | ........ | ....... |
| 10 | 24 | No. 25 | .1495 | 32 | No. 21 | .1590 | 2 | 4 1/2 | 1 25/32 | 1.7812 | ....... | ........ | ....... |
| 12 | 24 | No. 16 | .1770 | 28 | No. 14 | .1820 | 2 1/4 | 4 1/2 | 2 1/32 | 2.0312 | ....... | ........ | ....... |
| 1/4 | 20 | No. 7 | .2010 | 28 | No. 3 | .2130 | 2 1/2 | 4 | 2 1/4 | 2.2500 | ....... | ........ | ....... |
| 5/16 | 18 | F | .2570 | 24 | I | .2720 | 2 3/4 | 4 | 2 1/2 | 2.5000 | ....... | ........ | ....... |
| 3/8 | 16 | 5/16 | .3125 | 24 | Q | .3320 | 3 | 4 | 2 3/4 | 2.7500 | ....... | ........ | ....... |
| 7/16 | 14 | U | .3680 | 20 | 25/64 | .3906 | 3 1/4 | 4 | 3 | 3.0000 | ....... | ........ | ....... |
| 1/2 | 13 | 27/64 | .4219 | 20 | 29/64 | .4531 | 3 1/2 | 4 | 3 1/4 | 3.2500 | ....... | ........ | ....... |
| 9/16 | 12 | 31/64 | .4844 | 18 | 33/64 | .5156 | 3 3/4 | 4 | 3 1/2 | 3.5000 | ....... | ........ | ....... |
| 5/8 | 11 | 17/32 | .5312 | 18 | 37/64 | .5781 | 4 | 4 | 3 3/4 | 3.7500 | ....... | ........ | ....... |

## 3. BOLTS, NUTS AND SCREWS

The distinction between bolts, studs, cap-screws, tap-bolts and machine-screws is shown in Fig. 14. The length includes the heads of flat-head screws, stove-bolts and stove-rods. It excludes the heads of bolts, cap-screws, tap-bolts, round- and fillister-head machine screws, round-head stove-bolts and stove-rods, and about one-half the head of round-head wood screws.

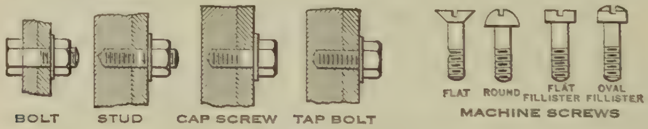


FIG. 14. Forms of Screw Fastenings

If it can be avoided, a hole in cast-iron never should be tapped, as the iron is weak and brittle, and tends to crumble. When it is necessary to tap cast-iron, studs should be used if the attached parts require frequent removal. The cast-iron thread would soon wear if cap-screws were used, whereas the stud is put in once for all.

**BOLTS.**—Standard bolt heads are classed as unfinished, *i.e.*, not machined on any surface, semi-finished, *i.e.*, machined under the head only, and finished, *i.e.*, machined on all surfaces. Bolts and nuts are further classified as: Regular series bolts and nuts, which are intended for general use; heavy bolts and heavy nuts, which are used where greater bearing surface is necessary, or where a large clearance between the bolt and the hole, or where a greater wrench bearing surface is essential; light series nuts, used where extreme saving in weight and material is desirable.

**Regular Bolt Heads and Nuts.**—The dimensions of American Standard regular bolt heads and nuts are given in Table 26. The width across the flats of all bolt heads is $1\ 1/2\ D$, adjusted to sixteenths of an inch, where $D$ = diameter of bolt, except for the sizes of 1/4 to 5/8 in. inclusive, of finished bolt heads. For these bolts, the width across flats is $(1\ 1/2\ D + 1/16)$, adjusted to sixteenths. The tolerance for the width across flats is as follows: Unfinished and semi-finished, minus $0.050\ D$ from basic; finished bolt heads minus $(0.015\ D + 0.006)$ from basic. The minimum width across rounded corners of square bolt heads is: Unfinished and semi-finished square bolt heads, 1.373 × minimum width across flats; minimum width across rounded corners of unfinished, semi-finished, and finished hexagon bolt heads is 1.14 × minimum width across flats. The nominal height of head is the distance from the top to the bearing surface. For unfinished bolts it is $2/3\ D$, adjusted to fractions, and for finished bolt heads it is ${}^3/_4\,D$. For semi-finished bolt heads, the nominal height is as follows:

| | | | | |
|---|---|---|---|---|
| Size, in. . . . . . . . . . . . . | ${}^1/_4$ to ${}^7/_{16}$ | ${}^1/_2$ to ${}^7/_8$ | 1 to 1 ${}^7/_8$ | 2 to 3 |
| Height. . . . . . . . . . . . . | $({}^2/_3\,D - {}^1/_{64})$ | $({}^2/_3\,D - {}^1/_{32})$ | $({}^2/_3\,D - {}^1/_{16})$ | $({}^2/_3\,D - {}^1/_8)$ |

These values all are adjusted to the nearest ${}^1/_{64}$ in., with a tolerance of $\pm(0.016\,D + 0.012)$ from nominal. This also is the tolerance for unfinished bolt heads. For finished bolt heads it is $\pm(0.015\,D + 0.003)$. The tops of all bolt heads are flat and chamfered at an angle to the top surface of 25 deg. for square, and 30 deg. for hexagon. The diameter of the top flat circle is the maximum width across flat. The maximum radius under the head for all classes of bolts and heads is as follows:

| | | | | |
|---|---|---|---|---|
| Size of bolt, in. . . . . . . . . . . . . . . . . . . . . . . | ${}^1/_4$ to ${}^1/_2$ | ${}^9/_{16}$ to 1 | 1 ${}^1/_8$ to 2 | 2 ${}^1/_4$ to 3 |
| Max. radius, in. . . . . . . . . . . . . . . . . . . . . . . | ${}^1/_{32}$ | ${}^1/_{16}$ | ${}^1/_8$ | ${}^3/_{16}$ |

**Heavy Bolt Heads.**—Table 26 gives the dimensions of American Standard heavy bolt heads. The width across flats is $(1\,{}^1/_2\,D + {}^1/_8)$ adjusted to sixteenths. The tolerance for width across flats and for minimum width of rounded corners is the same as for regular bolt heads. Nominal height of head, and tolerances, are as follows: Unfinished bolt heads, height, $({}^3/_4\,D + {}^1/_{16})$, tolerance, minus $(0.016\,D + 0.012)$ from nominal; finished bolt

**Table 26.—Dimensions of American Standard Regular and Heavy Bolt Heads**

A.S.A., B18.2—1933. All dimensions in inches

| Diam. of Bolt, $D$ | Square and Hexagon | | | | | | | | Finished, Hexagon | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Width Across Flats | | Width Across Corners, Min. | | Unfinished Height | | Semi-finished Height | | Width Across Flats | | Width Across Corners | Height | |
| | Max. (Basic) | Min. | Square | Hex. | Max. | Min. | Max. | Min. | Max. | Min. | Min. | Max. | Min. |
| **Regular Bolt Heads** | | | | | | | | | | | | | |
| 1/4 | 0.3750 | 0.363 | 0.498 | 0.414 | 0.188 | 0.156 | 0.172 | 0.140 | 0.4375 | 0.428 | 0.488 | 0.194 | 0.181 |
| 5/16 | .5000 | .484 | .665 | .552 | .220 | .186 | .205 | .171 | .5625 | .552 | .629 | .242 | .227 |
| 3/8 | .5625 | .544 | .747 | .620 | .268 | .232 | .252 | .216 | .6250 | .613 | .699 | .289 | .273 |
| 7/16 | .6250 | .603 | .828 | .687 | .316 | .278 | .300 | .262 | .7500 | .737 | .840 | .338 | .319 |
| 1/2 | .7500 | .725 | .995 | .827 | .348 | .308 | .317 | .277 | .8125 | .799 | .911 | .386 | .365 |
| 9/16 | .8750 | .847 | 1.163 | .966 | .396 | .354 | .365 | .323 | .8750 | .861 | .982 | .432 | .410 |
| 5/8 | .9375 | .906 | 1.244 | 1.033 | .444 | .400 | .413 | .369 | 1.0000 | .985 | 1.123 | .481 | .456 |
| 3/4 | 1.1250 | 1.088 | 1.494 | 1.240 | .524 | .476 | .493 | .445 | 1.1250 | 1.108 | 1.263 | .577 | .548 |
| 7/8 | 1.3125 | 1.269 | 1.742 | 1.447 | .620 | .568 | .589 | .537 | 1.3125 | 1.293 | 1.474 | .672 | .640 |
| 1 | 1.5000 | 1.450 | 1.991 | 1.653 | .684 | .628 | .622 | .566 | 1.5000 | 1.479 | 1.686 | .768 | .732 |
| 1 1/8 | 1.6875 | 1.631 | 2.239 | 1.859 | .780 | .720 | .718 | .658 | 1.6875 | 1.665 | 1.898 | .864 | .824 |
| 1 1/4 | 1.8750 | 1.813 | 2.489 | 2.067 | .876 | .812 | .813 | .749 | 1.8750 | 1.850 | 2.109 | .959 | .916 |
| 1 3/8 | 2.0625 | 1.994 | 2.738 | 2.273 | .940 | .872 | .878 | .810 | 2.0625 | 2.036 | 2.321 | 1.055 | 1.008 |
| 1 1/2 | 2.2500 | 2.175 | 2.986 | 2.480 | 1.036 | .964 | .974 | .902 | 2.2500 | 2.222 | 2.533 | 1.151 | 1.100 |
| 1 5/8 | 2.4375 | 2.356 | 3.235 | 2.686 | 1.132 | 1.056 | 1.069 | .993 | 2.4375 | 2.407 | 2.744 | 1.246 | 1.191 |
| 1 3/4 | 2.6250 | 2.538 | 3.485 | 2.893 | 1.196 | 1.116 | 1.134 | 1.054 | 2.6250 | 2.593 | 2.956 | 1.342 | 1.283 |
| 1 7/8 | 2.8125 | 2.719 | 3.733 | 3.100 | 1.292 | 1.208 | 1.230 | 1.146 | 2.8125 | 2.778 | 3.168 | 1.437 | 1.375 |
| 2 | 3.0000 | 2.900 | 3.982 | 3.306 | 1.388 | 1.300 | 1.263 | 1.175 | 3.0000 | 2.964 | 3.379 | 1.533 | 1.467 |
| 2 1/4 | 3.3750 | 3.263 | 4.480 | 3.720 | 1.548 | 1.452 | 1.423 | 1.327 | 3.3750 | 3.335 | 3.802 | 1.724 | 1.651 |
| 2 1/2 | 3.7500 | 3.625 | 4.977 | 4.133 | 1.708 | 1.604 | 1.583 | 1.479 | 3.7500 | 3.707 | 4.226 | 1.916 | 1.835 |
| 2 3/4 | 4.1250 | 3.988 | 5.476 | 4.546 | 1.884 | 1.774 | 1.744 | 1.632 | 4.1250 | 4.078 | 4.649 | 2.107 | 2.018 |
| 3 | 4.5000 | 4.350 | 5.973 | 4.959 | 2.060 | 1.940 | 1.935 | 1.815 | 4.5000 | 4.449 | 5.072 | 2.298 | 2.202 |
| **Heavy Bolt Heads** | | | | | | | | | | | | | |
| 1/2 | 0.8750 | 0.850 | 1.167 | 0.969 | 0.458 | 0.418 | 0.426 | 0.336 | ........ | 0.862 | 0.983 | 0.448 | 0.427 |
| 9/16 | 0.9375 | 0.906 | 1.244 | 1.033 | .490 | .448 | .459 | .417 | ........ | 0.923 | 1.052 | .511 | .489 |
| 5/8 | 1.0625 | 1.031 | 1.416 | 1.175 | .553 | .509 | .522 | .478 | ........ | 1.047 | 1.194 | .575 | .550 |
| 3/4 | 1.2500 | 1.213 | 1.665 | 1.383 | .649 | .601 | .618 | .570 | ........ | 1.233 | 1.406 | .702 | .673 |
| 7/8 | 1.4375 | 1.394 | 1.914 | 1.589 | .745 | .693 | .714 | .662 | ........ | 1.418 | 1.617 | .829 | .796 |
| 1 | 1.6250 | 1.575 | 2.162 | 1.796 | .841 | .785 | .778 | .722 | ........ | 1.604 | 1.829 | .956 | .920 |
| 1 1/8 | 1.8125 | 1.756 | 2.411 | 2.002 | .936 | .876 | .874 | .814 | ........ | 1.790 | 2.041 | 1.082 | 1.043 |
| 1 1/4 | 2.0000 | 1.938 | 2.661 | 2.209 | 1.032 | .968 | .970 | .906 | ........ | 1.975 | 2.252 | 1.210 | 1.166 |
| 1 3/8 | 2.1875 | 2.119 | 2.909 | 2.416 | 1.128 | 1.060 | 1.065 | .997 | ........ | 2.161 | 2.464 | 1.336 | 1.289 |
| 1 1/2 | 2.3750 | 2.300 | 3.158 | 2.622 | 1.224 | 1.152 | 1.161 | 1.089 | ........ | 2.347 | 2.676 | 1.463 | 1.412 |
| 1 5/8 | 2.5625 | 2.481 | 3.406 | 2.828 | 1.319 | 1.243 | 1.257 | 1.181 | ........ | 2.532 | 2.886 | 1.590 | 1.535 |
| 1 3/4 | 2.7500 | 2.663 | 3.656 | 3.036 | 1.415 | 1.335 | 1.353 | 1.273 | ........ | 2.718 | 3.099 | 1.716 | 1.659 |
| 1 7/8 | 2.9375 | 2.844 | 3.905 | 3.242 | 1.511 | 1.427 | 1.448 | 1.364 | ........ | 2.903 | 3.309 | 1.844 | 1.781 |
| 2 | 3.1250 | 3.025 | 4.153 | 3.449 | 1.607 | 1.519 | 1.482 | 1.394 | ........ | 3.089 | 3.522 | 1.971 | 1.905 |
| 2 1/4 | 3.5000 | 3.388 | 4.652 | 3.862 | 1.798 | 1.702 | 1.673 | 1.577 | ........ | 3.460 | 3.944 | 2.224 | 2.151 |
| 2 1/2 | 3.8750 | 3.750 | 5.149 | 4.275 | 1.990 | 1.886 | 1.865 | 1.761 | ........ | 3.832 | 4.368 | 2.477 | 2.397 |
| 2 3/4 | 4.2500 | 4.113 | 5.647 | 4.689 | 2.181 | 2.069 | 2.056 | 1.944 | ........ | 4.203 | 4.791 | 2.732 | 2.643 |
| 3 | 4.6250 | 4.475 | 6.144 | 5.102 | 2.373 | 2.253 | 2.248 | 2.128 | ........ | 4.574 | 5.214 | 2.986 | 2.890 |

heads, height, $(D - 1/16)$, tolerance, minus $(0.015\,D + 0.003)$ from nominal. Nominal heights and tolerances of semi-finished bolts are as follows:

| | | | |
|---|---|---|---|
| Size, in. | 1/2 to 7/8 | 1 to 1 7/8 | 2 to 3 |
| Height | $(3/4\,D + 1/32)$ | $3/4\,D$ | $(3/4\,D - 1/16)$ |

Tolerance, all sizes, $\pm(0.016\,D + 0.012)$ from nominal.

Chamfer of top of head and radius under head are the same as for regular bolt heads.

### Table 27.—Length of Thread on Bolts

| Diam. of Bolt, in. | Length of Bolt, inches | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 1 to 1 1/2 | 1 5/8 to 2 | 2 1/8 to 2 1/2 | 2 5/8 to 3 | 3 1/8 to 4 | 4 1/8 to 8 | 8 1/8 to 12 | 12 1/8 to 20 |
| 1/4 | 3/4 | 7/8 | 1 | 1 | 1 1/8. | 1 1/4 | ....... | ........ |
| 5/16 | 3/4 | 7/8 | 1 | 1 | 1 1/8 | 1 1/4 | ....... | ........ |
| 3/8 | 7/8 | 1 | 1 1/8 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | ........ |
| 7/16 | 7/8 | 1 | 1 1/8 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | ........ |
| 1/2 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 5/8 | 1 3/4 |
| 9/16 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 5/8 | 1 3/4 |
| 5/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 5/8 | 1 3/4 | 2 |
| 3/4 | ....... | 1 1/4 | 1 3/8 | 1 1/2 | 1 5/8 | 1 3/4 | 2 | 2 1/4 |
| 7/8 | ....... | 1 1/4 | 1 1/2 | 1 3/4 | 1 7/8 | 2 | 2 1/4 | 2 1/2 |
| 1 | ....... | ....... | 1 3/4 | 2 | 2 1/4 | 2 1/2 | 2 3/4 | 3 |
| 1 1/8 | ....... | ....... | ........ | 2 1/4 | 2 1/2 | 2 3/4 | 3 | 3 1/4 |
| 1 1/4 | ....... | ....... | ........ | ....... | 2 1/2 | 3 | 3 1/4 | 3 1/2 |

### Table 28.—Approximate Weight of 100 Machine Bolts, Square Heads and Square Nuts

| Length, under Head to Point, in. | Diameter | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/4 in. | 3/8 in. | 7/16 in. | 1/2 in. | 9/16 in. | 5/8 in. | 3/4 in. | 7/8 in. | 1 in. | 1 1/4 in. | 1 1/2 in. | 1 3/4 in. | 2 in. |
| 1 1/4 | 3.1 | 8.4 | 12.5 | 17.7 | 24.3 | 30.7 | 50.4 | ...... | ...... | ....... | ..... | ...... | ..... |
| 1 1/2 | 3.4 | 9.2 | 13.6 | 19.1 | 26.0 | 32.8 | 53.5 | ...... | ...... | ....... | ..... | ...... | ..... |
| 2 | 4.1 | 10.8 | 15.7 | 21.8 | 29.5 | 37.1 | 59.7 | 89.4 | 125.7 | ....... | ..... | ...... | ..... |
| 2 1/2 | 4.8 | 12.3 | 17.8 | 24.6 | 33.0 | 41.4 | 65.9 | 97.3 | 136.8 | 246.3 | ..... | ...... | ..... |
| 3 | 5.5 | 13.8 | 19.9 | 27.4 | 36.5 | 45.7 | 72.1 | 105.7 | 147.8 | 263.5 | 470 | ...... | ..... |
| 3 1/2 | 6.2 | 15.3 | 21.8 | 29.8 | 40.0 | 50.0 | 78.3 | 114.2 | 158.9 | 280.8 | 495 | ...... | ..... |
| 4 | 6.9 | 16.9 | 24.0 | 32.6 | 43.5 | 54.4 | 84.5 | 122.6 | 169.9 | 298.1 | 520 | 720 | ..... |
| 4 1/2 | 7.5 | 18.4 | 26.1 | 35.4 | 46.7 | 58.3 | 90.3 | 130.5 | 179.4 | 314.1 | 545 | 753 | ..... |
| 5 | 8.2 | 19.9 | 28.2 | 38.1 | 50.2 | 62.6 | 96.5 | 138.9 | 190.4 | 331.4 | 570 | 786 | 1180 |
| 5 1/2 | 8.9 | 21.5 | 30.3 | 40.9 | 53.7 | 66.9 | 102.7 | 147.4 | 201.5 | 348.6 | 595 | 820 | 1225 |
| 6 | 9.6 | 23.0 | 32.4 | 43.7 | 57.2 | 71.3 | 108.9 | 155.8 | 212.5 | 365.9 | 620 | 854 | 1270 |
| 6 1/2 | 10.3 | 24.6 | 34.5 | 46.4 | 60.7 | 75.6 | 115.1 | 164.3 | 223.6 | 383.1 | 645 | 888 | 1315 |
| 7 | 11.0 | 26.1 | 36.6 | 49.2 | 64.2 | 79.9 | 121.3 | 172.7 | 234.6 | 400.4 | 670 | 922 | 1360 |
| 7 1/2 | 11.7 | 27.7 | 38.8 | 51.9 | 67.6 | 84.2 | 127.6 | 181.2 | 245.6 | 417.7 | 695 | 956 | 1405 |
| 8 | 12.4 | 29.2 | 40.9 | 54.7 | 71.1 | 88.5 | 133.8 | 189.6 | 256.7 | 434.9 | 725 | 990 | 1450 |
| 9 | 13.7 | 32.4 | 44.9 | 60.0 | 77.8 | 96.8 | 145.7 | 205.9 | 278.0 | 468.2 | 775 | 1058 | 1540 |
| 10 | 15.1 | 35.5 | 49.1 | 65.5 | 84.8 | 105.4 | 158.2 | 222.8 | 300.0 | 502.7 | 825 | 1126 | 1630 |
| 11 | 16.5 | 38.6 | 53.4 | 71.0 | 91.8 | 114.1 | 170.6 | 239.8 | 322.2 | 537.3 | 875 | 1194 | 1720 |
| 12 | 17.9 | 41.7 | 57.6 | 76.5 | 93.8 | 122.7 | 183.0 | 256.7 | 344.3 | 571.8 | 925 | 1262 | 1810 |
| 13 | 19.3 | 44.8 | 61.8 | 82.0 | 105.5 | 131.0 | 195.4 | 273.6 | 366.3 | 606.3 | 975 | 1330 | 1900 |
| 14 | 20.6 | 47.9 | 66.0 | 87.6 | 112.5 | 139.6 | 207.9 | 290.5 | 388.4 | 640.8 | 1025 | 1398 | 1990 |
| 15 | 22.0 | 51.0 | 70.3 | 93.1 | 119.5 | 148.2 | 220.3 | 307.4 | 410.5 | 675.3 | 1075 | 1468 | 2080 |
| 16 | 23.4 | 54.1 | 74.5 | 98.6 | 126.4 | 156.9 | 232.7 | 324.3 | 432.6 | 709.8 | 1125 | 1536 | 2170 |
| 17 | 24.8 | 57.2 | 78.7 | 104.1 | 133.4 | 165.5 | 245.1 | 341.2 | 454.7 | 744.3 | 1175 | 1604 | 2260 |
| 18 | 26.2 | 60.3 | 82.9 | 109.7 | 140.4 | 174.1 | 257.6 | 358.1 | 476.8 | 778.9 | 1225 | 1672 | 2350 |
| 20 | 28.9 | 66.5 | 91.4 | 120.7 | 154.4 | 191.4 | 282.4 | 392.0 | 521.0 | 847.9 | 1325 | 1808 | 2530 |
| 22 | 31.7 | 72.7 | 99.9 | 131.7 | 168.4 | 208.6 | 307.3 | 425.8 | 565.1 | 916.9 | 1425 | 1944 | 2710 |
| 24 | 34.4 | 78.9 | 108.3 | 142.8 | 182.4 | 225.9 | 332.1 | 459.6 | 609.3 | 986.0 | 1525 | 2080 | 2890 |
| 26 | 37.2 | 85.2 | 116.8 | 153.8 | 196.3 | 243.1 | 357.0 | 493.4 | 653.5 | 1055.0 | 1625 | 2216 | 3070 |
| 28 | 40.0 | 91.4 | 125.2 | 164.9 | 210.3 | 260.4 | 381.8 | 527.3 | 697.7 | 1124.0 | 1725 | 2352 | 3250 |
| 30 | 42.7 | 97.6 | 133.7 | 175.9 | 224.3 | 277.7 | 406.7 | 561.1 | 741.9 | 1193.0 | 1825 | 2488 | 3450 |
| Weight per 100 Nuts, lb. | | | | | | | | | | | | | |
| Square | 0.7 | 2.5 | 3.9 | 5.7 | 8.1 | 9.9 | 16.8 | 26.9 | 40.1 | 77.8 | 162 | 257 | 381 |
| Hexagon | 0.6 | 2.1 | 3.2 | 4.8 | 6.7 | 8.3 | 14.0 | 22.3 | 33.4 | 64.0 | 134 | 207 | 307 |
| Diff. | 0.1 | 0.4 | 0.7 | 0.9 | 1.4 | 1.6 | 2.8 | 4.6 | 6.7 | 13.8 | 28 | 50 | 74 |
| Weight of 100 Heads, lb. | | | | | | | | | | | | | |
| Square | 0.8 | 2.4 | 4.0 | 5.9 | 8.8 | 11.4 | 20.0 | 31.4 | 44.9 | 90.9 | 144 | 231 | 345 |
| Hexagon | 0.7 | 2.2 | 3.5 | 5.3 | 7.9 | 10.3 | 17.0 | 28.2 | 39.4 | 83.9 | 132 | 215 | 302 |
| Diff. | 0.1 | 0.2 | 0.5 | 0.6 | 0.9 | 1.1 | 3.0 | 3.2 | 5.5 | 7.0 | 12 | 16 | 43 |
| For Weight of Bolts with Hexagon Heads and Hexagon Nuts | | | | | | | | | | | | | |
| Subtract | 0.2 | 0.6 | 1.2 | 1.5 | 2.3 | 2.7 | 5.8 | 7.8 | 12.2 | 20.8 | 40 | 66 | 117 |

**Length of Thread on Bolts.**—Table 27 from *Machinery's Handbook* gives the length of thread on bolts up to $1\,^1/_4$ in. diameter and up to 20 in. long. For bolts exceeding these dimensions, the length of thread is made equal to 3 times the bolt diameter.

**NUTS.**—American Standard nuts are classified as regular and jam nuts, and are further classified as unfinished, semi-finished and finished nuts. Unfinished nuts are threaded but not machined on any surface. Semi-finished nuts are threaded and machined on the bearing surface only. Finished nuts are threaded and machined on all surfaces. The width across the flats of all types of nuts is $1\,^1/_2\,D$, except for sizes $^1/_4$ to $^5/_8$ in., where width across flats equals $1\,^1/_2\,D + {}^1/_{16}$, with a tolerance of minus $0.050\,D$ from basic for unfinished and semi-finished nuts, and of minus $(0.015\,D + 0.006)$ from basic for finished nuts. Minimum width across rounded corners is the same as for bolt heads. The nominal thickness is the overall distance from the top to the bearing surface and is equal to $^7/_8\,D$ for unfinished and finished nuts. For semi-finished nuts it is as follows:

| Size, in. . . . . . . . . . | $^1/_4$ to $^7/_{16}$ | $^1/_2$ to $1\,^1/_8$ | $1\,^1/_4$ to $2\,^1/_4$ | $2\,^1/_2$ to 3 |
|---|---|---|---|---|
| Thickness. . . . . . . . | $(^1/_2\,D + {}^1/_{64})$ | $(^1/_2\,D + {}^3/_{64})$ | $(^1/_2\,D + {}^1/_{16})$ | $(^1/_2\,D + {}^5/_{32})$ |

**Table 29.—Dimensions of American Standard Regular and Heavy Nuts**

A.S.A., B18.2—1933. All dimensions in inches

| Diam., $D$ | Unfinished and Semi-finished Square and Hexagon | | | | | | | | Finished Hexagon | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Width Across Flats | | Width Across Corners, Min. | | Unfinished Thickness | | Semi-finished Thickness | | Width Across Flats | | Width Across Corners, Min. | Thickness | |
| | Max. (Basic) | Min. | Sq. | Hex. | Max. | Min. | Max. | Min. | Max. | Min. | | Max. | Min. |
| **Regular Nuts** | | | | | | | | | | | | | |
| $^1/_4$ | 0.4375 | 0.425 | 0.584 | 0.485 | 0.235 | 0.203 | 0.219 | 0.187 | 0.4375 | 0.428 | 0.488 | 0.225 | 0.212 |
| $^5/_{16}$ | .5625 | .547 | .751 | .624 | .283 | .249 | .267 | .233 | .5625 | .552 | .629 | .273 | .258 |
| $^3/_8$ | .6250 | .606 | .832 | .691 | .346 | .310 | .331 | .295 | .6250 | .613 | .699 | .336 | .320 |
| $^7/_{16}$ | .7500 | .728 | 1.000 | .830 | .394 | .356 | .378 | .340 | .7500 | .737 | .840 | .384 | .366 |
| $^1/_2$ | .8125 | .788 | 1.082 | .898 | .458 | .418 | .442 | .402 | .8125 | .799 | .911 | .448 | .428 |
| $^9/_{16}$ | .8750 | .847 | 1.163 | .966 | .521 | .479 | .490 | .448 | .8750 | .861 | .982 | .511 | .489 |
| $^5/_8$ | 1.0000 | .969 | 1.330 | 1.104 | .569 | .525 | .553 | .509 | 1.0000 | .985 | 1.123 | .559 | .535 |
| $^3/_4$ | 1.1250 | 1.088 | 1.494 | 1.240 | .680 | .632 | .665 | .617 | 1.1250 | 1.108 | 1.263 | .670 | .642 |
| $^7/_8$ | 1.3125 | 1.269 | 1.742 | 1.447 | .792 | .740 | .776 | .724 | 1.3125 | 1.293 | 1.474 | .781 | .750 |
| 1 | 1.5000 | 1.450 | 1.991 | 1.653 | .903 | .847 | .887 | .831 | 1.5000 | 1.479 | 1.686 | .893 | .858 |
| $1\,^1/_8$ | 1.6875 | 1.631 | 2.239 | 1.859 | 1.030 | .970 | .999 | .939 | 1.6875 | 1.665 | 1.898 | 1.020 | .980 |
| $1\,^1/_4$ | 1.8750 | 1.813 | 2.489 | 2.067 | 1.126 | 1.062 | 1.063 | .999 | 1.8750 | 1.850 | 2.109 | 1.115 | 1.072 |
| $1\,^3/_8$ | 2.0625 | 1.994 | 2.738 | 2.273 | 1.237 | 1.169 | 1.175 | 1.107 | 2.0625 | 2.036 | 2.321 | 1.227 | 1.180 |
| $1\,^1/_2$ | 2.2500 | 2.175 | 2.986 | 2.480 | 1.349 | 1.277 | 1.286 | 1.214 | 2.2500 | 2.222 | 2.533 | 1.338 | 1.288 |
| $1\,^5/_8$ | 2.4375 | 2.356 | 3.235 | 2.686 | 1.460 | 1.384 | 1.397 | 1.321 | 2.4375 | 2.407 | 2.744 | 1.449 | 1.395 |
| $1\,^3/_4$ | 2.6250 | 2.538 | 3.485 | 2.893 | 1.571 | 1.491 | 1.509 | 1.429 | 2.6250 | 2.593 | 2.956 | 1.560 | 1.503 |
| $1\,^7/_8$ | 2.8125 | 2.719 | 3.733 | 3.100 | 1.683 | 1.599 | 1.620 | 1.536 | 2.8125 | 2.778 | 3.168 | 1.672 | 1.610 |
| 2 | 3.0000 | 2.900 | 3.982 | 3.306 | 1.794 | 1.706 | 1.732 | 1.643 | 3.0000 | 2.964 | 3.379 | 1.783 | 1.718 |
| $2\,^1/_4$ | 3.3750 | 3.263 | 4.480 | 3.720 | 2.017 | 1.921 | 1.954 | 1.858 | 3.3750 | 3.335 | 3.802 | 2.005 | 1.932 |
| $2\,^1/_2$ | 3.7500 | 3.625 | 4.977 | 4.133 | 2.240 | 2.136 | 2 146 | 2.042 | 3.7500 | 3.707 | 4.226 | 2.228 | 2.148 |
| $2\,^3/_4$ | 4.1250 | 3.988 | 5.476 | 4.546 | 2.462 | 2.350 | 2.369 | 2.257 | 4.1250 | 4.078 | 4 649 | 2.450 | 2.363 |
| 3 | 4.5000 | 4.350 | 5.973 | 4.959 | 2.685 | 2.565 | 2.591 | 2.471 | 4.5000 | 4.449 | 5.072 | 2.673 | 2 578 |
| **Heavy Nuts** | | | | | | | | | | | | | |
| $^1/_2$ | 0.8750 | 0.850 | 1.167 | 0.969 | 0.520 | 0.480 | 0.504 | 0.464 | 0.8750 | 0.862 | 0.983 | 0.511 | 0.490 |
| $^9/_{16}$ | 0.9375 | 0.906 | 1.244 | 1.033 | .584 | .542 | .568 | .526 | 0.9375 | 0.923 | 1.051 | .574 | .551 |
| $^5/_8$ | 1.0625 | 1.031 | 1.416 | 1.175 | .647 | .603 | .631 | .587 | 1.0625 | 1.047 | 1.194 | .637 | .613 |
| $^3/_4$ | 1.2500 | 1.213 | 1.665 | 1.383 | .774 | .726 | .758 | .710 | 1.2500 | 1.233 | 1.406 | .764 | .736 |
| $^7/_8$ | 1.4375 | 1.394 | 1.914 | 1.589 | .901 | .849 | .885 | .833 | 1.4375 | 1.418 | 1.617 | .891 | .859 |
| 1 | 1.6250 | 1.575 | 2.162 | 1.796 | 1.028 | .972 | 1.012 | .956 | 1.6250 | 1.604 | 1.829 | 1.018 | .982 |
| $1\,^1/_8$ | 1.8125 | 1.756 | 2.411 | 2.002 | 1.155 | 1.095 | 1.139 | 1.079 | 1.8125 | 1.790 | 2.041 | 1.145 | 1.105 |
| $1\,^1/_4$ | 2.0000 | 1.938 | 2.661 | 2.209 | 1.282 | 1.218 | 1.220 | 1.156 | 2.0000 | 1.975 | 2.252 | 1 272 | 1.228 |
| $1\,^3/_8$ | 2.1875 | 2.119 | 2.909 | 2.416 | 1.409 | 1.341 | 1.347 | 1.279 | 2.1875 | 2.161 | 2.464 | 1.399 | 1.351 |
| $1\,^1/_2$ | 2.3750 | 2.300 | 3.158 | 2.622 | 1.536 | 1.464 | 1.474 | 1.402 | 2.3750 | 2.347 | 2.676 | 1.526 | 1.475 |
| $1\,^5/_8$ | 2.5625 | 2.481 | 3.406 | 2.828 | 1.663 | 1.587 | 1.601 | 1.525 | 2.5625 | 2.532 | 2.886 | 1.652 | 1.598 |
| $1\,^3/_4$ | 2.7500 | 2.663 | 3.656 | 3.036 | 1.790 | 1.710 | 1.728 | 1.648 | 2.7500 | 2.718 | 3.099 | 1.779 | 1.721 |
| $1\,^7/_8$ | 2.9375 | 2.844 | 3.905 | 3.242 | 1.917 | 1.833 | 1.855 | 1.771 | 2.9375 | 2.903 | 3.309 | 1.906 | 1.844 |
| 2 | 3.1250 | 3.025 | 4.153 | 3.449 | 2.044 | 1.956 | 1.984 | 1.894 | 3.1250 | 3.089 | 3.522 | 2.033 | 1.967 |
| $2\,^1/_4$ | 3.5000 | 3.388 | 4.652 | 3.862 | 2.298 | 2.202 | 2.236 | 2.140 | 3.5000 | 3.460 | 3.944 | 2.287 | 2.213 |
| $2\,^1/_2$ | 3.8750 | 3.750 | 5.149 | 4.275 | 2.552 | 2.448 | 2.458 | 2.354 | 3.8750 | 3.832 | 4.368 | 2.541 | 2.460 |
| $2\,^3/_4$ | 4.2500 | 4.113 | 5.647 | 4.689 | 2.806 | 2.694 | 2.712 | 2.600 | 4.2500 | 4.203 | 4.791 | 2.794 | 2.706 |
| 3 | 4.6250 | 4.475 | 6.144 | 5.102 | 3.060 | 2.940 | 2.966 | 2.846 | 4.6250 | 4.574 | 5.214 | 3 048 | 2.952 |
| $3\,^1/_4$ | 5.0000 | 4.838 | 6.643 | 5.515 | 3.314 | 3.186 | 3.220 | 3.092 | 5.0000 | 4.945 | 5.637 | 3.302 | 3.198 |
| $3\,^1/_2$ | 5.3750 | 5.200 | 7.140 | 5.928 | 3.568 | 3.432 | 3.474 | 3.338 | 5.3750 | 5.317 | 6.061 | 3.556 | 3.445 |
| $3\,^3/_4$ | 5.7500 | 5.563 | 7.638 | 6.342 | 3.822 | 3.678 | 3.728 | 3.584 | 5.7500 | 5.688 | 6.484 | 3.809 | 3.691 |
| 4 | 6.1250 | 5.925 | 8.135 | 6.755 | 4.076 | 3.924 | 3.982 | 3.830 | 6.1250 | 6.059 | 6.907 | 4.063 | 3 937 |

The tolerance for the thickness of unfinished and semi-finished nuts is $\pm(0.016\,D + 0.012)$ from nominal. For finished nuts it is $\pm(0.015\,D + 0.003)$ from nominal. The top of square and hexagon nuts are washer crowned or flat and chamfered, the chamfer being the same as for bolt heads. Bearing surface of semi-finished nuts is machined plain with chamfered corners, or machined to a washer face 1/64 in. thick and of diameter equal to maximum width across flats; for finished nuts it is a washer face of the same dimensions as a semi-finished nut. Table 29 gives dimensions of regular and heavy nuts.

**Jam Nuts.**—Jam nuts are similar in all respects to the regular nuts, except thickness. The thicknesses of the several classes of jam nuts are as follows:

| Size, Regular | 1/4 to 7/16 | 1/2 to 1 1/8 | 1 1/4 to 2 | 2 1/2 to 3 |
|---|---|---|---|---|
| Unfinished | $(1/2\,D + 1/32)$ | $(1/2\,D + 1/16)$ | $(1/2\,D + 1/8)$ | $(1/2\,D + 1/4)$ |
| Semi-finished | $(1/2\,D + 3/64)$ | $(1/2\,D + 3/64)$ | $(1/2\,D + 1/16)$ | $(1/2\,D + 5/16)$ |
| Finished | $(1/2\,D + 1/32)$ | $(1/2\,D + 1/16)$ | $(1/2\,D + 1/8)$ | $(1/2\,D + 1/4)$ |

| Size, Heavy | 1/2 to 1 1/8 | 1 1/4 to 2 1/4 | 2 1/2 to 4 |
|---|---|---|---|
| Unfinished | $(1/2\,D + 1/16)$ | $(1/2\,D + 1/8)$ | $(1/2\,D + 1/4)$ |
| Semi-finished | $(1/2\,D + 3/64)$ | $(1/2\,D + 1/16)$ | $(1/2\,D + 5/16)$ |
| Finished | $(1/2\,D + 1/16)$ | $(1/2\,D + 1/8)$ | $(1/2\,D + 1/4)$ |

**Table 30.—Dimensions of American Standard Regular and Heavy Jam Nuts**

A.S.A. B18.2—1933. All dimensions in inches

| Diam., $D$ | Unfinished and Semi-finished Square and Hexagon | | | | | | | | Finished Hexagon | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Width Across Flats | | Width Across Corners, Min. | | Unfinished Thickness | | Semi-finished Thickness | | Width Across Flats | | Width Across Corners, Min. | Thickness | |
| | Max. (Basic) | Min. | Sq. | Hex. | Max. | Min. | Max. | Min. | Max. | Min. | | Max. | Min. |
| **Regular Nuts** | | | | | | | | | | | | | |
| 1/4 | 0.4375 | 0.425 | 0.584 | 0.485 | 0.172 | 0.140 | 0.157 | 0.125 | 0.4375 | 0.428 | 0.488 | 0.163 | 0.150 |
| 5/16 | .5625 | .547 | .751 | .624 | .205 | .171 | .189 | .155 | .5625 | .552 | .629 | .195 | .180 |
| 3/8 | .6250 | .606 | .832 | .691 | .237 | .201 | .221 | .185 | .6250 | .613 | .699 | .227 | .211 |
| 7/16 | .7500 | .728 | 1.000 | .830 | .269 | .231 | .253 | .215 | .7500 | .737 | .840 | .259 | .241 |
| 1/2 | .8125 | .788 | 1.082 | .898 | .333 | .293 | .317 | .277 | .8125 | .799 | .911 | .323 | .303 |
| 9/16 | .8750 | .847 | 1.163 | .966 | .365 | .323 | .349 | .307 | .8750 | .861 | .982 | .355 | .333 |
| 5/8 | 1.0000 | .969 | 1.330 | 1.104 | .397 | .353 | .381 | .337 | 1.0000 | .985 | 1.123 | .387 | .363 |
| 3/4 | 1.1250 | 1.088 | 1.494 | 1.240 | .462 | .414 | .446 | .398 | 1.1250 | 1.108 | 1.263 | .451 | .424 |
| 7/8 | 1.3125 | 1.269 | 1.742 | 1.447 | .526 | .474 | .510 | .458 | 1.3125 | 1.293 | 1.474 | .516 | .484 |
| 1 | 1.5000 | 1.450 | 1.991 | 1.653 | .591 | .535 | .575 | .519 | 1.5000 | 1.479 | 1.686 | .580 | .545 |
| 1 1/8 | 1.6875 | 1.631 | 2.239 | 1.859 | .655 | .595 | .639 | .579 | 1.6875 | 1.665 | 1.898 | .644 | .606 |
| 1 1/4 | 1.8750 | 1.813 | 2.489 | 2.067 | .782 | .718 | .720 | .656 | 1.8750 | 1.850 | 2.109 | .771 | .729 |
| 1 3/8 | 2.0625 | 1.994 | 2.738 | 2.273 | .847 | .779 | .784 | .716 | 2.0625 | 2.036 | 2.321 | .836 | .789 |
| 1 1/2 | 2.2500 | 2.175 | 2.986 | 2.480 | .911 | .839 | .849 | .777 | 2.2500 | 2.222 | 2.533 | .900 | .850 |
| 1 5/8 | 2.4375 | 2.356 | 3.235 | 2.686 | .976 | .900 | .913 | .837 | 2.4375 | 2.407 | 2.744 | .965 | .910 |
| 1 3/4 | 2.6250 | 2.538 | 3.485 | 2.893 | 1.040 | .960 | .978 | .898 | 2.6250 | 2.593 | 2.956 | 1.029 | .971 |
| 1 7/8 | 2.8125 | 2.719 | 3.733 | 3.100 | 1.105 | 1.021 | 1.042 | .958 | 2.8125 | 2.778 | 3.168 | 1.094 | 1.031 |
| 2 | 3.0000 | 2.900 | 3.982 | 3.306 | 1.169 | 1.081 | 1.107 | 1.019 | 3.0000 | 2.964 | 3 379 | 1.158 | 1.093 |
| 2 1/4 | 3 3750 | 3.263 | 4.480 | 3.720 | 1.298 | 1.202 | 1.236 | 1.140 | 3.3750 | 3.335 | 3.802 | 1.286 | 1.214 |
| 2 1/2 | 3.7500 | 3.625 | 4.977 | 4.133 | 1.552 | 1.448 | 1.458 | 1.354 | 3.7500 | 3.707 | 4.226 | 1.540 | 1.460 |
| 2 3/4 | 4.1250 | 3.988 | 5 476 | 4 546 | 1.681 | 1.569 | 1.587 | 1.475 | 4.1250 | 4.078 | 4.649 | 1.669 | 1.581 |
| 3 | 4.5000 | 4.350 | 5.973 | 4.959 | 1.810 | 1.690 | 1.716 | 1.596 | 4.5000 | 4.449 | 5.072 | 1.798 | 1.703 |
| **Heavy Nuts** | | | | | | | | | | | | | |
| 1/2 | 0.8750 | 0.850 | 1.167 | 0 969 | 0.333 | 0.293 | 0.317 | 0.277 | 0.8750 | 0 862 | 0.983 | 0.323 | 0.303 |
| 9/16 | .9375 | .906 | 1.244 | 1.033 | .365 | .323 | .349 | .307 | 0.9375 | 0.923 | 1.051 | .355 | .333 |
| 5/8 | 1.0625 | 1.031 | 1.416 | 1.175 | .397 | .353 | .381 | .337 | 1.0625 | 1.047 | 1.194 | .387 | .363 |
| 3/4 | 1.2500 | 1.213 | 1.665 | 1.383 | .462 | .414 | .446 | .398 | 1.2500 | 1.233 | 1.406 | .452 | .424 |
| 7/8 | 1.4375 | 1.394 | 1.914 | 1.589 | .526 | .474 | .510 | .458 | 1.4375 | 1.418 | 1.617 | .516 | .485 |
| 1 | 1.6250 | 1.575 | 2.162 | 1.796 | .591 | .535 | .575 | .519 | 1.6250 | 1.604 | 1.829 | .580 | .545 |
| 1 1/8 | 1.8125 | 1.756 | 2.411 | 2.002 | .655 | .595 | .639 | .579 | 1.8125 | 1.790 | 2.041 | .644 | .606 |
| 1 1/4 | 2.0000 | 1.938 | 2.661 | 2.209 | .782 | .718 | .720 | .656 | 2.0000 | 1.975 | 2.252 | .772 | .729 |
| 1 3/8 | 2.1875 | 2.119 | 2.909 | 2 416 | .847 | .779 | .784 | .716 | 2.1875 | 2.161 | 2.464 | .836 | .789 |
| 1 1/2 | 2.3750 | 2.300 | 3.158 | 2 622 | .911 | .839 | .849 | .777 | 2.3750 | 2.347 | 2.676 | .900 | .850 |
| 1 5/8 | 2.5625 | 2.481 | 3.406 | 2 828 | .976 | .900 | .913 | .837 | 2.5625 | 2.532 | 2.886 | .965 | .910 |
| 1 3/4 | 2.7500 | 2.663 | 3.656 | 3 036 | 1.040 | .960 | .978 | .898 | 2.7500 | 2.718 | 3.099 | 1.029 | .972 |
| 1 7/8 | 2.9375 | 2.844 | 3.905 | 3.242 | 1.105 | 1.021 | 1.042 | .958 | 2.9375 | 2.903 | 3.309 | 1.094 | 1.031 |
| 2 | 3.1250 | 3.025 | 4.153 | 3.449 | 1.169 | 1.081 | 1.106 | 1.018 | 3.1250 | 3.089 | 3.522 | 1.158 | 1.093 |
| 2 1/4 | 3.5000 | 3.388 | 4.652 | 3.862 | 1.298 | 1.202 | 1.235 | 1.139 | 3.5000 | 3.460 | 3.944 | 1.287 | 1.214 |
| 2 1/2 | 3.8750 | 3.750 | 5.149 | 4.275 | 1.552 | 1.448 | 1.458 | 1.354 | 3.8750 | 3.832 | 4.368 | 1.540 | 1.460 |
| 2 3/4 | .4.2500 | 4.113 | 5.647 | 4.689 | 1.681 | 1.569 | 1.587 | 1.475 | 4.2500 | 4.203 | 4.791 | 1.669 | 1.582 |
| 3 | 4.6250 | 4.475 | 6.144 | 5.102 | 1.810 | 1.690 | 1.716 | 1.596 | 4.6250 | 4.574 | 5.214 | 1.798 | 1.703 |
| 3 1/4 | 5.0000 | 4.838 | 6.643 | 5.515 | 1.939 | 1.811 | 1.845 | 1.717 | 5.0000 | 4.945 | 5.637 | 1.927 | 1.823 |
| 3 1/2 | 5.3750 | 5.200 | 7.140 | 5.928 | 2.068 | 1.932 | 1.974 | 1.838 | 5.3750 | 5.317 | 6.061 | 2.056 | 1.945 |
| 3 3/4 | 5.7500 | 5.563 | 7.638 | 6.342 | 2.197 | 2.053 | 2.103 | 1.959 | 5.7500 | 5.688 | 6.484 | 2.184 | 2.066 |
| 4 | 6.1250 | 5.925 | 8.135 | 6.755 | 2.326 | 2.174 | 2.232 | 2.080 | 6.1250 | 6.059 | 6.907 | 2.313 | 2.187 |

The tops and bottoms of double crown finished jam nuts are flat, and chamfered the same as finished regular nuts. Dimensions of jam nuts are given in Table 30.

**Light Nuts.**—Table 31 gives the proportions of American Standard light nuts and American Standard light castle nuts. The tolerance for width across flats is minus $(0.015\,D + 0.006)$ from basic. Minimum width across corners of hexagon is $(1.14 \times$ minimum width across flats$)$. Nominal thickness of a light nut is distance from top to bearing surface, and equals $^{7}/_{8}\,D$ adjusted to fractions. Tolerance for thickness is $\pm(0.015\,D + 0.003)$ from nominal. On light castle nuts the height of the top circle above flats is not less than the depth of the slot. On light nuts, the finished top is flat and chamfered at an angle of 30 deg. with top surface; diameter of top circle is minimum width across flats, with a tolerance of ± 15%. The bearing surface of both light nuts and light castle nuts is washer-faced; thickness of washer is $^{1}/_{64}$ in., and diameter is the maximum width across flats with a tolerance of ± 5%. The bearing surface must be at right angles to the axis of the threaded hole within a tolerance of 2 deg. for 1-in. nuts or smaller, and 1 deg. for nuts of over 1-in. size. Both light nuts and light castle nuts usually are supplied with fine threads. Semi-finished nuts are threaded, and are machined only on bearing surfaces.

**Table 31.—Dimensions of American Standard Light and Castle Nuts**

A.S.A., B18.2—1933. All dimensions in inches

| Diam. of Bolt, *D* | Width Across Flats | | Width Across Corners, Min. | Light Nuts | | Castle Nuts | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Thickness | | Thickness | | Nominal Height of Flats | Slot | |
| | Max. (Basic) | Min. | | Max. | Min. | Max. Overall | Min. Overall | | Width | Depth |
| 1/4 | 0.4375 | 0.428 | 0.488 | 0.226 | 0.212 | 0.288 | 0.275 | 3/16 | 5/64 | 3/32 |
| 5/16 | .5000 | .489 | .557 | .273 | .258 | .335 | .321 | 15/64 | 5/64 | 3/32 |
| 3/8 | .5625 | .551 | .628 | .337 | .320 | .414 | .398 | 9/32 | 1/8 | 1/8 |
| 7/16 | .6250 | .612 | .698 | .385 | .365 | .462 | .444 | 21/64 | 1/8 | 1/8 |
| 1/2 | .7500 | .737 | .840 | .448 | .427 | .573 | .553 | 3/8 | 1/8 | 3/16 |
| 9/16 | .8750 | .861 | .982 | .496 | .473 | .620 | .598 | 27/64 | 5/32 | 3/16 |
| 5/8 | .9375 | .922 | 1.051 | .559 | .535 | .731 | .707 | 15/32 | 5/32 | 1/4 |
| 3/4 | 1.0625 | 1.045 | 1.191 | .671 | .642 | .826 | .799 | 9/16 | 5/32 | 1/4 |
| 7/8 | 1.2500 | 1.231 | 1.403 | .782 | .750 | .921 | .891 | 21/32 | 5/32 | 1/4 |
| 1 | 1.4375 | 1.417 | 1.615 | .893 | .857 | 1.017 | .983 | 3/4 | 5/32 | 1/4 |
| 1 1/8 | 1.6250 | 1.602 | 1.826 | 1.004 | .965 | 1.173 | 1.139 | 27/32 | 7/32 | 5/16 |
| 1 1/4 | 1.8125 | 1.788 | 2.038 | 1.116 | 1.072 | 1.272 | 1.229 | 15/16 | 7/32 | 5/16 |
| 1 3/8 | 2.0000 | 1.973 | 2.249 | 1.227 | 1.180 | 1.402 | 1.348 | 1 | 7/32 | 5/16 |
| 1 1/2 | 2.1875 | 2.159 | 2.461 | 1.338 | 1.287 | 1.525 | 1.475 | 1 1/8 | 1/4 | 3/8 |

**Machine-screw and Stove-bolt Nuts.**—Dimensions are given in Table 32. The width across flats for sizes $^{1}/_{4}$ to $^{3}/_{8}$ in. equals $(1\,^{1}/_{2}\,D + {}^{1}/_{16})$ adjusted to sixteenths. Tops and bottoms of square nuts and bottoms of hexagon nuts are flat. In all other respects except thickness the nuts correspond to American Standard semi-finished regular nuts.

**Table 32.—American Standard Square and Hexagon Machine-screw and Stove-bolt Nuts**

A.S.A., B18.2—1933. All dimensions in inches

| Diameter of Screw, *D* | | Width Across Flats | | Width Across Corners of Hexagon, Min. | Thickness | |
|---|---|---|---|---|---|---|
| | | Max. (Basic) | Min. | | Max. | Min. |
| No. 0 | 0.0600 | 0.1562 | 0.150 | 0.171 | 0.050 | 0.043 |
| No. 1 | .0730 | .1562 | .150 | .171 | .050 | .043 |
| No. 2 | .0860 | .1875 | .180 | .205 | .066 | .057 |
| No. 3 | .0990 | .1875 | .180 | .205 | .066 | .057 |
| No. 4 | .1120 | .2500 | .241 | .275 | .098 | .087 |
| No. 5 | .1250 | .3125 | .302 | .344 | .114 | .102 |
| No. 6 | .1380 | .3125 | .302 | .344 | .114 | .102 |
| No. 8 | .1640 | .3437 | .332 | .378 | .130 | .117 |
| No. 10 | .1900 | .3750 | .362 | .413 | .130 | .117 |
| No. 12 | .2160 | .4375 | .423 | .482 | .161 | .148 |
| 1/4 | .2500 | .4375 | .423 | .482 | .193 | .178 |
| 5/16 | .3125 | .5625 | .545 | .621 | .225 | .208 |
| 3/8 | .3750 | .6250 | .607 | .692 | .257 | .239 |

**Table 33.—Dimensions of American Standard Wrench Openings**

A.S.A., B18.2—1933. All dimensions in inches

| Nominal Size of Wrench, also Basic Width Across Flats of Bolt Heads and Nuts | Allowance between Bolt Head or Nut and Jaws of Wrench | Wrench Opening | | Nominal Size of Wrench, also Basic Width Across Flats of Bolt Heads and Nuts | Allowance between Bolt Head or Nut and Jaws of Wrench | Wrench Opening | | Nominal Size of Wrench, also Basic Width Across Flats of Bolt Heads and Nuts | Allowance between Bolt Head or Nut and Jaws of Wrench | Wrench Opening | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Min. | Max. | | | Min. | Max. | | | Min. | Max. |
| 5/32 0.1562 | 0.002 | 0.158 | 0.163 | 1 1/4 1.2500 | 0.007 | 1.257 | 1.267 | 2 3/4 2.7500 | 0.015 | 2.765 | 2.783 |
| 3/16 .1875 | .002 | .189 | .194 | 1 5/16 1.3125 | .008 | 1.320 | 1.331 | 2 13/16 2.8125 | .015 | 2.827 | 2.845 |
| 1/4 .2500 | .002 | .252 | .257 | 1 3/8 1.3750 | .008 | 1.383 | 1.394 | 2 15/16 2.9375 | .016 | 2.953 | 2.972 |
| 5/16 .3125 | .003 | .315 | .321 | 1 7/16 1.4375 | .008 | 1.445 | 1.456 | 3 3.0000 | .016 | 3.016 | 3.035 |
| 11/32 .3437 | .003 | .347 | .353 | 1 1/2 1.5000 | .008 | 1.508 | 1.520 | 3 1/8 3.1250 | .017 | 3.142 | 3.162 |
| 3/8 .3750 | .003 | .378 | .384 | 1 5/8 1.6250 | .009 | 1.634 | 1.646 | 3 3/8 3.3750 | .018 | 3.393 | 3.414 |
| 7/16 .4375 | .003 | .440 | .446 | 1 11/16 1.6875 | .009 | 1.696 | 1.708 | 3 1/2 3.5000 | .018 | 3.518 | 3.540 |
| 1/2 .5000 | .003 | .503 | .509 | 1 13/16 1.8125 | .010 | 1.822 | 1.835 | 3 3/4 3.7500 | .020 | 3.770 | 3.793 |
| 9/16 .5625 | .004 | .566 | .573 | 1 7/8 1.8750 | .010 | 1.885 | 1.898 | 3 7/8 3.8750 | .020 | 3.895 | 3.918 |
| 5/8 .6250 | .004 | .629 | .636 | 2 2.0000 | .011 | 2.011 | 2.025 | 4 1/8 4.1250 | .022 | 4.147 | 4.172 |
| 3/4 .7500 | .005 | .755 | .763 | 2 1/16 2.0625 | .011 | 2.073 | 2.087 | 4 1/4 4.2500 | .022 | 4.272 | 4.297 |
| 13/16 .8125 | .005 | .817 | .825 | 2 3/16 2.1875 | .012 | 2.199 | 2.214 | 4 1/2 4.5000 | .024 | 4.524 | 4.550 |
| 7/8 .8750 | .005 | .880 | .888 | 2 1/4 2.2500 | .012 | 2.262 | 2.277 | 4 5/8 4.6250 | .024 | 4.649 | 4.676 |
| 15/16 .9375 | .006 | .943 | .952 | 2 3/8 2.3750 | .013 | 2.388 | 2.404 | 5 5.0000 | .026 | 5.026 | 5.056 |
| 1 1.0000 | .006 | 1.006 | 1.015 | 2 7/16 2.4375 | .013 | 2.450 | 2.466 | 5 3/8 5.3750 | .028 | 5.403 | 5.434 |
| 1 1/16 1.0625 | .006 | 1.068 | 1.077 | 2 9/16 2.5625 | .014 | 2.576 | 2.593 | 5 3/4 5.7500 | .030 | 5.780 | 5.813 |
| 1 1/8 1.1250 | .007 | 1.132 | 1.142 | 2 5/8 2.6250 | .014 | 2 639 | 2.656 | 6 1/8 6.1250 | .032 | 6.157 | 6 192 |

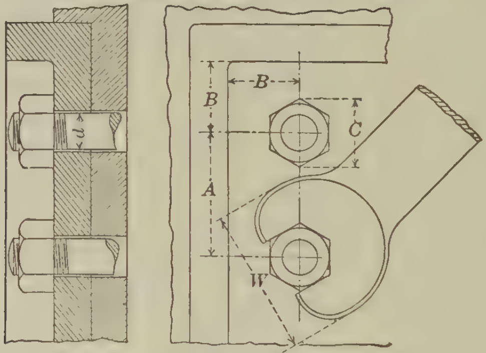


FIG. 15. Wrench Clearance Dimensions

**BOLT SPACING FOR WRENCH CLEARANCE.**—The following formulas are given in *Machinery's Handbook* for the minimum spacing between bolts, and between bolts and adjacent projections, to afford sufficient clearance for the usual form of open-end wrench.

$$A = 1/2\,W + 1/2\,C + 1/2\,d; \quad B = 0.48\,W.$$

See Fig. 15 for notation. All dimensions are in inches. A value of $W = (3.5\,d + 1/8)$ will cover the most of the commercial open-end wrenches. See Tables 29 to 32 for values of $C$.

**WRENCH HEAD OPENINGS** for standard bolts and nuts are given in Table 33. The minimum wrench openings provide a positive clearance between maximum nut and minimum wrench opening. Tolerances on wrench openings are plus only.

**STOVE BOLTS** are made with both round and countersink heads, in diameters of 1/8, 5/32, 3/16, 7/32, 1/4, 5/16 and 3/8 in. They may be obtained in practically any length desired, and usually are threaded the entire length. See Fig. 16.

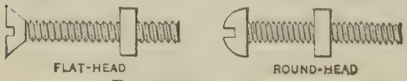


FIG. 16. Stove Bolts

**Table 34.—Size and Dimensions of Stove Bolts and Nuts**

| Size, in. | 1/8 | 5/32 | 3/16 | 7/32 | 1/4 | 5/16 | 3/8 |
|---|---|---|---|---|---|---|---|
| Threads per in. | 32 | 30 | 24 | 24 | 18 | 18 | 16 |
| Long Diam., square nut, in. | 0.398 | 0.485 | 0.574 | 0.619 | 0.707 | 0.796 | 0.884 |
| Long. Diam., hexagon nut, in. | 0.325 | 0.397 | 0.469 | 0.505 | 0.577 | 0.649 | 0.722 |
| Short Diam. of nut, in. | 9/32 | 11/32 | 13/32 | 7/16 | 1/2 | 9/16 | 5/8 |
| Thickness of nut, in. | 0.093 | 0.110 | 0.125 | 0.156 | 0.1875 | 0.218 | 0.250 |

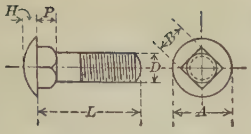


FIG. 17. Square Neck Carriage Bolt

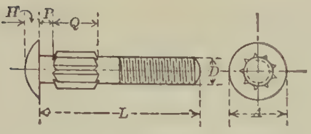


FIG. 18. Ribbed Carriage Bolt

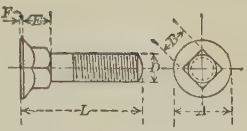


FIG. 19. Countersunk Carriage Bolt

**Table 35.—Dimensions of American Standard Carriage Bolts**

A.S.A. B18e–1928. All dimensions in inches

| Size, $D$ | No. 10 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 3/4 |
|---|---|---|---|---|---|---|---|---|---|
| Threads per in. | 24 | 20 | 18 | 16 | 14 | 13 | 12 | 11 | 10 |
| Square Neck Carriage Bolts (Fig. 17) | | | | | | | | | |
| Diam. of head, $A$ | 7/16 | 9/16 | 11/16 | 13/16 | 15/16 | 1 1/16 | 1 3/16 | 1 5/16 | 1 9/16 |
| Height of head, $H$ | 3/32 | 1/8 | 5/32 | 3/16 | 7/32 | 1/4 | 9/32 | 5/16 | 3/8 |
| Depth of square, $P$ | 3/16 | 7/32 | 1/4 | 9/32 | 5/16 | 11/32 | 3/8 | 13/32 | 15/32 |
| Width of square, $B$ | 0.19 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 3/4 |
| Length $L$, min.* | 1/2 | 1/2 | 3/4 | 3/4 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/2 |
| " $L$, max.* | 3 1/2 | 4 1/2 | 6 1/2 | 8 | 10 | 10 | 10 | 10 | 10 |
| Ribbed Carriage Bolts (Fig. 18) | | | | | | | | | |
| Diam. of head, $A$ | 7/16 | 9/16 | 11/16 | 13/16 | 15/16 | 1 1/16 | ....... | ....... | ....... |
| Height of head, $H$ | 3/32 | 1/8 | 5/32 | 3/16 | 7/32 | 1/4 | ....... | ....... | ....... |
| Distance of Rib below head, $P$ | 3/32 | 3/32 | 3/32 | 3/32 | 3/32 | 3/32 | ....... | ....... | ....... |
| Length of Rib, $Q$, $L \leqq 1\ 1/8$ | 3/8 | 3/8 | 3/8 | 3/8 | 3/8 | 3/8 | ....... | ....... | ....... |
| " " " $Q$, $L \geqq 1\ 1/4$ | 1/2 | 1/2 | 1/2 | 1/2 | 1/2 | 1/2 | ....... | ....... | ....... |
| No. of Ribs, max. | 9 | 10 | 12 | 12 | 14 | 16 | ....... | ....... | ....... |
| Length $L$, min.* | 1 | 1 | 1 | 1 | 1 | 1 | ....... | ....... | ....... |
| " $L$, max.* | 3 1/2 | 4 1/2 | 6 1/2 | 8 | 10 | 10 | ....... | ....... | ....... |
| Countersunk Head Carriage Bolts (Fig. 19) | | | | | | | | | |
| Diam. of head, $A$ | ....... | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 5/8 |
| Feed thickness, $F$ | ....... | 1/64 | 1/32 | 1/32 | 1/32 | 1/32 | 1/32 | 1/32 | 3/64 |
| Depth of square, $E$ | ....... | 9/32 | 11/32 | 13/32 | 15/32 | 17/32 | 19/32 | 21/32 | 25/32 |
| Width of square, $B$ | ....... | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 3/4 |
| Length $L$, min.* | ....... | 3/4 | 3/4 | 3/4 | 3/4 | 1 | 1 1/4 | 1 1/4 | 1 1/2 |
| " $L$ max.* | ....... | 4 1/2 | 6 1/2 | 8 | 10 | 10 | 10 | 10 | 10 |

* Lengths increase by increments of 1/4 in. from 1/2 to 4 in. inclusive; by increments of 1/2 in. from 4 1/2 to 10 in. inclusive.

**MACHINE SCREWS.**—Various forms of American Standard machine screws are shown in Fig. 20. The unthreaded body diameter will have approximately the same tolerances as the pitch diameter of American Standard screw threads, Class 2 free fit, for corresponding number of threads per inch. See p. 9–07. The formulas for determining the various dimensions of the several types of screws are as follows: Head diameter, flat head and oval head, $A_{\max} = (2.040\,D - 0.003)$; $A_{\min} = (1.960\,D - 0.013)$; round head, $A_{\max} = 1.887\,D$, $A_{\min} = (1.813\,D - 0.010)$; fillister head, $A_{\max} = (1.670\,D - 0.004)$; $A_{\min} = (1.610\,D - 0.014)$. Height of head, flat head and oval head, $H_{\max} = (0.619\,D - 0.002)$; $H_{\min} = (0.552\,D - 0.007)$; round head, $H_{\max} = (0.636\,D + 0.015)$; $H_{\min} = (0.624\,D + 0.005)$; fillister head, $H_{\max} = (0.660\,D - 0.002)$; $H_{\min} = (0.600\,D - 0.007)$. Width of slot, all types, $J_{\max} = (0.182\,D + 0.020)$; $J_{\min} = (0.164\,D + 0.010)$. Depth of slot, flat head, $T_{\max} = (0.288\,D - 0.002)$; $T_{\min} = (0.192\,D - 0.002)$; round head, $T_{\max} = (0.362\,D + 0.017)$; $T_{\min} = (0.268\,D + 0.013)$; oval head, $T_{\max} = (0.556\,D - 0.003)$; $T_{\min} = (0.460\,D - 0.003)$; fillister head, $T_{\max} = (0.440\,D - 0.001)$; $T_{\min} = (0.374\,D - 0.011)$. Height of oval, fillister head, $F_{\max} = (0.280\,D + 0.004)$; $F_{\min} = (0.220\,D - 0.001)$; oval head, $F_{\max} = (0.304\,D + 0.003)$; $F_{\min} = (0.268\,D - 0.001)$. Countersink angle of both flat and oval head screws is 82 deg. max., 80 deg. min. Table 36 shows the dimension of American Standard machine screw heads. The preferred lengths of the various sizes range as follows, although all lengths are not regularly manufactured for all types of head.

Coarse Thread Series:

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal Size | 2 | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 1/4 | 5/16 | 3/8 |
| Length $L$, in. | 1/8–7/8 | 1/8–7/8 | 1/8–1 1/2 | 1/8–1 3/4 | 1/8–2 1/4 | 1/8–3 | 3/16–3 | 1/4–3 | 5/16–3 | 1/2–3 | 1/2–3 |

Fine Thread Series:

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Nominal Size | 2 | 3 | 4 | 6 | 8 | 10 | 12 | 1/4 | 5/16 | 3/8 |
| Length, $L$, in. | 1/8–1/2 | 1/8–3/4 | 1/8–3/4 | 1/8–1 1/2 | 3/16–3 | 3/16–3 | 5/16–3 | 5/16–3 | 1/2–3 | 5/8–3 |

### Table 36.—Dimensions of American Standard Machine Screw Heads

A.S.A., B18c—1930. All dimensions in inches. See Fig. 20

| Nominal Size | Max. Diam., $D$ | Head Diam., min., $A$ | Height of Head, min., $H$ | Width of Slot, min., $J$ | Depth of Slot, min., $T$ | Height of Oval, min., $F$ | Head Diam., min., $A$ | Height of Head, min., $H$ | Width of Slot, min., $J$ | Depth of Slot, min., $T$ | Height of Oval, min., $F$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Flat Head | | | | | Round Head | | | | |
| 2 | 0.086 | 0.156 | 0.040 | 0.024 | 0.015 | ....... | 0.146 | 0.059 | 0.024 | 0.036 | ....... |
| 3 | .099 | .181 | .048 | .026 | .017 | ....... | .169 | .067 | .026 | .040 | ....... |
| 4 | .112 | .207 | .055 | .028 | .020 | ....... | .193 | .075 | .028 | .043 | ....... |
| 5 | .125 | .232 | .062 | .031 | .022 | ....... | .217 | .083 | .031 | .047 | ....... |
| 6 | .138 | .257 | .069 | .033 | .024 | ....... | .240 | .091 | .033 | .050 | ....... |
| 8 | .164 | .308 | .084 | .037 | .029 | ....... | .287 | .107 | .037 | .057 | ....... |
| 10 | .190 | .359 | .098 | .041 | .034 | ....... | .334 | .124 | .041 | .064 | ....... |
| 12 | .216 | .410 | .112 | .045 | .039 | ....... | .382 | .140 | .045 | .071 | ....... |
| 1/4 | .250 | .477 | .131 | .051 | .046 | ....... | .443 | .161 | .051 | .080 | ....... |
| 5/16 | .3125 | .600 | .166 | .061 | .058 | ....... | .557 | .200 | .061 | .097 | ....... |
| 3/8 | .375 | .722 | .200 | .072 | .070 | ....... | .670 | .239 | .072 | .114 | ....... |
| | | Oval Head | | | | | Fillister Head | | | | |
| 2 | 0.086 | 0.156 | 0.040 | 0.024 | 0.037 | 0.022 | 0.124 | 0.045 | 0.024 | 0.021 | 0.018 |
| 3 | .099 | .181 | .048 | .026 | .043 | .026 | .145 | .052 | .026 | .026 | .021 |
| 4 | .112 | .207 | .055 | .028 | .049 | .029 | .166 | .060 | .028 | .031 | .024 |
| 5 | .125 | .232 | .062 | .031 | .055 | .033 | .187 | .068 | .031 | .036 | .027 |
| 6 | .138 | .257 | .069 | .033 | .060 | .036 | .208 | .076 | .033 | .041 | .029 |
| 8 | .164 | .308 | .084 | .037 | .072 | .043 | .250 | .091 | .037 | .050 | .035 |
| 10 | .190 | .359 | .098 | .041 | .084 | .050 | .292 | .107 | .041 | .060 | .041 |
| 12 | .216 | .410 | .112 | .045 | .096 | .057 | .334 | .123 | .045 | .070 | .047 |
| 1/4 | .250 | .477 | .131 | .051 | .112 | .066 | .389 | .143 | .051 | .083 | .054 |
| 5/16 | .3125 | .600 | .166 | .061 | .141 | .083 | .490 | .181 | .061 | .106 | .068 |
| 3/8 | .375 | .722 | .200 | .072 | .170 | .100 | .590 | .218 | .072 | .129 | .082 |

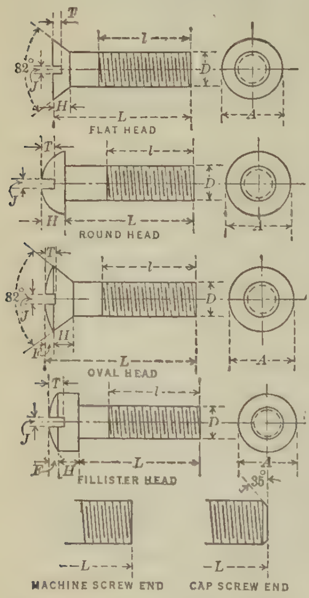


Fig. 20. Machine and Cap Screws

The lengths advance by increments of $^1/_{16}$ in. from $^1/_8$ to $^1/_2$ in.; by increments of $^1/_8$ in. from $^5/_8$ to 1 in.; and by increments of $^1/_4$ in. from 1 $^1/_4$ to 3 in.

SLOTTED HEAD CAP SCREWS are made with flat, button and fillister heads. The unthreaded body diameter has the same tolerances as the major thread diameter of American Standard screw threads, Class 2 free fit, for corresponding number of threads. See p. 9-07. The formulas for the various dimensions of the several types are as follows: Head diameter, $A_{\text{max}}$, all types, = nominal diam. (no formula). Minimum diameters are: flat head, $A_{\text{min}} = A_{\text{max}} - (0.03\ A_{\text{max}} + 0.008)$; button head, $A_{\text{min}} = A_{\text{max}} - (0.02\ A_{\text{max}} + 0.010)$; fillister head, $A_{\text{min}} = A_{\text{max}} - (0.02\ A_{\text{max}} + 0.004)$. Height of head: Flat head, nominal $H$ is computed from $D_{\text{max}}$ and $A_{\text{max}}$ with 81 deg. angle; button head, $H_{\text{max}} = {}^7/_{16}\ A_{\text{max}}$; $H_{\text{min}} = H_{\text{max}} - (0.03\ A_{\text{max}} + 0.003)$; fillister head, $H_{\text{max}} = {}^2/_3\ D$ to nearest sixty-fourth in.; $H_{\text{min}} = H_{\text{max}} - (0.03\ A_{\text{max}} + 0.004)$. Width of slot, all types, $J_{\text{max}} = S + (0.05\ S + 0.003)$; $J_{\text{min}} = S - (0.05\ S + 0.003)$, where $S$ = approximate cutter width = $(0.160\ D + 0.024)$. Depth of slot: Flat head $T_{\text{max}} = 0.500\ H$ nominal; $T_{\text{min}} = (T_{\text{max}} - 0.08\ D)$; button head, $T_{\text{max}} = {}^2/_3\ H_{\text{min}}$; $T_{\text{min}} = (T_{\text{max}} - 0.08\ D)$; fillister head, $T_{\text{max}} = (0.50\ F_{\text{min}} + H_{\text{min}})$; $T_{\text{min}} = (T_{\text{max}} - 0.08\ D)$. Height of oval, fillister head, $F_{\text{max}} = (0.110\ A_{\text{max}} + 0.002)$; $F_{\text{min}} = 0.100\ A_{\text{max}}$.

The length of a cap screw is measured from the largest diameter of bearing surface of the head to the extreme point on a line parallel to the axis of the screw. The difference between consecutive lengths of screws is as follows: Screws $^1/_4$ to 1 in., $^1/_8$ in.; 1 to 4 in., $^1/_4$ in.; 4 to 6 in., $^1/_2$ in. The threads are coarse pitch and the length of thread $l = 2\ D + {}^1/_4$. Dimensions of American Standard cap screws are given in Table 37.

### Table 37.—Dimensions of American Standard Cap Screws

A.S.A., B18c—1930. All dimensions in inches. See Fig. 20

| Nominal Size | Max. Diam., $D$ | Head, $A$ |  | Height of Head, $H$ |  | Width of Slot, $J$ |  | Depth of Slot, $T$ |  | Height of Oval, $F$ |  |
|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. |
| **Flat Head** |  |  |  |  |  |  |  |  |  |  |  |
| 1/4 | 0.2500 | 1/2 | 0.477 | 0.146 | ....... | 0.070 | 0.058 | 0.073 | 0.053 | ....... | ....... |
| 5/16 | .3125 | 5/8 | .598 | .183 | ....... | .079 | .065 | .091 | .066 | ....... | ....... |
| 3/8 | .3750 | 3/4 | .719 | .220 | ....... | .088 | .074 | .110 | .080 | ....... | ....... |
| 7/16 | .4375 | 13/16 | .780 | .220 | ....... | .098 | .083 | .110 | .075 | ....... | ....... |
| 1/2 | .5000 | 7/8 | .841 | .220 | ....... | .110 | .094 | .110 | .070 | ....... | ....... |
| 9/16 | .5625 | 1 | .962 | .256 | ....... | .123 | .106 | .128 | .083 | ....... | ....... |
| 5/8 | .6250 | 1 1/8 | 1.083 | .293 | ....... | .138 | .119 | .146 | .096 | ....... | ....... |
| 3/4 | .7500 | 1 3/8 | 1.326 | .366 | ....... | .154 | .134 | .183 | .123 | ....... | ....... |
| **Button Head** |  |  |  |  |  |  |  |  |  |  |  |
| 1/4 | 0.2500 | 7/16 | 0.418 | 0.191 | 0.175 | 0.070 | 0.058 | 0.117 | 0.097 | ....... | ....... |
| 5/16 | .3125 | 9/16 | .541 | .246 | .226 | .079 | .065 | .151 | .126 | ....... | ....... |
| 3/8 | .3750 | 5/8 | .602 | .273 | .251 | .088 | .074 | .167 | .137 | ....... | ....... |
| 7/16 | .4375 | 3/4 | .725 | .328 | .302 | .098 | .083 | .202 | .167 | ....... | ....... |
| 1/2 | .5000 | 13/16 | .786 | .355 | .328 | .110 | .094 | .219 | .179 | ....... | ....... |
| 9/16 | .5625 | 15/16 | .908 | .410 | .379 | .123 | .106 | .253 | .208 | ....... | ....... |
| 5/8 | .6250 | 1 | .970 | .438 | .405 | .138 | .119 | .270 | .220 | ....... | ....... |
| 3/4 | .7500 | 1 1/4 | 1.215 | .547 | .506 | .154 | .134 | .337 | .277 | ....... | ....... |
| **Fillister Head** |  |  |  |  |  |  |  |  |  |  |  |
| 1/4 | 0.2500 | 3/8 | 0.363 | 11/64 | 0.157 | 0.070 | 0.058 | 0.097 | 0.077 | 0.044 | 0.038 |
| 5/16 | .3125 | 7/16 | .424 | 13/64 | .186 | .079 | .065 | .115 | .090 | .050 | .044 |
| 3/8 | .3750 | 9/16 | .547 | 1/4 | .229 | .088 | .074 | .142 | .112 | .064 | .056 |
| 7/16 | .4375 | 5/8 | .608 | 19/64 | .274 | .098 | .083 | .168 | .133 | .071 | .063 |
| 1/2 | .5000 | 3/4 | .731 | 21/64 | .301 | .110 | .094 | .188 | .148 | .084 | .075 |
| 9/16 | .5625 | 13/16 | .792 | 3/8 | .347 | .123 | .106 | .214 | .169 | .091 | .081 |
| 5/8 | .6250 | 7/8 | .853 | 27/64 | .392 | .138 | .119 | .240 | .190 | .099 | .088 |
| 3/4 | .7500 | 1 | .976 | 1/2 | .466 | .154 | .134 | .283 | .233 | .112 | .100 |
| 7/8 | .8750 | 1 1/8 | 1.098 | 19/32 | .556 | .173 | .151 | .334 | .264 | .126 | .113 |
| 1 | 1.0000 | 1 5/16 | 1.282 | 21/32 | .613 | .194 | .170 | .372 | .292 | .146 | .131 |

### Table 38.—Dimensions of American Standard Hexagon-head Cap Screws and Square-head Set Screws

A.S.A., B18.2—1933. All dimensions in inches

| Diam. of Screw, $D$ | Hexagon-head Cap Screws |  |  |  |  | Square-head Set-screws |  |  |  |
|---|---|---|---|---|---|---|---|---|---|
|  | Width Across Flats |  | Width Across Corners, Min. | Height |  | Width Across Flats |  | Height |  |
|  | Max. (Basic) | Min. |  | Max. | Min. | Max. (Basic) | Min. | Max. | Min. |
| 1/4 | 0.4375 | 0.428 | 0.488 | 0.194 | 0.181 | 0.2500 | 0.241 | 0.197 | 0.179 |
| 5/16 | .5000 | .489 | .557 | .242 | .227 | .3125 | .302 | .245 | .224 |
| 3/8 | .5625 | .551 | .628 | .289 | .273 | .3750 | .362 | .293 | .270 |
| 7/16 | .6250 | .612 | .698 | .338 | .319 | .4375 | .423 | .341 | .315 |
| 1/2 | .7500 | .737 | .840 | .386 | .365 | .5000 | .484 | .389 | .361 |
| 9/16 | .8125 | .798 | .910 | .432 | .410 | .5625 | .545 | .437 | .407 |
| 5/8 | .8750 | .860 | .980 | .481 | .456 | .6250 | .607 | .485 | .452 |
| 3/4 | 1.0000 | .983 | 1.121 | .577 | .548 | .7500 | .729 | .582 | .544 |
| 7/8 | 1.1250 | 1.106 | 1.261 | .672 | .640 | .8750 | .852 | .678 | .635 |
| 1 | 1.3125 | 1.292 | 1.473 | .768 | .732 | 1.0000 | .974 | .774 | .726 |
| 1 1/8 | 1.5000 | 1.477 | 1.684 | .864 | .824 | 1.1250 | 1.097 | .870 | .817 |
| 1 1/4 | 1.6875 | 1.663 | 1.896 | .959 | .916 | 1.2500 | 1.219 | .967 | .909 |
| 1 3/8 | ......... | ......... | ......... | ......... | ......... | 1.3750 | 1.342 | 1.063 | 1.000 |
| 1 1/2 | ......... | ......... | ......... | ......... | ......... | 1.5000 | 1.464 | 1.159 | 1.091 |

**Hexagon-head Cap Screws** are machined on all surfaces. The width across the flat of 1/4- to 7/16-in. screws is $(D + 3/16)$; for 1/2- to 7/8-in. screws, $(D + 1/4)$; for 1-in. screws, $(D + 5/16)$; for 1 1/8 in. screws $(D + 3/8)$; for 1 1/4 in. screws, $(D + 7/16)$. The tolerance for width across flat is minus $(0.015\,D + 0.003)$ from nominal. The tops and bearing faces correspond to American Standard regular finished hexagon bolt heads. Dimensions are given in Table 38.

**Socket-head Cap Screws** are shown in Fig. 21, and dimensions are given in Table 39. The body diameter $D$ refers to the unthreaded portion and is the nominal diameter of the screw. The thread may be either American National Standard coarse thread or fine thread. The heads are flat and chamfered at an angle $E$ of 30 deg. ± 2 deg. The socket depth is made as great as practicable. The length of thread $l$ is measured from the extreme point to the last usable thread and is as follows: For American National coarse thread, $l = 2\,D + 1/2$ in., where $l > 1/2\,L$; $l = 1/2\,L$ where $l > 2\,D + 1/2$ in. For American National fine thread, $l = 1\,1/2\,D + 1/2$ in. where $l > 3/8\,L$; $l = 3/8\,L$ where $l > 1\,1/2\,D + 1/2$ in.

The point chamfer $Z$ is an angle of $35°, +5°, -0°$.

### Table 39.—Dimensions of American Standard Socket-head Cap Screws

A.S.A., B18.3—1936. All dimensions in inches. See Fig. 21

| Size or Diam., D | | Head Dimensions | | | | | | Socket Dimensions | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diam., A | | Height, H | | Side Height, S | | Hexagon | | Fluted | | | |
| | | | | | | | | Width Across Flats, J | | No. of Flutes | Minor Diam., K | Major Diam., M | Land Width, N |
| Nominal | Max. | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | | Max. * | Max. * | Max. * |
| No. 8 | 0.1640 | 9/32 | 0.276 | 0.164 | 0.160 | 0.1522 | 0.1484 | 0.1270 | 1/8 | 6 | 0.127 | 0.147 | 0.035 |
| " 10 | .1900 | 5/16 | .306 | .190 | .185 | .1765 | .1717 | .1582 | 5/32 | 6 | .127 | .147 | .035 |
| " 12 | .2160 | 11/32 | .337 | .216 | .211 | .2005 | .1957 | .1582 | 5/32 | 6 | .160 | .185 | .042 |
| 1/4 | .2500 | 3/8 | .367 | 1/4 | .244 | .2317 | .2265 | .1895 | 3/16 | 6 | .190 | .219 | .052 |
| 5/16 | .3125 | 7/16 | .429 | 5/16 | .306 | .2894 | .2834 | .2207 | 7/32 | 6 | .221 | .256 | .062 |
| 3/8 | .3750 | 9/16 | .553 | 3/8 | .368 | .3469 | .3405 | .3155 | 5/16 | 6 | .312 | .380 | .092 |
| 7/16 | .4375 | 5/8 | .615 | 7/16 | .430 | .4046 | .3974 | .3155 | 5/16 | 6 | .312 | .380 | .092 |
| 1/2 | .5000 | 3/4 | .739 | 1/2 | .492 | .4620 | .4546 | .3780 | 3/8 | 6 | .386 | .463 | .112 |
| 9/16 | .5625 | 13/16 | .801 | 9/16 | .554 | .5196 | .5116 | .3780 | 3/8 | 6 | .386 | .463 | .112 |
| 5/8 | .6250 | 7/8 | .863 | 5/8 | .616 | .5771 | .5687 | .5030 | 1/2 | 6 | .506 | .600 | .142 |
| 3/4 | .7500 | 1 | .987 | 3/4 | .741 | .6920 | .6830 | .5655 | 9/16 | 6 | .568 | .654 | .157 |
| 7/8 | .8750 | 1 1/8 | 1.111 | 7/8 | .865 | .8069 | .7971 | .5655 | 9/16 | 6 | .568 | .654 | .157 |
| 1 | 1.0000 | 1 5/16 | 1.297 | 1 | .989 | .9220 | .9112 | .6290 | 5/8 | 6 | .631 | .790 | .184 |
| 1 1/8 | 1.1250 | 1 1/2 | 1.483 | 1 1/8 | 1.113 | 1.0372 | 1.0254 | .7540 | 3/4 | 6 | .756 | .957 | .221 |
| 1 1/4 | 1.2500 | 1 3/4 | 1.733 | 1 1/4 | 1.238 | 1.1516 | 1.1398 | .7540 | 3/4 | 6 | .756 | .957 | .221 |
| 1 3/8 | 1.3750 | 1 7/8 | 1.855 | 1 3/8 | 1.361 | 1.2675 | 1.2533 | .7540 | 3/4 | 6 | .756 | .957 | .221 |
| 1 1/2 | 1 5000 | 2 | 1.979 | 1 1/2 | 1.485 | 1.3821 | 1.3679 | 1.0040 | 1 | 6 | 1 007 | 1.275 | .298 |

* Minimum diameter is 0.002 in. less than maximum diameter for sizes No. 8 to 7/16 inclusive; 0.003 in. less for sizes 1/2 to 7/8 in. inclusive; 0.004 in. less for sizes 1 to 1 1/2 in. inclusive.

### Table 40.—Dimensions of American Standard Socket Set-screws

A.S.A., B18.3—1936. All dimensions in inches. See Fig. 22

| Size, Nominal Diam. D | Point Dimensions | | | | | | | | Socket Dimensions | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Cup and Flat Point Screws | | Oval Point Screws | Cone Point Screws | | Dog Point Screws | | | Hexagon | | Fluted | | | |
| | | | | Cone Point Angle Y | | | | | | | | | | |
| | Point Diam. C | | Point Radius R | 118°±2° for lengths up to and incl. | 90° ± 2° for lengths over to and incl. | Diam. P Max. | Point Length | | Width Across Flats, J | | No. of Flutes | Minor Diam. K Max. * | Major Diam. M Max. * | Land Width N Max. * |
| | Mean | Max. | | | | | Full Q | Half q | Max. | Min. | | | | |
| No. 5 | 1/16 | 0.067 | 3/32 | 1/8 | 3/16 | 0.083 | 0.06 | 0.03 | 0.0635 | 1/16 | 4 | 0.053 | 0.071 | 0.022 |
| " 6 | 0.069 | .074 | 7/64 | 1/8 | 3/16 | .092 | .07 | .03 | .0635 | 1/16 | 4 | .056 | .079 | .025 |
| " 8 | 5/64 | .087 | 1/8 | 3/16 | 1/4 | .109 | .08 | .04 | .0791 | 5/64 | 6 | .079 | .098 | .022 |
| " 10 | 3/32 | .102 | 9/64 | 3/16 | 1/4 | .127 | .09 | .04 | .0947 | 3/32 | 6 | .097 | .113 | .027 |
| " 12 | 7/64 | .115 | 5/32 | 3/16 | 1/4 | .144 | .11 | .06 | .0947 | 3/32 | 6 | .097 | .113 | .027 |
| 1/4 | 1/8 | .132 | 3/16 | 1/4 | 5/16 | 5/32 | 1/8 | 1/16 | .1270 | 1/8 | 6 | .127 | .147 | .032 |
| 5/16 | 11/64 | .172 | 15/64 | 5/16 | 3/8 | 13/64 | 5/32 | 5/64 | .1582 | 5/32 | 6 | .160 | .185 | .041 |
| 3/8 | 13/64 | .212 | 9/32 | 3/8 | 7/16 | 1/4 | 3/16 | 3/32 | .1895 | 3/16 | 6 | .190 | .219 | .052 |
| 7/16 | 15/64 | .252 | 21/64 | 7/16 | 1/2 | 19/64 | 7/32 | 7/64 | .2207 | 7/32 | 6 | .221 | .256 | .062 |
| 1/2 | 9/32 | .291 | 3/8 | 1/2 | 9/16 | 11/32 | 1/4 | 1/8 | .2520 | 1/4 | 6 | .254 | .297 | .072 |
| 9/16 | 5/16 | .332 | 27/64 | 9/16 | 5/8 | 25/64 | 9/32 | 9/64 | .2520 | 1/4 | 6 | .254 | .297 | .072 |
| 5/8 | 23/64 | .371 | 15/32 | 5/8 | 3/4 | 15/32 | 5/16 | 5/32 | .3155 | 5/16 | 6 | .315 | .380 | .092 |
| 3/4 | 7/16 | .450 | 9/16 | 3/4 | 7/8 | 9/16 | 3/8 | 3/16 | .3780 | 3/8 | 6 | .386 | .463 | .112 |
| 7/8 | 33/64 | .530 | 21/32 | 7/8 | 1 | 21/32 | 7/16 | 7/32 | .5030 | 1/2 | 6 | .506 | .600 | .142 |
| 1 | 19/32 | .609 | 3/4 | 1 | 1 1/8 | 3/4 | 1/2 | 1/4 | .5655 | 9/16 | 6 | .568 | .654 | .157 |
| 1 1/8 | 43/64 | .689 | 27/32 | 1 1/8 | 1 1/4 | 27/32 | 9/16 | 9/32 | .5655 | 9/16 | 6 | .568 | .654 | .157 |
| 1 1/4 | 3/4 | .767 | 15/16 | 1 1/4 | 1 1/2 | 15/16 | 5/8 | 5/16 | .6290 | 5/8 | 6 | .631 | .790 | .184 |
| 1 3/8 | 53/64 | .848 | 1 1/32 | 1 3/8 | 1 5/8 | 1 1/32 | 11/16 | 11/32 | .6290 | 5/8 | 6 | .631 | .790 | .184 |
| 1 1/2 | 29/32 | .926 | 1 1/8 | 1 1/2 | 1 3/4 | 1 1/8 | 3/4 | 3/8 | .7540 | 3/4 | 6 | .756 | .958 | .221 |
| 1 3/4 | 1 1/16 | 1.086 | 1 5/16 | 1 3/4 | 2 | 1 5/16 | 7/8 | 7/16 | 1.0040 | 1 | 6 | 1.007 | 1.275 | .298 |
| 2 | 1 7/32 | 1 244 | 1 1/2 | 2 | 2 1/4 | 1 1/2 | 1 | 1/2 | 1.0040 | 1 | 6 | 1.007 | 1.275 | .298 |

* Minimum diameter is 0.001 in. less than maximum diameter for sizes No. 5 to 8 inclusive; 0.002 in. less for sizes No. 10 to 9/16 in. inclusive; 0.003 in. less for sizes 5/8 to 7/8 in. inclusive; 0.004 in. less for sizes 1 to 2 in. inclusive.

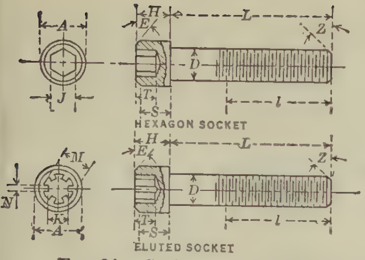


Fig. 21. Socket-head Cap Screw

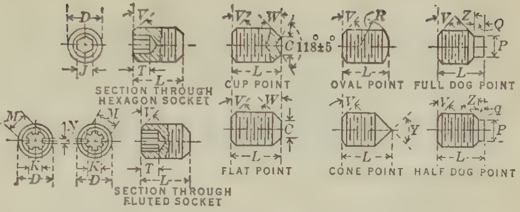


Fig. 22. Socket Set-screw

The length under the head $L$ is measured on a line parallel to the axis, from the plane of the bearing surface to the plane of the flat of the point. The increments of length are as follows:

| | | | |
|---|---|---|---|
| Screw length, in. | 1/4 to 1 | 1 to 4 | 4 to 6 |
| Increment, in. | 1/8 | 1/4 | 1/2 |

**SET-SCREWS** are made both with square heads or with sockets into which a wrench may be inserted for tightening.

**Square Head Set-screws.**—The width across the flats is equal to the diameter $D$ of screw, with a tolerance of minus $(0.020D + 0.006)$ from basic, except for 1/4- and 5/16-in. screws. Tolerances for these sizes are given in Table 38. The minimum width across rounded corners is 1.373 × minimum width across flats. The height of head is $3/4\,D$ with a tolerance of $\pm(0.020\,D + 0.004)$ from nominal. The maximum width of the neck under the head, for sizes of 3/8 in. and larger, is twice the pitch of the thread. The sizes under 3/8 in. are not necked, but are beveled not more than 40 deg. under the head. The radius of the crown of the head is 2 1/2 $D$. The dimensions are given in Table 38.

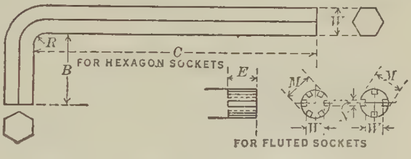


Fig. 23. Wrench for Socket Screws

**Table 41.—Dimensions of American Standard Socket Screw Wrenches**

A.S.A., B18.3—1936. All dimensions in inches. See Fig. 23

| Screw Size D | | Length Short Arm B | | Length Long Arm, C | | | | Hexagon | | | Fluted | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Short Series | | Long Series | | Width Across Flats W | | Radius of Bend R | Flutes | | | Major Diam. M | Minor Diam. W | Radius of Bend R |
| Set | Cap | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | | No. | Width Max. N | Length E | Max. | Max. | |
| No. 5 | ....... | 21/32 | 15/32 | 1 27/32 | 1 21/32 | ....... | ....... | 1/16 | 0.0615 | 1/16 | 4 | 0.0260 | 1/8 | 0.0690 | 0.0510 | 1/16 |
| " 6 | ....... | 21/32 | 15/32 | 1 27/32 | 1 21/32 | ....... | ....... | 1/16 | .0615 | 1/16 | 4 | .0280 | 1/8 | .0760 | .0530 | 1/16 |
| " 8 | ....... | 45/64 | 33/64 | 1 31/32 | 1 25/32 | ....... | ....... | 5/64 | .0771 | 5/64 | 6 | .0260 | 1/8 | .0940 | .0780 | 5/64 |
| " 10 | ....... | 3/4 | 9/16 | 2 3/32 | 1 29/32 | ....... | ....... | 3/32 | .0927 | 3/32 | 6 | .0290 | 5/32 | .1100 | .0940 | 3/32 |
| " 12 | ....... | 3/4 | 9/16 | 2 3/32 | 1 29/32 | ....... | ....... | 3/32 | .0927 | 3/32 | 6 | .0290 | 5/32 | .1100 | .0940 | 3/32 |
| 1/4 | No. 8 | 27/32 | 21/32 | 2 11/32 | 2 5/32 | 3 27/32 | 3 21/32 | 1/8 | .1235 | 1/8 | 6 | .0370 | 3/16 | .1440 | .1230 | 1/8 |
| 5/16 | 10, 12 | 15/16 | 3/4 | 2 19/32 | 2 13/32 | 4 7/32 | 4 1/32 | 5/32 | .1547 | 5/32 | 6 | .0440 | 7/32 | .1830 | .1580 | 5/32 |
| 3/8 | 1/4 | 1 1/32 | 27/32 | 2 27/32 | 2 21/32 | 4 19/32 | 4 13/32 | 3/16 | .1860 | 3/16 | 6 | .0550 | 1/4 | .2160 | .1720 | 3/16 |
| 7/16 | 5/16 | 1 1/8 | 15/16 | 3 3/32 | 2 29/32 | 4 31/32 | 4 25/32 | 7/32 | .2172 | 7/32 | 6 | .0655 | 9/32 | .2510 | .2010 | 7/32 |
| 1/2 | ....... | 1 7/32 | 1 1/32 | 3 11/32 | 3 5/32 | 5 11/32 | 5 5/32 | 1/4 | .2485 | 1/4 | 6 | .0775 | 5/16 | .2910 | .2390 | 1/4 |
| 9/16 | ....... | 1 7/32 | 1 1/32 | 3 11/32 | 3 5/32 | 5 11/32 | 5 5/32 | 1/4 | .2485 | 1/4 | 6 | .0875 | 11/32 | .3310 | .2730 | 9/32 |
| 5/8 | 3/8<br>7/16 | 1 11/32 | 1 5/32 | 3 27/32 | 3 21/32 | 6 3/32 | 5 29/32 | 5/16 | .3110 | 5/16 | 6 | .0975 | 3/8 | .3720 | .3070 | 5/16 |
| 3/4 | 1/2<br>9/16 | 1 15/32 | 1 9/32 | 4 11/32 | 4 5/32 | 6 27/32 | 6 21/32 | 3/8 | .3735 | 3/8 | 6 | .1185 | 7/16 | .4540 | .3770 | 3/8 |
| 7/8 | 5/8 | 1 23/32 | 1 17/32 | 5 11/32 | 5 5/32 | 8 11/32 | 8 5/32 | 1/2 | .4985 | 1/2 | 6 | .1460 | 1/2 | .5950 | .5000 | 1/2 |
| 1 | ....... | 1 27/32 | 1 21/32 | 5 27/32 | 5 21/32 | 9 3/32 | 8 29/32 | 9/16 | .5600 | 9/16 | 6 | .1590 | 9/16 | .6480 | .5620 | 1/2 |
| 1 1/8 | 3/4, 7/8 | 1 27/32 | 1 21/32 | 5 27/32 | 5 21/32 | 9 3/32 | 8 29/32 | 9/16 | .5600 | 9/16 | 6 | .1615 | 5/8 | .6480 | .5620 | 9/16 |
| 1 1/4<br>1 3/8 | 1 | 1 31/32 | 1 25/32 | 6 11/32 | 6 5/32 | 9 27/32 | 9 21/32 | 5/8 | .6225 | 5/8 | 6 | .1875 | 11/16 | .7840 | .6250 | 5/8 |
| 1 1/2 | 1 1/8<br>1 1/4<br>1 3/8 | 2 7/32 | 2 1/32 | 7 11/32 | 7 5/32 | 11 11/32 | 11 5/32 | 3/4 | .7475 | 3/4 | 6 | .2245 | 13/16 | .9550 | .7500 | 3/4 |
| 1 3/4 | ....... | 2 23/32 | 2 17/32 | 9 11/32 | 9 5/32 | 14 11/32 | 14 5/32 | 1 | .9975 | 1 | 6 | .3015 | 15/16 | 1.2680 | 1.0000 | 7/8 |
| 2 | 1 1/2 | 2 23/32 | 2 17/32 | 9 11/32 | 9 5/32 | 14 11/32 | 14 5/32 | 1 | .9975 | 1 | 6 | .3015 | 1 1/16 | 1.2680 | 1.0000 | 1 |

**Socket Set-screws.**—Socket set screws are made in several forms as shown in Fig. 22. The half-dog-point screw is used where the usable length of thread is less than the nominal diameter. Threads conform to the American National Standard thread. See p. 9–04. The sockets may be either hexagonal or fluted. Dimensions of both types are given in Table 40. The socket depth $T$ is made as great as practicable, but definite values are not specified. The socket end of the screw is flat and chamfered at an angle $V = 35°, +5°, -0°$. The chamfer $W$ on the point end of cup-point screws is $45°, +5°, -0°$. The chamfer $Z$ of dog-point screws is $35° + 5°, -0°$. Dimensions of socket set screws are given in Table 40. The length of socket screws increases by the following increments:

| | | | | |
|---|---|---|---|---|
| Screw length, in. | 1/4 to 5/8 | 5/8 to 1 | 1 to 4 | 4 to 6 |
| Increment, in. | 1/16 | 1/8 | 1/4 | 1/2 |

**SELF-TAPPING SCREWS** are screws that may be driven into an untapped hole, forming the thread in the hole as they are driven. One form, with a sharp, hardened, widely spaced V-thread, is used to join sheet-metal parts or fasten sheet metal to aluminum, bakelite or other relatively soft substances. Another form, shown in Fig. 24 is tapered for a short distance from the end. A short slot cut in the tapered section forms, in effect, a single-flute tap which cuts a thread in the material as the screw is screwed into it. These screws can be used with the softer materials and also with steel, cast iron, brass and bronze. Sizes of screws and sizes of holes for them are given in Table 42.

**LAG SCREWS AND COACH SCREWS.**—Lag screws and coach screws are made with a cone point or a gimlet point, and have a wood-screw thread. Lag screws have a square, and coach screws a hexagon head. See Fig. 25 and Table 43.

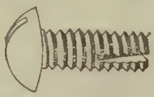


Fig. 24. Self-tapping Screw

Fig. 25. Lag Screw

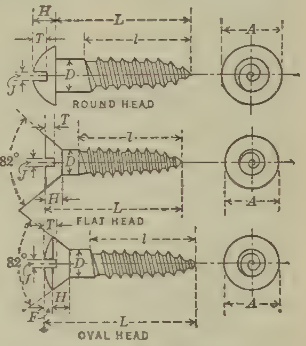


Fig. 26. Wood Screws

**WOOD SCREWS.**—Wood screws are made with round, flat, and oval heads as shown in Fig. 26. Dimensions of the various forms are given in Table 44. The various dimensions are determined by the following formulas: Head diameter, round head, $A_{max} = 1.887\,D$; $A_{min} = (1.813\,D - 0.010)$; flat head and oval head, $A_{max} = (2.04\,D - 0.003)$; $A_{min} = (1.96\,D - 0.013)$. Height of head, round head, $H_{max} = (0.636\,D + 0.015)$; $H_{min} = (0.624\,D + 0.005)$; flat and oval head, $H_{max} = (0.619\,D - 0.002)$; $H_{min} =$

### Table 42.—Hole Sizes for Self-tapping Screws

(Shakeproof Lock Washer Co., Chicago)

| Size of Screw | Threads per in. | Thickness of Material, or Depth Screw Enters, in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 1/32 to 1/16 | | 1/16 to 1/8 | | 1/8 to 3/8 | | 3/8 up | |
| | | Hole Diam., in. | Drill Size | Hole Diam., in. | Drill Size | Hole Diam., in. | Drill Size | Hole Diam., in. | Drill Size |
| No. 4 | 40 | 0.089 | 43 | 0.089 | 43 | ...... | ...... | ...... | ...... |
| No. 6 | 32 | .113 | 33 | .120 | 31 | 0.120 | 31 | ...... | ...... |
| No. 8 | 32 | .136 | 29 | .144 | 27 | .147 | 26 | 0.147 | 26 |
| No. 10 | 24 | .154 | 23 | .161 | 20 | .166 | 19 | .169 | 18 |
| No. 12 | 24 | .180 | 15 | .185 | 13 | .189 | 12 | .189 | 12 |
| 1/4 | 20 | .199 | 8 | .209 | 4 | .221 | 2 | .221 | 2 |
| 5/16 | 18 | .257 | F | .261 | G | .266 | H | .272 | 1 |
| 3/8 | 16 | .302 | N | .312 | 5/16 | .323 | P | .323 | P |

### Table 43.—Dimensions of Lag Screws and Coach Screws

| Diameter, in. | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 3/4 | 7/8 | 1 |
|---|---|---|---|---|---|---|---|---|---|---|
| Threads per in. | 10 | 9 1/2 | 7 | 7 | 6 | 5 | 5 | 4 1/2 | 4 1/2 | 3 |
| Thickness of head, in. | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 17/32 | 5/8 | 3/4 | 7/8 |
| Short Diam. of Head, in. | 3/8 | 15/32 | 9/16 | 21/32 | 3/4 | 27/32 | 15/16 | 1 1/8 | 15/16 | 1 1/2 |

Length of Thread—All Diameters

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Length of screw, in. | 1 1/2 | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 6 | 8 | 10 | 12 |
| Length of thread, in. | To head | 1 1/2 | 2 | 2 1/4 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 6 | 7 | 7 |

$(0.552D - 0.007)$. Width of slot, all types, $J_{\max} = (0.182D + 0.020)$; $J_{\min} = (0.164D + 0.010)$. Depth of slot, round head, $T_{\max} = (0.362D + 0.017)$; $T_{\min} = (0.268D + 0.013)$; flat head, $T_{\max} = (0.288D - 0.002)$; $T_{\min} = (0.192D - 0.002)$;

**Table 44.—Dimensions of American Standard Wood Screws**

A.S.A., B18c—1930. All dimensions in inches. See Fig. 26

| No. | Diam. $D$ | Head Diam. $A$ | | Height of Head $H$ | | Width of Slot $J$ | | Depth of Slot $T$ | | Height of Oval $F$ | | Range of Length, $L$ * | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | Steel | | Brass | |
| | | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. |
| | | | | | | | | **Round Head** | | | | | | | |
| 0 | 0.060 | 0.113 | 0.099 | 0.053 | 0.042 | 0.031 | 0.020 | 0.039 | 0.029 | ...... | ...... | 3/8 | 1/4 | 3/8 | 1/4 |
| 1 | .073 | .138 | .122 | .061 | .051 | .033 | .022 | .043 | .033 | ...... | ...... | 1/2 | 1/4 | 1/2 | 1/4 |
| 2 | .086 | .162 | .146 | .070 | .059 | .036 | .024 | .048 | .036 | ...... | ...... | 3/4 | 1/4 | 3/4 | 1/4 |
| 3 | .099 | .187 | .169 | .078 | .067 | .038 | .026 | .053 | .040 | ...... | ...... | 1 | 1/4 | 3/4 | 1/4 |
| 4 | .112 | .211 | .193 | .086 | .075 | .040 | .028 | .058 | .043 | ...... | ...... | 1 1/2 | 1/4 | 1 | 1/4 |
| 5 | .125 | .236 | .217 | .095 | .083 | .043 | .031 | .062 | .047 | ...... | ...... | 1 1/2 | 3/8 | 1 1/2 | 3/8 |
| 6 | .138 | .260 | .240 | .103 | .091 | .045 | .033 | .067 | .050 | ...... | ...... | 2 1/2 | 3/8 | 1 1/2 | 3/8 |
| 7 | .151 | .285 | .264 | .111 | .099 | .047 | .035 | .072 | .053 | ...... | ...... | 2 1/2 | 3/8 | 1 1/2 | 1/2 |
| 8 | .164 | .309 | .287 | .119 | .107 | .050 | .037 | .076 | .057 | ...... | ...... | 2 3/4 | 3/8 | 2 | 1/2 |
| 9 | .177 | .334 | .311 | .128 | .115 | .052 | .039 | .081 | .060 | ...... | ...... | 3 | 1/2 | 2 | 5/8 |
| 10 | .190 | .359 | .334 | .136 | .124 | .055 | .041 | .086 | .064 | ...... | ...... | 3 1/2 | 1/2 | 2 1/2 | 5/8 |
| 11 | .203 | .383 | .358 | .144 | .132 | .057 | .043 | .090 | .067 | ...... | ...... | 3 1/2 | 5/8 | 2 1/2 | 3/4 |
| 12 | .216 | .408 | .382 | .152 | .140 | .059 | .045 | .095 | .071 | ...... | ...... | 4 | 5/8 | 3 | 3/4 |
| 14 | .242 | .457 | .429 | .169 | .156 | .064 | .050 | .105 | .078 | ...... | ...... | 4 | 7/8 | 3 | 1 |
| 16 | .268 | .506 | .476 | .185 | .172 | .069 | .054 | .114 | .085 | ...... | ...... | 4 † | 1 | ...... | ...... |
| 18 | .294 | .555 | .523 | .202 | .188 | .074 | .058 | .123 | .092 | ...... | ...... | 3 †‡ | 1 1/4 | ...... | ...... |
| 20 | .320 | .604 | .570 | .219 | .205 | .078 | .062 | .133 | .099 | ...... | ...... | 2 1/2†‡ | 1 1/2 | ...... | ...... |
| 24 | .372 | .702 | .664 | .252 | .237 | .088 | .071 | .152 | .113 | ...... | ...... | ...... | ...... | ...... | ...... |
| | | | | | | | | **Flat Head** | | | | | | | |
| 0 | 0.060 | 0.119 | 0.105 | 0.035 | 0.026 | 0.031 | 0.020 | 0.015 | 0.010 | ...... | ...... | 3/8 | 1/4 | 3/8 | 1/4 |
| 1 | .073 | .146 | .130 | .043 | .033 | .033 | .022 | .019 | .012 | ...... | ...... | 1/2 | 1/4 | 1/2 | 1/4 |
| 2 | .086 | .172 | .156 | .051 | .040 | .036 | .024 | .023 | .015 | ...... | ...... | 3/4 | 1/4 | 3/4 | 1/4 |
| 3 | .099 | .199 | .181 | .059 | .048 | .038 | .026 | .027 | .017 | ...... | ...... | 1 | 1/4 | 3/4 | 1/4 |
| 4 | .112 | .225 | .207 | .067 | .055 | .040 | .028 | .030 | .020 | ...... | ...... | 1 1/2 | 1/4 | 1 | 1/4 |
| 5 | .125 | .252 | .232 | .075 | .062 | .043 | .031 | .034 | .022 | ...... | ...... | 1 1/2 | 3/8 | 1 | 3/8 |
| 6 | .138 | .279 | .257 | .083 | .069 | .045 | .033 | .038 | .024 | ...... | ...... | 2 1/2 | 3/8 | 1 1/2 | 3/8 |
| 7 | .151 | .305 | .283 | .091 | .076 | .047 | .035 | .041 | .027 | ...... | ...... | 2 1/2 | 3/8 | 1 1/2 | 1/2 |
| 8 | .164 | .332 | .308 | .100 | .084 | .050 | .037 | .045 | .029 | ...... | ...... | 2 3/4 | 3/8 | 2 | 1/2 |
| 9 | .177 | .358 | .334 | .108 | .091 | .052 | .039 | .049 | .032 | ...... | ...... | 3 | 1/2 | 2 | 5/8 |
| 10 | .190 | .385 | .359 | .116 | .098 | .055 | .041 | .053 | .034 | ...... | ...... | 3 1/2 | 1/2 | 2 1/2 | 5/8 |
| 11 | .203 | .411 | .385 | .124 | .105 | .057 | .043 | .056 | .037 | ...... | ...... | 3 1/2 | 5/8 | 2 1/2 | 3/4 |
| 12 | .216 | .438 | .410 | .132 | .112 | .059 | .045 | .060 | .039 | ...... | ...... | 4 | 5/8 | 3 1/2 | 3/4 |
| 14 | .242 | .491 | .461 | .148 | .127 | .064 | .050 | .068 | .044 | ...... | ...... | 5 | 7/8 | 3 1/2 | 1 |
| 16 | .268 | .544 | .512 | .164 | .141 | .069 | .054 | .075 | .049 | ...... | ...... | 5 | 1 | 3 1/2 | 2 |
| 18 | .294 | .597 | .563 | .180 | .155 | .074 | .058 | .083 | .054 | ...... | ...... | 5 | 1 1/4 | 3 1/2 | 2 |
| 20 | .320 | .650 | .614 | .196 | .170 | .078 | .062 | .090 | .059 | ...... | ...... | 5 | 1 1/2 | ...... | ...... |
| 24 | .372 | .756 | .716 | .228 | .198 | .088 | .071 | .105 | .069 | ...... | ...... | 5 | 3 | ...... | ...... |
| | | | | | | | | **Oval Head** | | | | | | | |
| 0 | 0.060 | 0.119 | 0.105 | 0.035 | 0.026 | 0.031 | 0.020 | 0.030 | 0.025 | 0.021 | 0.015 | ...... | ...... | 3/8 | 1/4 |
| 1 | .073 | .146 | .130 | .043 | .033 | .033 | .022 | .038 | .031 | .025 | .019 | ...... | ...... | 1/2 | 1/4 |
| 2 | .086 | .172 | .156 | .051 | .040 | .036 | .024 | .045 | .037 | .029 | .022 | ...... | ...... | 3/4 | 1/4 |
| 3 | .099 | .199 | .181 | .059 | .048 | .038 | .026 | .052 | .043 | .033 | .026 | ...... | 3/8 | 3/4 | 1/4 |
| 4 | .112 | .225 | .207 | .067 | .055 | .040 | .028 | .059 | .049 | .037 | .029 | 3/4 | 3/8 | 1 | 1/4 |
| 5 | .125 | .252 | .232 | .075 | .062 | .043 | .031 | .067 | .055 | .041 | .033 | 1 | 1/2 | 1 | 3/8 |
| 6 | .138 | .279 | .257 | .083 | .069 | .045 | .033 | .074 | .060 | .045 | .036 | 1 3/4 | 1/2 | 1 1/2 | 3/8 |
| 7 | .151 | .305 | .283 | .091 | .076 | .047 | .035 | .081 | .066 | .049 | .039 | 1 3/4 | 1/2 | 1 1/2 | 1/2 |
| 8 | .164 | .332 | .308 | .100 | .084 | .050 | .037 | .088 | .070 | .053 | .043 | 1 3/4 | 1/2 | 2 | 1/2 |
| 9 | .177 | .358 | .334 | .108 | .091 | .052 | .039 | .095 | .078 | .057 | .046 | 1 3/4 | 3/4 | 2 | 5/8 |
| 10 | .190 | .385 | .359 | .116 | .098 | .055 | .041 | .103 | .084 | .061 | .050 | 2 | 3/4 | 2 | 5/8 |
| 11 | .203 | .411 | .385 | .124 | .105 | .057 | .043 | .110 | .090 | .065 | .053 | 2 | 1 | ...... | ...... |
| 12 | .216 | .438 | .410 | .132 | .112 | .059 | .045 | .117 | .096 | .069 | .057 | 2 1/2 | 7/8 | 2 § | 3/4 |
| 14 | .242 | .491 | .461 | .148 | .127 | .064 | .050 | .132 | .108 | .077 | .064 | 2 1/2 | 1 | 2 | 1 |
| 16 | .268 | .544 | .512 | .164 | .141 | .069 | .054 | .146 | .120 | .084 | .071 | 2 | 1 1/2 | ...... | ...... |
| 18 | .294 | .597 | .563 | .180 | .155 | .074 | .058 | .160 | .132 | .092 | .078 | 2 | 1 1/2 | ...... | ...... |
| 20 | .320 | .650 | .614 | .196 | .170 | .078 | .062 | .175 | .144 | .100 | .085 | 2 | 1 1/2 | ...... | ...... |
| 24 | .372 | .756 | .716 | .228 | .198 | .088 | .071 | .204 | .168 | .116 | .099 | ...... | ...... | ...... | ...... |

* Length $L$ advances by increments of 1/8 in. from 1/4 in. to 1 in.; by increments of 1/4 in. from 1 in. to 3 in.; by increments of 1/2 in. from 3 in. to 5 in.

† 2 3/4-in. length omitted. ‡ 2 1/2-in. length omitted. § 7/8-in. length omitted.

### Table 45.—Weight of 100 Lag Screws

| Length under Head to Point | Diameter, in. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 |
| in. | lb. | lb. | lb. | lb. | lb. | lb. | lb. | lb. | lb. | lb. | lb. |
| 1 1/2 | 4.2 | 6.5 | 9.2 | 13.0 | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 1 3/4 | 4.7 | 7.1 | 10.0 | 13.8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 | 5.2 | 7.7 | 10.9 | 14.9 | 23.0 | 24.8 | ...... | ...... | ...... | ...... | ...... |
| 2 1/4 | 5.7 | 8.4 | 11.8 | 16.1 | 24.5 | 27.3 | ...... | ...... | ...... | ...... | ...... |
| 2 1/2 | 6.2 | 9.2 | 12.7 | 17.4 | 26.0 | 29.0 | 43.0 | ...... | ...... | ...... | ...... |
| 3 | 7.2 | 10.6 | 14.6 | 19.0 | 29.2 | 32.9 | 48.3 | 75.0 | ...... | ...... | ...... |
| 3 1/2 | 8.2 | 12.0 | 16.6 | 21.5 | 32.5 | 36.9 | 53.8 | 78.5 | 90 | ...... | ...... |
| 4 | 9.2 | 13.5 | 18.8 | 24.0 | 35.9 | 41.0 | 59.6 | 82.0 | 99 | ...... | ...... |
| 4 1/2 | 10.2 | 15.0 | 20.7 | 26.5 | 39.3 | 44.9 | 65.5 | 86.0 | 108 | ...... | ...... |
| 5 | 11.3 | 16.5 | 22.8 | 29.0 | 42.7 | 48.8 | 71.5 | 90.0 | 118 | 150 | ...... |
| 5 1/2 | 12.4 | 18.0 | 24.9 | 31.5 | 46.1 | 52.7 | 77.5 | 98.0 | 128 | 163 | ...... |
| 6 | 13.5 | 19.5 | 27.0 | 34.0 | 49.5 | 56.6 | 83.5 | 106.0 | 138 | 176 | 240 |
| 7 | ...... | ...... | 31.1 | 39.0 | 56.3 | 64.5 | 95.5 | 122.5 | 158 | 203 | 270 |
| 8 | ...... | ...... | 35.2 | 44.0 | 63.1 | 72.5 | 107.6 | 139.0 | 178 | 230 | 300 |
| 9 | ...... | ...... | ...... | 49.0 | 69.9 | 80.5 | 119.8 | 155.5 | 198 | 257 | 332 |
| 10 | ...... | ...... | ...... | 54.0 | 76.7 | 88.5 | 131.0 | 172.0 | 219 | 284 | 365 |
| 11 | ...... | ...... | ...... | ...... | 83.5 | 96.5 | 143.1 | 188.5 | 240 | 311 | 395 |
| 12 | ...... | ...... | ...... | ...... | 90.5 | 104.5 | 155.4 | 205.0 | 261 | 338 | 425 |
| 13 | ...... | ...... | ...... | ...... | ...... | 112.5 | 167.6 | 221.5 | 282 | 365 | 459 |
| 14 | ...... | ...... | ...... | ...... | ...... | 121.0 | 179.8 | 238.0 | 304 | 393 | 493 |
| 15 | ...... | ...... | ...... | ...... | ...... | 129.5 | 192.0 | 255.0 | 326 | 421 | 527 |
| 16 | ...... | ...... | ...... | ...... | ...... | 138.0 | 204.0 | 272.0 | 348 | 449 | 562 |
| Thds. per in. | 10 | 7 | 7 | 6 | 5 | 5 | 4 1/2 | 4 1/2 | 3 | 3 | 3 |

oval head, $T_{\max} = (0.556\,D - 0.003)$; $T_{\min} = (0.460\,D - 0.003)$. Height of oval, oval head screws, $F_{\max} = (0.304\,D + 0.003)$; $F_{\min} = (0.268\,D - 0.001)$. Countersink angle, flat and oval head screws, max., 82 deg., min., 80 deg. The shape of the head of round head screws is semi-elliptical.

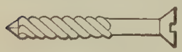


Fig. 27. Drive Screw

**DRIVE SCREWS.** (American Screw Co.)—Drive screws, Fig. 27, are driven in with a hammer. They have greater holding power than nails and can be withdrawn with an ordinary screwdriver. They are made with flat heads only, from 1/2 to 3 1/2 in. long, and from No. 4 to No. 20 diam.

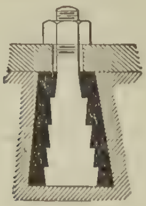


Fig. 28. Rag Bolt

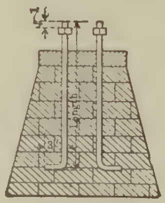


Fig. 29. Built-in Anchor Bolt

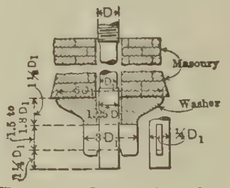


Fig. 30. Cottered Anchor Bolt

**FOUNDATION BOLTS** used for holding machine frames, engine beds, roof trusses, etc., to concrete or masonry may be divided into two classes, *viz.*, those embedded in the masonry and those passing through it. The rag bolt having a tapered shank with jagged edges is shown in Fig. 28. A hole wider at the bottom than at the top is cut in the masonry; the bolt is placed in position and molten lead is poured between the masonry and bolt. A built-in anchor bolt is shown in Fig. 29. A form of cottered bolt passing entirely through the masonry, with a washer and cotter at its lower end is shown in Fig. 30. The proportions are taken from Marshall (Machine Design and Drawing).

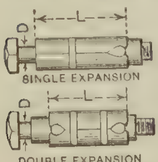


Fig. 31. Expansion Bolt

**EXPANSION BOLTS** are used for attaching wood or iron parts to masonry. With the Star expansion bolt, Fig. 31, a hole is drilled in the masonry of such size that the expansion nut or shield will fit it closely. When the bolt is screwed up the nut expands and binds firmly in the masonry. In most forms, however, the body of the bolt is divided and when the head of the bolt is turned, the sections forming the body of the bolt are

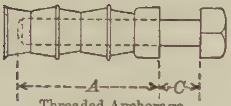
**Table 46.—Dimensions of Star Expansion Bolts.** (See Fig. 31)

| Diam. of Bolt, $D$, in. | Double Expansion | | | Single Expansion | | |
|---|---|---|---|---|---|---|
| | Drill, in. | Length $L$, in. | Length of Bolt, in. | Drill, in. | Length $L$, in. | Length of bolt, in. |
| 1/4 | 1/2 | 1 1/2 | 1 3/4; 2 to 6* | 1/2 | 1 | 1 1/2 to 5* |
| 5/16 | 9/16 | 1 7/8 | 2 to 6* | 9/16 | 1 1/2 | 1 1/2 to 5;* 5 to 6† |
| 3/8 | 11/16 | 2 3/8 | 2 1/2 to 8* | 11/16 | 1 3/4 | 2 to 5* |
| 7/16 | 7/8 | 2 1/2 | 3 to 8* | 13/16 | 1 3/4 | 2 to 5;* 5 to 6† |
| 1/2 | 7/8 | 2 5/8 | 3 to 8;* 8 to 12† | 7/8 | 1 3/4 | 2 to 5;* 5 to 12† |
| 5/8 | 1 | 3 1/4 | 3 1/2 to 8;* 8 to 12† | 1 | 2 | 2 to 5;* 5 to 12† |
| 3/4 | 1 1/4 | 4 | 4 to 8;* 8 to 12† | 1 3/16 | 3 | 3 1/2 to 5;* 5 to 12† |
| 7/8 | 1 1/2 | 4 3/4 | 5 to 8;* 8 to 12† | 1 3/8 | 3 3/4 | 4 to 5;* 5 to 12† |
| 1 | 1 5/8 | 5 | 5 1/2 to 8;* 8 to 12† | 1 7/16 | 3 3/4 | 4 to 5;* 5 to 12† |

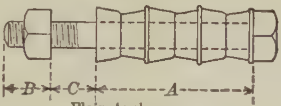
* Advancing by 1/2 in. † Advancing by 1 in.

**Table 47.—Dimensions of Cinch Anchors**

(National Lead Co., New York)



Threaded Anchorage
Length of Bolt = $A+C$



Plain Anchorage
Length of Bolt = $A+B+C$

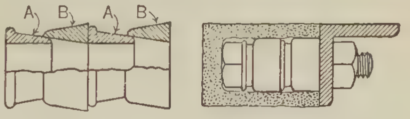
$A$ = length of anchorage; $B$ = thickness of nut; $C$ = thickness of work
All dimensions in inches

| Bolt size........ | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 | 7/8 | 1 |
|---|---|---|---|---|---|---|---|---|---|---|
| Length $A$ 2 unit | 7/8 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/2 | 1 5/8 | 2 | 2 1/4 | 2 5/8 |
| Length $A$ 3 unit | 1 3/8* | 1 1/2 | 1 1/2 | 2 1/8 | 2 1/8 | 2 3/8 | 2 5/8 | 3 1/4 | 3 5/8 | 4 1/4 |
| Length $B$....... | 1/4 | 3/8 | 3/8 | 1/2 | 1/2 | 5/8 | 3/4 | 1 | 1 1/4 | 1 1/2 |

* 1 1/4 in. for plain anchorage.

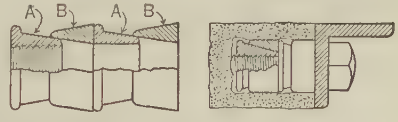
forced outward, binding in the masonry. The diameter of the expansion bolt is the size of the bolt itself and not the diameter of the shield or expansion member. The length of the bolt is found by adding together the thickness of material to be fastened and the length of the shield. In using expansion bolts, a hole of the size shown in Table 46 is drilled, into which the shield is inserted. The bolt is run through the object to be fastened into the shield and tightened.

The Cinch anchor made by the National Lead Co., New York, shown in Fig. 32, can be used wherever an expansion bolt can be applied. It consists of the conical malleable iron wedges $A$, and the lead composition sleeves $B$. The plain anchor is assembled on a bolt, which is inserted in the hole in the structure. It is then calked in by a pipe or sleeve placed over the bolt. When the bolt is drawn up by the nut the wedges are forced into the lead sleeves, expanding them into the hole, making an exceptionally strong anchorage. In the threaded anchorage, the innermost sleeve has a thread, forming a nut into which a standard bolt can be screwed. It is calked in place by means of a dummy bolt which is removed after calking is finished. With the plain anchorage the bolt cannot be removed, whereas it can be removed from the threaded anchorage. These anchorages are made with two units and three units. Dimensions are given in Table 47.



Two-Unit Plain Anchorage

Two-Unit Plain Anchor Installed



Two-Unit Threaded Anchorage

Two-Unit Threaded Anchor Installed

Fig. 32. Cinch Anchor

**THE HOOK BOLT** is used to connect pieces where it is impossible or impracticable to put a bolt hole through one of the pieces to be connected. Hook bolts are often used to attach shaft hangers to the flanges of joists and girders. Spooner (Machine Design, Construction and Drawing) gives the proportions shown in Fig. 33.

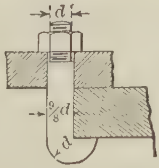


Fig. 33. Hook Bolt

**Table 48.—Dimensions of Standard T-bolts, T-slots and T-nuts.** (See Fig. 34)
A.S.A. B5a-1927. All dimensions in inches

| Diam. of Bolt, *D* | T-Bolt | | | T-Slot | | | | | T-Nuts | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Threads per inch | Width Across Flats, *H*, max.* | Height of Head, *T*, max.† | Width of Throat, *S* | Depth of Throat, *B* Max. | Depth of Throat, *B* Min. | Head Space Width, *W* Max.‡ | Head Space Depth, *C* Max.§ | Width and Length of Nut, *H* * | Height of Nut, *T* † | Width of Tongue, *J*¶ | Thickness of Tongue, *N* |
| 1/4 | 20 | 15/32 | 5/32 | 9/32 | 3/8 | 1/8 | 9/16 | 15/64 | 9/16 | 3/16 | 0.330 | 3/32 |
| 5/16 | 18 | 9/16 | 3/16 | 11/32 | 7/16 | 5/32 | 21/32 | 17/64 | 11/16 | 1/4 | .418 | 1/8 |
| 3/8 | 16 | 11/16 | 1/4 | 7/16 | 9/16 | 7/32 | 25/32 | 21/64 | 7/8 | 5/16 | .543 | 7/32 |
| 1/2 | 13 | 7/8 | 5/16 | 9/16 | 11/16 | 5/16 | 31/32 | 25/64 | 1 1/8 | 13/32 | .668 | 7/32 |
| 5/8 | 11 | 1 1/8 | 13/32 | 11/16 | 7/8 | 7/16 | 1 1/4 | 31/64 | 1 5/16 | 17/32 | .783 | 1/4 |
| 3/4 | 10 | 1 5/16 | 17/32 | 13/16 | 1 1/16 | 9/16 | 1 15/32 | 5/8 | 1 11/16 | 11/16 | 1.033 | 5/16 |
| 1 | 8 | 1 11/16 | 11/16 | 1 1/16 | 1 1/4 | 3/4 | 1 27/32 | 53/64 | 2 1/16 | 15/16 | 1.273 | 3/8 |
| 1 1/4 | 7 | 2 1/16 | 15/16 | 1 5/16 | 1 9/16 | 1 | 2 7/32 | 1 3/32 | 2 1/2 | 1 3/16 | 1.523 | 7/16 |
| 1 1/2 | 6 | 2 1/2 | 1 3/16 | 1 9/16 | 1 15/16 | 1 1/4 | 2 31/32 | 1 11/32 | ...... | ...... | ...... | ...... |

* A tolerance of −0.031 in. is allowed on all sizes.

† A tolerance of −0.016 in. is allowed on sizes 1/4 to 5/8 in. inclusive, and of −0.031 in. on sizes 3/4 to 1 1/2 in. inclusive.

‡ A tolerance of −0.063 in. is allowed on sizes 1/4 to 5/8 in. inclusive, and of −0.094 in. on sizes 3/4 to 1 1/2 in. inclusive.

§ A tolerance of −0.031 in. is allowed on sizes 1/4 to 3/4 in. inclusive, of −0.047 in. on 1 in. size, and of −0.063 in. on 1 1/4 and 1 1/2 in. size.

¶ A tolerance of −0.010 in. is allowed on sizes 1/4 to 5/8 in. inclusive, and of −0.015 in. on sizes 3/4 to 1 1/4 in.

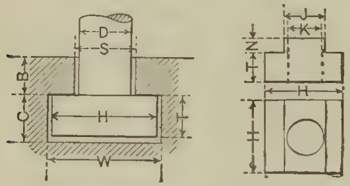


FIG. 34. T-Slot Bolts and Nuts

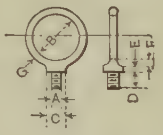


FIG. 35. Lifting Eye Bolt

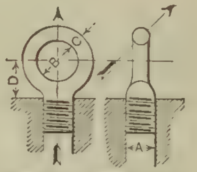


FIG. 36. Special Eye Bolt

**EYE-BOLTS.**—Lifting eye-bolts, Fig. 35, are used for the purpose of lifting machines or machine parts by means of ropes or chains attached to the eyes. The eye-bolt is screwed into a hole near the center of gravity of the piece. Table 49 from *Machinery's* Handbook gives proportions and strength of eye-bolts. Special eye-bolts with increased body, representing the General Electric Co. practice, are given in Fig. 36 and Table 50. These bolts are drop forgings and the arrows indicate the direction of pull to raise the weight. According to the General Electric Co. these sizes should be used in place of the following commercial eye-bolts sizes: 3/4, 1, 1 1/4, 1 1/2 and 2 in., respectively.

**Table 49.—Dimensions of Lifting Eye-bolts**

| Dimensions in Inches. (See Fig. 35) | | | | | | | Threads per in. | Strength at Bottom of Thread at 10,000 lb. per sq. in. Tensile Strength | Strength of Unstudded Chain Made from Bar of Diam. *G* |
|---|---|---|---|---|---|---|---|---|---|
| *A* | *B* | *C* | *D* | *E* | *F* | *G* | | | |
| 1/2 | 2 1/8 | 1 | 3/4 | 1/4 | 1/2 | 5/16 | 13 | 1,250 | 1,170 |
| 5/8 | 2 1/4 | 1 1/4 | 1 | 5/16 | 5/8 | 7/16 | 11 | 2,020 | 2,295 |
| 3/4 | 2 3/8 | 1 7/16 | 1 1/8 | 5/16 | 11/16 | 1/2 | 10 | 3,020 | 3,000 |
| 7/8 | 2 1/2 | 1 11/16 | 1 3/8 | 3/8 | 3/4 | 5/8 | 9 | 4,195 | 4,685 |
| 1 | 2 3/4 | 1 7/8 | 1 1/2 | 1/2 | 7/8 | 3/4 | 8 | 5,510 | 6,750 |
| 1 1/8 | 2 7/8 | 2 1/8 | 1 5/8 | 1/2 | 1 | 13/16 | 7 | 6,930 | 7,920 |
| 1 1/4 | 3 | 2 3/8 | 1 3/4 | 1/2 | 1 1/8 | 7/8 | 7 | 8,900 | 9,190 |
| 1 3/8 | 3 1/8 | 2 5/8 | 1 7/8 | 9/16 | 1 3/16 | 1 | 6 | 10,540 | 12,000 |
| 1 1/2 | 3 1/4 | 2 3/4 | 2 | 5/8 | 1 1/4 | 1 1/16 | 6 | 12,940 | 13,545 |
| 1 5/8 | 3 3/8 | 3 | 2 1/8 | 11/16 | 1 3/8 | 1 1/8 | 5 1/2 | 15,150 | 15,185 |
| 1 3/4 | 3 1/2 | 3 1/4 | 2 1/4 | 3/4 | 1 1/2 | 1 1/4 | 5 | 17,440 | 18,750 |
| 1 7/8 | 3 5/8 | 3 1/2 | 2 3/8 | 13/16 | 1 5/8 | 1 5/16 | 5 | 20,490 | 20,670 |
| 2 | 3 3/4 | 3 3/4 | 2 1/2 | 7/8 | 1 3/4 | 1 3/8 | 4 1/2 | 23,000 | 22,685 |

**Table 50.—Dimensions and Safe Load of Special Eye-bolts**

| Size, in. | Threads per in. | Dimensions in Inches. (See Fig. 36) | | | | Safe Weight, lb. | |
|---|---|---|---|---|---|---|---|
| | | *A* | *B* | *C* | *D* | Straight Lift | Lift at 45° |
| 1 1/4 | 7 | 1 1/4 | 1 1/2 | 5/8 | 1 3/8 | 1,500 | 600 |
| 1 1/2 | 6 | 1 1/2 | 1 7/8 | 13/16 | 1 3/4 | 2,500 | 800 |
| 2 | 4 1/2 | 2 | 2 1/8 | 1 | 2 1/8 | 4,000 | 1500 |
| 2 1/2 | 4 | 2 1/2 | 2 1/2 | 1 1/8 | 2 1/4 | 5,500 | 2600 |
| 3 | 3 1/2 | 3 | 3 1/4 | 1 1/2 | 3 1/4 | 9,000 | 3600 |
| 3 1/2 | 3 1/4 | 3 1/2 | 4 | 1 7/8 | 4 | 14,000 | 4800 |

**PATCH BOLTS.**—The function of the patch bolt, Fig. 37, is to fasten patches on the sheets of boilers. They should be used only where it is impossible to use rivets. Stock sizes have 12 threads, of the Am. National Standard, per inch. The length of patch bolts is measured from the point to the large diameter of the head. The stock sizes made by the Pittsburgh Screw & Bolt Co. are: 1/2 in. diameter by 3/4, 7/8 or 1 in. long, and 5/8, 3/4 and 1 in. diameter by 7/8, 1, 1 1/8, 1 1/4 or 1 1/2 in. long.

**STANDARD STUDS.**—The Upson Nut Co., Cleveland, gives the following formulas for the dimensions of standard stud bolts. Let $A$ = diameter of stud; $B$ = length of short thread; $C$ = length of unthreaded portion; $D$ = length of long thread; $E$ = total length of stud, all in inches. Then $B = A + 1/8$; $C = A$; $D = E - (B + C)$.

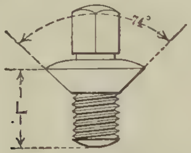


FIG. 37. Patch Bolt

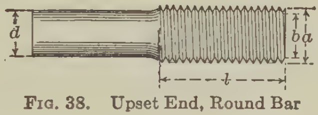


FIG. 38. Upset End, Round Bar

**Table 51.—Dimensions of Upset Ends for Round Bars**

See Fig. 38. American Standard Thread, Free Fit, Class 2.

(American Bridge Co. Standard)

| Bar | | | Upset | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | | | | | | Area | |
| Diameter, *d*, in. | Area, sq. in. | Weight per foot, lb. | Diameter, *a*, in. | Length, *l*, in. | Additional length for upset + 10%, in. | Diameter at root of thread, *b*, in. | At root of thread, sq. in. | Excess over area of bar, percent |
| 3/4 | 0.442 | 1.50 | 1 | 4 | 5 | 0.838 | 0.551 | 24.7 |
| 7/8 | .601 | 2.04 | 1 1/4 | 4 | 5 1/2 | 1.064 | 0.890 | 48.0 |
| 1 | .785 | 2.67 | 1 3/8 | 4 | 4 | 1.158 | 1.054 | 34.2 |
| 1 1/8 | .994 | 3.38 | 1 1/2 | 4 | 4 | 1.283 | 1.294 | 30.2 |
| 1 1/4 | 1.227 | 4.17 | 1 5/8 | 4 | 4 | 1.389 | 1.515 | 23.5 |
| 1 3/8 | 1.485 | 5.05 | 1 3/4 | 4 | 4 | 1.490 | 1.744 | 17.5 |
| 1 1/2 | 1.767 | 6.01 | 2 | 4 1/2 | 4 1/2 | 1.711 | 2.300 | 30.2 |
| 1 5/8 | 2.074 | 7.05 | 2 1/8 | 4 1/2 | 4 | 1.836 | 2.649 | 27.7 |
| 1 3/4 | 2.405 | 8.18 | 2 1/4 | 5 | 4 | 1.961 | 3.021 | 25.6 |
| 1 7/8 | 2.761 | 9.39 | 2 3/8 | 5 | 4 | 2.086 | 3.419 | 23.8 |
| 2 | 3.142 | 10.68 | 2 1/2 | 5 1/2 | 4 | 2.175 | 3.716 | 18.3 |
| 2 1/8 | 3.547 | 12.06 | 2 5/8 | 5 1/2 | 3 1/2 | 2.300 | 4.156 | 17.2 |
| 2 1/4 | 3.976 | 13.52 | 2 7/8 | 6 | 4 1/2 | 2.550 | 5.108 | 28.4 |
| 2 3/8 | 4.430 | 15.06 | 3 | 6 | 4 1/2 | 2.629 | 5.428 | 22.5 |
| 2 1/2 | 4.909 | 16.69 | 3 1/4 | 6 1/2 | 5 1/2 | 2.879 | 6.509 | 32.6 |
| 2 5/8 | 5.412 | 18.40 | 3 1/4 | 6 1/2 | 4 1/2 | 2.879 | 6.509 | 20.3 |
| 2 3/4 | 5.940 | 20.19 | 3 1/2 | 7 | 5 1/2 | 3.100 | 7.549 | 27.1 |
| 2 7/8 | 6.492 | 22.07 | 3 3/4 | 7 | 6 | 3.317 | 8.641 | 33.1 |
| 3 | 7.069 | 24.03 | 3 3/4 | 7 | 5 | 3.317 | 8.641 | 22.2 |
| 3 1/8 | 7.670 | 26.08 | 4 | 7 1/2 | 6 | 3.567 | 9.993 | 30.3 |
| 3 1/4 | 8.296 | 28.21 | 4 | 7 1/2 | 5 | 3.567 | 9.993 | 20.5 |
| 3 3/8 | 8.946 | 30.42 | 4 1/4 | 8 | 5 1/2 | 3.798 | 11.330 | 26.6 |
| 3 1/2 | 9.621 | 32.71 | 4 1/4 | 8 | 5 | 3.798 | 11.330 | 17.8 |
| 3 5/8 | 10.321 | 35.09 | 4 1/2 | 8 1/2 | 5 1/2 | 4.028 | 12.741 | 23.4 |
| 3 3/4 | 11.045 | 37.55 | 4 3/4 | 8 1/2 | 6 | 4.255 | 14.221 | 28.8 |
| 3 7/8 | 11.793 | 40.10 | 4 3/4 | 8 1/2 | 5 1/2 | 4.255 | 14.221 | 20.6 |

### Table 52.—Dimensions of Upset Ends for Square Bars

See Fig. 39. American Standard Thread, Free Fit, Class 2

(American Bridge Co. Standard)

| Bar | | | Upset | | | | | |
|---|---|---|---|---|---|---|---|---|
| Side of Square $d$, in. | Area, sq. in. | Weight per foot, lb. | Diameter $a$, in. | Length $l$, in. | Additional Length for Upset +10% in. | Diameter at Root of Thread $b$, in. | Area | |
| | | | | | | | At Root of Thread, sq. in. | Excess Over Area of Bar, % |
| 3/4 | 0.563 | 1.91 | 1 1/8 | 4 | 4 | 0.939 | 0.693 | 23.2 |
| 7/8 | 0.766 | 2.60 | 1 1/4 | 4 | 4 | 1.064 | 0.890 | 16.2 |
| 1 | 1.000 | 3.40 | 1 1/2 | 4 | 4 | 1.283 | 1.294 | 29.4 |
| 1 1/8 | 1.266 | 4.30 | 1 5/8 | 4 | 3 1/2 | 1.389 | 1.515 | 19.7 |
| 1 1/4 | 1.563 | 5.31 | 1 7/8 | 4 1/2 | 4 1/2 | 1.615 | 2.049 | 31.1 |
| 1 3/8 | 1.891 | 6.43 | 2 | 4 1/2 | 4 | 1.711 | 2.300 | 21.7 |
| 1 1/2 | 2.250 | 7.65 | 2 1/4 | 5 | 5 | 1.961 | 3.021 | 34.3 |
| 1 5/8 | 2.641 | 8.98 | 2 3/8 | 5 | 4 1/2 | 2.086 | 3.419 | 29.5 |
| 1 3/4 | 3.063 | 10.41 | 2 1/2 | 5 1/2 | 4 1/2 | 2.175 | 3.716 | 21.3 |
| 1 7/8 | 3.516 | 11.95 | 2 3/4 | 5 1/2 | 5 | 2.425 | 4.619 | 31.4 |
| 2 | 4.000 | 13.60 | 2 7/8 | 6 | 5 | 2.550 | 5.108 | 27.7 |
| 2 1/8 | 4.516 | 15.35 | 3 | 6 | 4 1/2 | 2.629 | 5.428 | 20.2 |
| 2 1/4 | 5.063 | 17.21 | 3 1/4 | 6 1/2 | 5 1/2 | 2.879 | 6.509 | 28.6 |
| 2 3/8 | 5.641 | 19.18 | 3 1/2 | 7 | 6 1/2 | 3.100 | 7.549 | 33.8 |
| 2 1/2 | 6.250 | 21.25 | 3 3/4 | 7 | 7 | 3.317 | 8.641 | 38.3 |
| 2 5/8 | 6.891 | 23.43 | 3 3/4 | 7 | 5 1/2 | 3.317 | 8.641 | 25.4 |
| 2 3/4 | 7.563 | 25.71 | 4 | 7 1/2 | 6 1/2 | 3.567 | 9.993 | 32.1 |
| 2 7/8 | 8.266 | 28.10 | 4 1/4 | 8 | 7 1/2 | 3.798 | 11.330 | 37.1 |
| 3 | 9.000 | 30.60 | 4 1/4 | 8 | 6 | 3.798 | 11.330 | 25.9 |
| 3 1/8 | 9.766 | 33.20 | 4 1/2 | 8 1/2 | 7 | 4.028 | 12.741 | 30.5 |
| 3 1/4 | 10.563 | 35.91 | 4 3/4 | 8 1/2 | 7 1/2 | 4.255 | 14.221 | 34.6 |

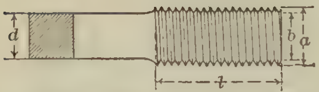


FIG. 39. Upset End, Square Bar

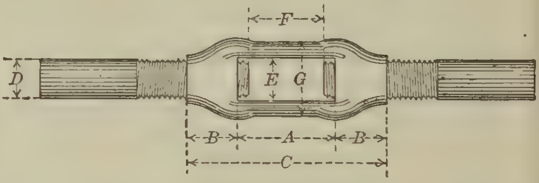


FIG. 40. Turnbuckle

**TURNBUCKLES** may be made of the following proportions: In Fig. 40, let $D$ = outside diameter of screw. Then length of tapped heads = 1.5 $D$; overall length $C = A + 3D$; $A$ = 6 in. up to $D$ = 4 in., and 9 in. for $D$ = 4 to 6 1/2 in. Dimensions of standard turnbuckles are given in Table 53.

### Table 53.—Dimensions of Turnbuckles (See Fig. 40)

Cleveland City Forge Co., Cleveland, Ohio.

| Diam. of Screw, $D$ | Length, $A$ | Clear, $F$ | Clear, $E$ | Nut, $B$ | Length, $C$ | Min. Width, $G$ | Weight, lb. | Diam. of Screw, $D$ | Length, $A$ | Clear, $F$ | Clear, $E$ | Nut, $B$ | Length, $C$ | Min. Width, $G$ | Weight, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3/8 | 6 | 5 | 9/16 | 9/16 | 7 1/8 | 1 1/32 | 0.41 | 2 1/2 | 6 | 5 | 3 | 3 3/4 | 13 1/2 | 5 | 23.25 |
| 1/2 | 6 | 5 | 11/16 | 3/4 | 7 1/2 | 1 5/16 | .70 | 2 3/4 | 6 | 5 | 3 1/4 | 4 1/8 | 14 1/4 | 5 5/8 | 31.50 |
| 5/8 | 6 | 5 | 13/16 | 29/32 | 7 13/16 | 1 1/2 | .89 | 3 | 6 | 5 | 3 5/8 | 4 1/2 | 15 | 6 1/8 | 39.50 |
| 3/4 | 6 | 5 | 15/16 | 1 1/16 | 8 1/8 | 1 23/32 | 1.20 | 3 1/4, 3 1/2 | 6 | 5 | 3 7/8 | 5 1/4 | 16 1/2 | 6 3/4 | 61.00 |
| 7/8 | 6 | 5 | 1 3/32 | 1 7/32 | 8 7/16 | 1 7/8 | 1.46 | 3 1/4, 3 1/2 | 9 | 8 | 3 7/8 | 5 1/4 | 19 1/2 | 6 3/4 | 70.00 |
| 1 | 6 | 5 | 1 9/32 | 1 3/8 | 8 3/4 | 2 1/32 | 2.27 | 3 3/4, 4 | 6 | 5 | 4 3/4 | 6 | 18 | 8 1/2 | 89.00 |
| 1 1/8 | 6 | 5 | 1 13/32 | 1 9/16 | 9 1/8 | 2 9/32 | 2.72 | 4 1/4, 4 1/2, 4 3/4 | 9 | 8 | 5 1/4 | 6 3/4 | 22 1/2 | 9 3/4 | 152.00 |
| 1 1/4 | 6 | 5 | 1 9/16 | 1 3/4 | 9 1/2 | 2 17/32 | 3.58 | 5 | 9 | 8 | 6 | 7 1/2 | 24 | 10 | 200.00 |
| 1 3/8 | 6 | 5 | 1 11/16 | 1 15/16 | 9 7/8 | 2 3/4 | 4.13 | 5 1/4 | 9 | 8 | 6 | 7 1/2 | 24 | 10 1/2 | 200.00 |
| 1 1/2 | 6 | 5 | 1 27/32 | 2 1/8 | 10 1/4 | 3 1/32 | 5.25 | 5 1/2 | 9 | 8 | 6 3/4 | 8 5/8 | 26 1/4 | 11 | 315.00 |
| 1 5/8 | 6 | 5 | 1 31/32 | 2 1/4 | 10 1/2 | 3 9/32 | 5.88 | 5 3/4 | 9 | 8 | 6 3/4 | 8 5/8 | 26 1/4 | 11 1/2 | 315.00 |
| 1 3/4 | 6 | 5 | 2 1/8 | 2 1/2 | 11 | 3 9/16 | 7.05 | 6 | 9 | 8 | 7 1/4 | 9 | 27 | 12 1/4 | 431.00 |
| 1 7/8, 2 | 6 | 5 | 2 3/8 | 2 3/4 | 11 1/2 | 4 | 9.95 | 6 1/4 | 9 | 8 | 7 1/4 | 9 | 27 | 12 3/4 | 431.00 |
| 2 1/4 | 6 | 5 | 2 11/16 | 3 3/8 | 12 3/4 | 4 5/8 | 18.00 | 6 1/2 | 9 | 8 | 7 1/2 | 9 3/4 | 28 1/2 | 13 1/4 | 520.00 |

## 4. STRENGTH OF SCREW FASTENINGS

**STRENGTH OF BOLTS.**—The load that a bolt in simple tension will sustain is

$$P = s\,A \qquad [7]$$

where $P$ = load, lb.; $A$ = minimum cross-sectional area of bolt, sq. in., *i.e.*, area at root of thread; $s$ = stress in bolt, lb. per sq. in. Faires (Design of Machine Elements) gives for the value of $s$ the formula $s = C\sqrt{A}$, from which by substitution in [7]

$$P = C\,A^{3/2} \qquad [8]$$

$C$ is a constant depending on the material used in the bolt. For ordinary steel screw stock of 56,000 lb. per sq. in. ultimate strength $C$ may be taken as 5000 for bolts up to 1 3/4 in. diameter. The value of $C$ for other materials is proportional to the ultimate strength. Thus if $S_a$, $S_s$ = respectively, ultimate strength of an alloy steel and ordinary screw stock, and $C_a$, $C_s$ = respectively, constants for the alloy and ordinary steels, $S_s/S_a = C_s/C_a$. For bolts over 1 3/4 in. diameter, of ordinary steel, the value of $s$ for equation [7] may be taken as 7000, $P$ in this case being the external load. For alloy steel bolts, values of $s$ up to 21,000 lb. may be used.

**Initial Tension,** or stress due to screwing up to the nut may vary over a wide range. Kimball and Barr state that it may be as high as 30,000 $D$, where $D$ is the nominal diameter of the bolt. As the result of experiments on different sizes of bolts tightened by various mechanics, they deduced that the initial load on a bolt, due to tightening, may be taken as $W = 16{,}000\,D$, where $W$ = load, lb. This is equivalent to a tensile stress in the bolt of $p = 27{,}500/D$ lb. per sq. in. The initial tension is indeterminate, as it will vary with the length of wrench used, the effort of the mechanic, and the character of the surfaces in frictional contact.

The action of screwing up also puts torsional stress on the bolt, which in the smaller sizes may exceed the tensile strength. The Kimball and Barr experiments indicated that a 1/2-in. bolt is too small where the nuts are to be screwed up tight. See Working Strength of Bolts, p. 9–44.

**Stress in Bolts Subject to Impact.**—Examples of bolts subject to impact or suddenly applied loads are long bolts acting as tension members in bridges, and bolts in connecting-rod ends. The bolt not only must have sufficient strength to sustain the static load, but have the resistance necessary to absorb the energy of the impacting load. Let $W$ = static value of load, lb.; $A$ = cross-sectional area of bolt, sq. in.; $h$ = height corresponding to velocity with which load is applied, in.; $s$ = unit stress in bolt, lb. per sq. in.; $\Delta$ = deformation produced in bolt by impact load $W\,h$, in. Then

$$s = (2\,W/A)\,\{(h/\Delta) + 1\} \qquad [9]$$

Faires (Design of Machine Elements) gives a modification of formula [9], which is more convenient for checking stress for a given set of conditions when the area remains constant. This formula is

$$s = (W/A) + (W/A)\sqrt{1 + (2\,h\,E\,A/l\,W)} \qquad [10]$$

where $E$ = modulus of elasticity of the bolt material, and $l$ = length of bolt, in.

The strength of a bolt is determined by its cross-sectional area at the root of the thread. Hence the unit stress is higher on this area than in the body of the bolt. The deformation in the threaded portion under shock loads will be greater than in the body of the bolt. If the unit stress in the body of the bolt is made equal to that in the threaded portion, the deformation will be uniform throughout the bolt, and its capacity to resist shock loads will be increased. Fig. 41 shows three methods that may be used. Tie rods for bridges and similar structures are made with upset ends on which the threads are cut, the diameter of the upset being such that the diameter at the root of the thread will be the same as that of the body of the rod.

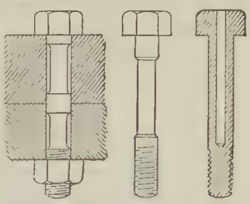


Fig. 41. Bolts for Shock Loads

An investigation to determine the impact and static tensile properties of bolts is reported in U. S. Bureau of Standards Research Paper No. 763, February, 1935. Impact and static tensile tests were made on bolts with nuts having American National Standard coarse threads, American National Standard fine threads, and Dardelet threads. Diameters were 3/8, 1/2, 5/8 and 3/4 in., and materials were heat treated chromium nickel steel, cold rolled Bessemer steel screw stock, Monel metal, a copper-silicon-manganese alloy, and brass. The following is a summary of the conclusions drawn from the tests:

**Impact Tests.**—Except for brass bolts and those which showed stripped threads or brittle failures, the work per unit volume required to rupture bolts was for each material and each type of thread approximately independent of the diameter of the bolt; the stretch in exposed threads was independent of the diameter. The impact work was greater with American National Standard fine threads than with American National Standard coarse threads, and approximately the same for Dardelet threads. Brass bolts with American National Standard fine threads averaged 12% less in impact work than bolts with Dardelet threads. In all cases, the impact work of Dardelet threads was greater than with American National Standard coarse threads.

**Static Tests.**—The static work with American National Standard fine threads was greater than with American National Standard coarse threads and approximately the same as with Dardelet threads. Brass bolts with American National Standard fine threads averaged 26% less static work than bolts with Dardelet threads. The static work in all cases with Dardelet threads was greater than with American National Standard coarse threads. In all cases, the maximum static load with Dardelet threads was approximately equal to the maximum static load with American National Standard fine threads and greater than with American National Standard coarse threads.

For bolts of the same size and having the same threads, the static bolt efficiency, *i.e.*, ratio of maximum static load to product of cross-sectional area of shank × ultimate tensile strength of material, was approximately the same for all classes of material.

**Stress Due to Combined External Load and Initial Tension.**—The maximum value of the combined stress on a bolt is the sum of the initial tension and external loads. This is obtained when the bolt is rigid and the parts which it connects are more or less yielding. The minimum value is either the initial or external load, whichever is greater, obtained with bolts that may yield to a certain extent. In practice, neither condition is attained, the actual stress lying some place between these two extremes. A form of construction in which the combined stress approaches the maximum is shown in Fig. 42, while Fig. 43 shows a type of connection in which the combined stress approaches the minimum value. The judgment of the designer must be exercised to determine the character of stress for which the bolt is to be designed. Kimball and Barr (Elements of Machine Design) quote from *The Locomotive* the following suggestions as a guide:

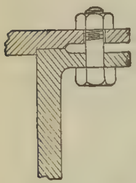


Fig. 42. Rigid Bolt and Yielding Member

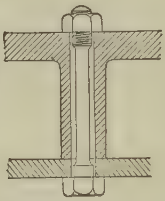


Fig. 43. Rigid Member and Yielding Bolt

When the conditions are such that the nut is apt to be screwed up hard, that is, when the initial load may be safely neglected, design for the external load alone.

When the initial load, due to screwing up, is apt to be considerable:

*a.* If a bolt is manifestly very much more yielding than the connected members, design the bolt simply for the initial load or for the external load, whichever is greater.

*b.* If the probable yield of the bolt is from one-half to once that of the connected members, consider the resultant load as the initial load plus from one-fourth to one-half the external load.

*c.* If the yield of the connected members is probably four or five times that of the bolts, take the resultant load as the initial load plus about three-fourths the external load.

*d.* In case of extreme relative yielding of the connected members, assume the resultant load to be nearly the sum of the initial and external loads.

**WORKING STRENGTH OF BOLTS.**—Forrest E. Cardullo has developed a formula, based on the Kimball and Barr experiments, for the working strength of bolts, which may be expressed as

$$S = s \times \text{working section} \qquad [11]$$

where $S$ = working strength, lb., and $s$ = permissible stress in the bolt, lb. per sq. in. The working section is determined by the formula

$$\text{Working section} = A - (a \times 2D) \qquad [12]$$

where $A$ = area of bolt at the root of the thread; $a$ = area of a $^1/_2$-in. bolt at the root of the thread; $D$ = nominal diameter of bolt. This formula takes account of the fact that the initial stress in a $^1/_2$-in. bolt, produced by screwing up of the nut, frequently equals or exceeds the tensile strength of the material. Table 54 is calculated from this formula on the basis of bolts threaded with American National Standard coarse threads.

**SAFE HOLDING CAPACITY OF SET SCREWS.**—(B. H. D. Pinkney, *Am. Mach.*, Oct. 15, 1914).—The diameter of a set-screw to secure a rotating body of known capacity may be found by formula [13] or [14]:

$$P = T/V \qquad [13]; \qquad P = 63{,}025\, H/nr \qquad [14]$$

where $T$ = torsional moment, in.-lb.; $r$ = radius of shaft, in.; $H$ = horsepower; $n$ = rev. per min., $P$ = resisting pressure, lb., at surface or shaft, which should be equal to, or less than, the safe holding power of a set-screw as follows:

| Size, in....... | $^1/_4$ | $^5/_{16}$ | $^3/_8$ | $^7/_{16}$ | $^1/_2$ | $^9/_{16}$ | $^5/_8$ | $^3/_4$ | $^7/_8$ | 1 | 1 $^1/_8$ | 1 $^1/_4$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Capacity, lb.. | 100 | 168 | 256 | 366 | 500 | 658 | 840 | 1280 | 1830 | 2500 | 3288 | 4198 |

The foregoing figures were based on experiments with $^1/_4$-in. and $^1/_2$-in. set-screws, in which it was found that the latter had a capacity of 5 times the former. Set-screws should be securely tightened with a wrench of sufficient leverage. Cup-point and flat-point set-screws are the best to use. The shaft should be provided with a " flat " of width equal to the diameter of the set-screw. The safe holding power of headless set-screws appears to be considerably less than those with heads.

**HOLDING POWER OF NAILS AND SCREWS.—Nails.**—The holding power of nails in wood varies with the specific gravity of the wood, which will vary in any given species. Hence, a considerable variation from published figures may exist in any nailed joint in a given species of wood. Table 55 gives the results of tests on the holding power of nails made by L. J. Markwardt, Senior Engineer, U. S. Forest Products Laboratory, and published in *Am. Lumberman*, July 13, 1929.

**Table 54.—Working Strength of Bolts**

| Diam. of Bolt, in. | Area at Root of Thread, sq. in. | Working Section, sq. in. | Permissible Stress in Bolt, lb. per sq. in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 5000 | 6000 | 7000 | 8000 | 10,000 | 12,000 | 15,000 | 20,000 |
| | | | Working Strength of Bolt, lb. | | | | | | | |
| 5/8 | 0.2018 | 0.0447 | 224 | 268 | 313 | 358 | 447 | 536 | 671 | 894 |
| 3/4 | .3020 | .1134 | 567 | 681 | 794 | 902 | 1,134 | 1,361 | 1,701 | 2,268 |
| 7/8 | .4193 | .1993 | 997 | 1,196 | 1,395 | 1,594 | 1,993 | 2,392 | 2,990 | 3,986 |
| 1 | .5510 | .2996 | 1,498 | 1,798 | 2,097 | 2,397 | 2,996 | 3,595 | 4,494 | 5,992 |
| 1 1/8 | .6931 | .4103 | 2,052 | 2,462 | 2,872 | 3,282 | 4,103 | 4,924 | 6,155 | 8,206 |
| 1 1/4 | .8898 | .5755 | 2,878 | 3,453 | 4,029 | 4,604 | 5,755 | 6,906 | 8,633 | 11,510 |
| 1 3/8 | 1.0541 | .7084 | 3,542 | 4,250 | 4,959 | 5,667 | 7,084 | 8,501 | 10,626 | 14,168 |
| 1 1/2 | 1.2938 | .9167 | 4,584 | 5,500 | 6,417 | 7,334 | 9,167 | 11,000 | 13,751 | 18,334 |
| 1 3/4 | 1.7441 | 1.3041 | 6,521 | 7,825 | 9,129 | 10,433 | 13,041 | 15,649 | 19,562 | 26,082 |
| 2 | 2.3001 | 1.7973 | 8,987 | 10,784 | 12,581 | 14,378 | 17,973 | 21,568 | 26,960 | 35,946 |
| 2 1/4 | 3.0212 | 2.4555 | 12,278 | 14,733 | 17,189 | 19,644 | 24,555 | 29,466 | 36,833 | 49,110 |
| 2 1/2 | 3.7160 | 3.0875 | 15,438 | 18,525 | 21,613 | 24,700 | 30,875 | 37,050 | 46,313 | 61,750 |
| 2 3/4 | 4.6194 | 4.0000 | 20,000 | 24,000 | 28,000 | 32,000 | 40,000 | 48,000 | 60,000 | 80,000 |
| 3 | 5.6209 | 4.8667 | 24,334 | 29,200 | 34,067 | 38,934 | 48,667 | 58,400 | 73,001 | 97,334 |
| 3 1/4 | 6.7205 | 5.9034 | 29,517 | 35,420 | 41,324 | 47,227 | 59,034 | 70,841 | 88,551 | 118,068 |
| 3 1/2 | 7.9183 | 7.0384 | 35,192 | 42,230 | 49,269 | 56,307 | 70,384 | 84,461 | 105,576 | 140,768 |
| 3 3/4 | 9.2143 | 8.2715 | 41,358 | 49,629 | 57,901 | 66,172 | 82,715 | 99,258 | 124,073 | 165,430 |
| 4 | 10.6084 | 9.6028 | 48,014 | 57,617 | 67,220 | 76,822 | 96,028 | 115,234 | 144,042 | 192,056 |

**Table 55.—Nail-holding Power of Various Species of Wood**

(7d. cement coated nails, driven 1 1/4 in. and pulled at once)

| Wood | Place of Growth | Specific Gravity, Oven-dried Wood | Average Holding Power of 1 Nail, lb., in | | |
|---|---|---|---|---|---|
| | | | End Surface | Radial Surface (edge-grain) | Tangential Surface (flat-grain) |
| Ash, white | Ark. | 0.64 | 385 | 455 | 452 |
| Basswood | Pa. | .41 | 138 | 199 | 194 |
| Birch, yellow | Wis. | .66 | 331 | 473 | 451 |
| Cedar, western red | Mont.<br>Wash. | .34 | 118 | 192 | 202 |
| Cedar, northern white | Wis. | .32 | 103 | 153 | 160 |
| Chestnut | Md.<br>Tenn. | .45 | 172 | 258 | 273 |
| Cottonwood, black | Wash. | .37 | 122 | 194 | 196 |
| Cypress, southern | La., Mo. | .47 | 144 | 266 | 291 |
| Douglas fir | Oreg.<br>Wash. | .51 | 183 | 273 | 296 |
| Elm, American | Pa. | .54 | 236 | 344 | 339 |
| Fir, California red | Calif. | .37 | 100 | 177 | 189 |
| Gum, tupelo | La., Mo. | .52 | 233 | 376 | 345 |
| Hemlock, eastern | Tenn.<br>Wis. | .42 | 127 | 225 | 230 |
| Hemlock, western | Wash. | .46 | 149 | 266 | 277 |
| Maple, black | Ind. | .62 | 357 | 480 | 415 |
| Maple, silver | Wis. | .51 | 280 | 333 | 338 |
| Maple, sugar | Ind. | 0.65 | 396 | 497 | 459 |
| Oak, red | Ark.<br>N. H.<br>Tenn. | .66 | 312 | 466 | 422 |
| Oak, white | Ark.<br>La. | .72 | 320 | 496 | 444 |
| Pine, longleaf | Fla.<br>La.<br>Miss. | .64 | 244 | 362 | 376 |
| Pine, Norway | Wis. | .51 | 165 | 273 | 282 |
| Pine, pitch | Tenn. | .54 | 235 | 325 | 330 |
| Pine, shortleaf | La. | .58 | 235 | 331 | 377 |
| Pine, northern white | Wis. | .39 | 136 | 220 | 225 |
| Pine, western white | Mont. | .45 | 134 | 255 | 246 |
| Pine, western yellow | Calif.<br>Oreg. | .44 | 122 | 224 | 233 |
| Poplar, yellow | Tenn. | .42 | 162 | 212 | 223 |
| Redwood | Calif. | .42 | 106 | 221 | 226 |
| Spruce, Engelmann | Colo. | .36 | 136 | 177 | 184 |
| Spruce, white | Wis. | .43 | 146 | 209 | 218 |

The resistance of nails to transverse loading, when used to fasten two planks together is, according to H. K. Hatt (*Proc.* Ind. Eng. Soc., 1900), $P = cd$, where $P$ = resistance, lb.; $d$ = size of nail, pennyweight; $c$ = a constant whose values are

| | Common Nails | | Finish Nails | | Fence Nails | |
|---|---|---|---|---|---|---|
| | Cut | Wire | Cut | Wire | Cut | Wire |
| Driven in oak | 82 | 70 | 49 | 52 | 100 | .... |
| " " yellow pine | 50 | 48 | 42 | 32 | 90 | 84 |

Joints will slip at about 60% of the loads given by the application of the formula.

**Resistance of Railroad Spikes.**—Values for the resistance of railroad spikes in various species of woods for both axial and lateral pull are given by H. K. Hatt in *Proc.* Am. Ry. Eng. Assoc., 1910, as follows:

| | Red Oak | Shortleaf Pine | Longleaf Pine | Loblolly Pine | Red Gum |
|---|---|---|---|---|---|
| Direct pull, lb. | 7639 | 4359 | 3955 | 3930 | 3883 |
| Lateral pull, lb. | 2026 | 1704 | 1650 | 1633 | 1619 |

The resistance of screw spikes was from 1.7 to 3.8 times greater than that of plain spikes in direct pull, and from 1.2 to 2.4 times greater in lateral pull.

**Resistance of Drift Bolts in Wood.**—Tests at the Watertown Arsenal in 1902 showed the resistance of drift bolts in wood driven parallel to the grain to be about one-half the resistance when driven across the grain. The holding power is in direct proportion to the area of contact and the compressive strength of the wood. There is no difference in the resistance of square drift bolts and round drift bolts of the same diameter. The tests gave the following values for ultimate resistance of drift bolts driven across the grain:

| | 1-in. bolt $^{15}/_{16}$-in. hole | 1-in. bolt $^{13}/_{16}$-in. hole | $^{3}/_{4}$-in. bolt $^{11}/_{16}$-in. hole | $^{3}/_{4}$-in. bolt $^{9}/_{16}$-in. hole | $^{1}/_{2}$-in. bolt $^{7}/_{16}$-in. hole | $^{1}/_{2}$-in. bolt $^{5}/_{16}$-in. hole |
|---|---|---|---|---|---|---|
| Douglas Fir | 2250 | 886 | 800 | 1000 | 374 | 1000 |
| Yellow Pine | 375 | 740 | ... | .... | ... | .... |
| White Oak | 3250 | 1300 | ... | .... | ... | .... |

**Wood Screws.**—Tests made by the U. S. Bureau of Standards to determine the holding power of wood screws in various species of wood (I. J. Fairchild, U. S. Bureau of Stds., *Tech. Bull.* No. 319, July 17, 1926) developed the formula

$$P = wl(120 + 20N) \qquad [15]$$

where $P$ = ultimate resistance to withdrawal, lb.; $l$ = length of screw embedded in wood; $N$ = commercial number representing diameter of screw = $(d - 0.06)\ 0.013$; $w$ = factor depending on the species of wood, values of which are given in Table 56. A wide variation from average values may occur in individual cases, ranging from −25% to +40%. The tests further showed that the diameter of the lead hole in soft wood should be about 70% of the diameter of the root of the thread, and in hard wood about 90%; holding power is not greatly diminished by the use of a lubricant, as soap; in general, long, slender screws are to be preferred, but may fail in tension before the limit of holding power is attained; holding power of screws in end grain is about 75% of holding power across the grain.

**Table 56.—Values of *w* in Fairchild Formula for Holding Power of Wood Screws**

| Wood | $w$ | Wood | $w$ | Wood | $w$ |
|---|---|---|---|---|---|
| Cypress | 1.00 | Maple (Hard) | 3.00 | Pine (White) | 1.00 |
| Douglas Fir | 1.45 | Oak (White) | 2.50 | Poplar (Yellow) | 1.33 |
| Hemlock (Western) | 1.35 | Pine (Georgia) | 2.25 | Spruce | 1.15 |
| Hickory | 4.50 | " (No. Carolina) | 1.90 | Sycamore | 1.75 |

Experiments made in Germany on the holding power of wood screws led to a formula that gives values fairly close to those given by formula [15]. This formula, transformed to U. S. units (D. A. Hampson, *Mech. Engg.*, April, 1929) is

$$H = R(MD + b) \qquad [16]$$

where $H$ = holding power, lb. per inch of screw embedment; $D$ = diameter of screw, in.; $R$, $M$, and $b$ = factors whose values are as follows:

| | Oak | Fir | Larch | Teak | Elm | Ash | Beech |
|---|---|---|---|---|---|---|---|
| $R$ = | 342 | 218 | 316 | 302 | 375 | 297 | 370 |
| $M$ = | 6.6 | 5.7 | 3.6 | 6.7 | 5.5 | 7.4 | 9.1 |
| $b$ = | 1.35 | 1.36 | 1.68 | 1.48 | 1.49 | 1.58 | 1.88 |

**Holding Power of Lag-screws.**—Tests of the holding power of lag-screws in various wood were made by A. J. Cox, at the University of Iowa, 1891, with results as follows:

| Kind of Wood | Size Screw, in. | Size Hole Bored, in. | Length in Tie, in. | Max. Resistance, lb. | No. Tests |
|---|---|---|---|---|---|
| Seasoned white oak.......... | $^5/_8$ | $^1/_2$ | $4\ ^1/_2$ | 8037 | 3 |
| " " " .......... | $^9/_{16}$ | $^7/_{16}$ | 3 | 6480 | 1 |
| " " " .......... | $^1/_2$ | $^3/_8$ | $4\ ^1/_2$ | 8780 | 2 |
| Yellow-pine stick............ | $^5/_8$ | $^1/_2$ | 4 | 3800 | 2 |
| White cedar, unseasoned...... | $^5/_8$ | $^1/_2$ | 4 | 3405 | 2 |

# RIVETS AND RIVETED FASTENINGS

**GENERAL CONSIDERATIONS.**—In making a riveted joint, the head is formed with a set or forming tool, by hand or by hydraulic or pneumatic machines. Machine riveting usually is better than hand work, as the pressure of the machine causes the rivet shank to fill completely the rivet hole. A rivet driven hot shrinks in cooling, pinching the riveted parts and making a firm joint. The cooling also leaves a tensile stress in the rivet, the amount of which cannot be measured. The pinching due to cooling of the rivet causes a frictional resistance between the riveted parts which Stoney found to be about 10,000 lb. per sq. in. of rivet area. American and English practice takes no account of this frictional resistance, but bases design on the tensile and shearing strength of the plates and rivets. Some French and German authorities base design entirely upon the frictional resistance.

## 1. RIVETS

**RIVET HEADS.**—Rivet heads are of many different forms, the particular form being determined by the use of the joint or fastening. Heads of greatest stiffness and strength are used when the rivet is to resist a tensile stress. Countersunk rivets weaken the plate and should be used only where projecting heads are objectionable. The proportions developed by the Hartford Steam Boiler Inspection and Insurance Co., and which have been adopted as standard in the A.S.M.E. Boiler Code, are shown in Fig. 1.

**RIVET HOLES.**—The rivet holes in a plate either may be punched or drilled. The latter is preferred as the action of the punch causes an injurious lateral flow of metal. It is current practice to make holes $^1/_{16}$ in. larger than the cold rivet diameter, except that for rivets not over $^7/_8$ in. diameter a hole of $^1/_{32}$ in. greater diameter than the rivet is preferred. Where work is to be reamed, it is usual to punch the holes $^1/_8$ or $^3/_{16}$ in. less than the finished diameter, and to drill or ream to the proper size after the parts are assembled.

It is good practice to remove the sharp corner at both ends of the reamed holes, so that a fillet will form where the body joins the head in finished rivets. Also, annealing will equalize the internal stresses caused by punching, and thus restore the strength of the plate.

The A.S.M.E. Boiler Code Committee recommends for boiler work that:

Rivet holes, except for attaching stays or angle bars to heads, shall be drilled full size with

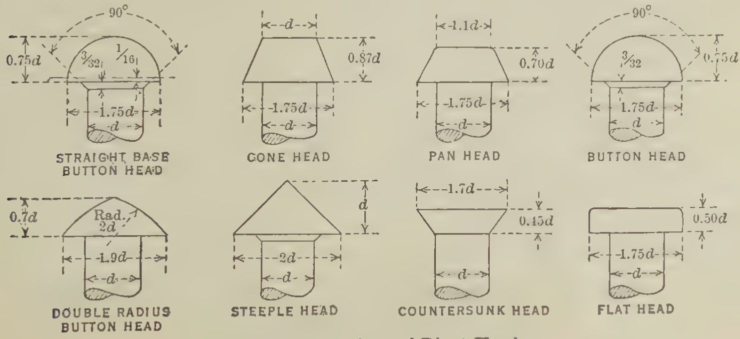


Fig. 1. Proportions of Rivet Heads

plates, butt-straps and heads bolted in position; or they may be punched not to exceed 1/4 in. less than full diameter for plates over 5/16 in. thick, and 1/8 in. less than full diameter for plates not exceeding 5/16 in. thick, and then drilled or reamed to full diameter with plates, butt-straps and heads bolted in position. After drilling rivet holes, the plates and butt-straps shall be separated and the burrs removed.

**ARRANGEMENT OF RIVETS.**—When several rows of rivets are used in lap-joints, the rivets may be arranged as: 1. Chain riveting; 2. Staggered riveting. See Fig. 2. See Figs. 7 to 13 for examples of the use of the two types in boiler work. For allowable shearing and bearing values of rivets in structural work, see Table 25, p. 17–79. For length of rivets for various thicknesses of joint see Table 26, p. 17–80.

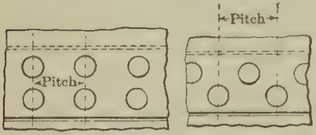


Fig. 2. Chain and Staggered Riveting

**RIVET SPACING.**—Rivet spacing has been the subject of much controversy and experiment. The rivets must be spaced far enough apart to allow proper clearance for use of the set during driving; 3/16 in. is the minimum clearance. However, rivets must not be spaced at such a distance that the fastened parts open between rivets, allowing leakage in a fluid pressure joint, or the accumulation of dirt or water to aggravate rusting in a structural joint. With the usual proportions of riveted joints, a distance from the center of the rivet to the edge of the plate equal to (rivet diameter × 1 1/2), is sufficient to prevent tearing or shearing. The proper rivet spacing for joints in boilers and pressure vessels is given on pages 6–24 to 6–28 of Kent's Mechanical Engineers' Handbook—Power, forming Vol. 2 of this series. See pages 17–14 to 17–19 of this book for standards of the American Institute of Steel Construction for structural work. See also Pitch of Rivets, p. 9–49.

**RIVET MATERIALS.**—It is well known that the strength of rivets is different in single and double shear. Kimball and Barr (Machine Design) recommend the following average values:

| | Shearing Strength, lb. per sq. in. | Crushing Strength, lb. per sq. in. |
|---|---|---|
| Iron rivets, single shear | 40,000 | 60,000 |
| Iron rivets, double shear | 39,000 | 72,000 |
| Steel rivets, single shear | 49,000 | 80,000 |
| Steel rivets, double shear | 42,000 | 100,000 |

The A.S.M.E. Boiler Code prescribes values as follows in computing the ultimate strength of rivets in single shear: Iron rivets, 38,000 lb. per sq. in.; steel rivets, 44,000 lb. per sq. in. For double shear, twice these values are used.

Specifications for steel for rivets are given in A.S.T.M. Specifications, Designation A131–36. See Table 6, p. 2–22. The tests to which the material shall be subjected are given in the notes appended to that table.

**PRESSURE REQUIRED TO DRIVE RIVETS** may be obtained from Fig. 3, which is based on formulas by Wilfred Lewis (*Am. Mach.*, July 13, 1911).

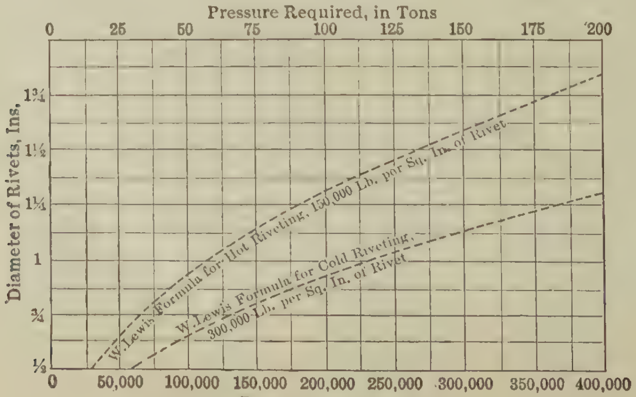


Fig. 3. Pressure Required to Drive Rivets

## 2. RIVETED JOINTS

**FORM OF RIVETED JOINTS FOR MORE THAN TWO PLATES.**—In tanks, boilers, and all pressure vessels requiring tight seams, the problem often is to join three or four plates. Fig. 4 shows method of making such joints.

Fig. 4–A is a junction of three plates, *m*, *n*, and *s*, where both seams are single-riveted lap-joints. The corner of *m* is drawn to an edge and tucked under *s*.

Fig. 4–B is a junction of three plates; one seam is single-riveted and the other is a double-riveted lap-joint. As before, the corner of *m* is drawn down and tucked under *s*.

Fig. 4–C is a junction of three plates with butt-straps; the longitudinal butt-strap is planed down and tucked under the cross butt-strap.

Fig. 4–D is the junction of three plates; the cross seam is a single-riveted lap-joint, and the longitudinal joint is a double-riveted butt-joint with double cover-plates. The upper cover-plate is planed on the end so that it can be tightly calked where it abuts against plate *n*.

Fig. 4–E is a method of joining four plates; both seams are single-riveted lap-joints, *n* and *s*, both being drawn down as shown.

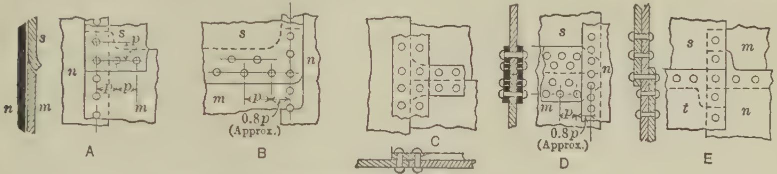


Fig. 4. Riveted Joints for More Than Two Plates

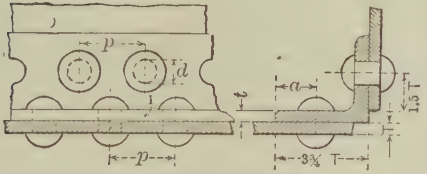


Fig. 5.

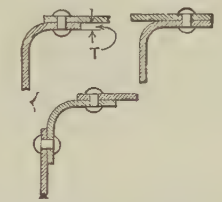


Fig. 6.

Methods of Joining Plates Not in the Same Plane

**Junction of Plates Not in the Same Plane** usually is effected by means of a rolled structural form. Fig. 5 shows an angle used with plates at right angles to each other. If $T$ = thickness of plate and $t$ = mean thickness of angle leg, then $t = T + {}^1/_{16}$ in. The width $w$ of each leg of the angle $= 3.75T$; the diameter $d$ of the rivet $= 2T$ and $a = {}^1/_2\,(w - t)$, where $w$ = width of angle.

Fig. 6 shows methods of connecting plates by flanging the plates themselves instead of using angles. Care should be taken not to use too sharp a radius of curvature and the flange should be annealed after it is bent. The inside radius of the flange must be greater than the thickness of the plate even with the mildest steel. Unwin recommends an inside radius of at least $4T$.

**SIZE AND PITCH OF RIVETS.**—Size and pitch of rivets for different forms of construction, as boiler work and structural work, are determined by the requirements of codes covering such construction. See Kent's Mechanical Engineers' Handbook—Power, pp. 6–25 to 6–28, forming Vol. 2 of this series, for extracts from the A.S.M.E. Boiler Code, and pages 17–14 to 17–19 of this book for riveting for structural work. For other types of construction, the following formulas given by Unwin (Elements of Machine Design) may be used.

Let $d$ = diameter of rivet, in.; $t$ = thickness of plate, in.; $f_b$ = resistance of plate or rivet to crushing, lb. per sq. in. of projected area of rivet; $f_s$ = shearing resistance of rivet, lb. per sq. in.; $f_t$ = tensile strength of plate, lb. per sq. in.; $p$ = pitch of rivets, in.

**Size of Rivet.**—For a rivet in single shear, $d \leqq 1.27\,t(f_b/f_s)$.

For a rivet in double shear, $d \leqq 0.635\,t(f_b/f_s)$.

**Pitch of Rivets.**—Single-riveted joint in single shear, $p = 0.785(d^2/t)(f_s/f_t) + d$.
Single-riveted joint in double shear, $p = 1.57(d^2/t)(f_s/f_t) + d$.
Double-riveted joint in single shear, $p = 0.785(d^2/t)(f_s/f_t) + d$.
Double-riveted joint in double shear, $p = 3.14(d^2/t)(f_s/f_t) + d$.
Triple-riveted joint in single shear, $p = 2.356(d^2/t)(f_s/f_t) + d$.
Triple-riveted joint in double shear, $p = 4.712(d^2/t)(f_s/f_t) + d$.

A joint may fail along a diagonal line between rivet holes unless the net diagonal section is from 30 to 35% greater than the net section along the transverse line between rivet holes. This excess of net section will be obtained with a diagonal pitch of $p_d = 0.667\,p + (d/3)$.

## Strength and Efficiency of Riveted Joints

A riveted joint may fail by shearing the rivets, tearing the plate between the rivets, crushing the rivets or plate, or by a combination of two or more of the above causes. To determine the efficiency or relative strength of a riveted joint, calculate the breaking strength for the different ways in which it may fail. That method of failure which gives the least value determines the actual strength of the joint. This value divided by the strength of the solid plate gives the efficiency of the joint.

The strength and efficiency of various types of riveted joints may be calculated by the following formulas. Tables 1 to 6 show the dimensions and efficiencies of standard boiler joints, as published in the Boiler Book of the Hartford Steam Boiler Inspection and Insurance Co. The dimensions $D$, $E$, and $F$ in these tables may be increased for convenience in riveting, without affecting the efficiency of the joint. They should not be decreased.

**Notation.**—$a$ = diameter of rivet hole, usually $^1/_{16}$ in. greater than rivet, $t$ = thickness of plate, $t_c$ = thickness of cover-plates, $p$ = pitch of inner row of rivets, $P$ = pitch of outer row of rivets, all in inches; and let $f_s$ = shearing strength of rivets; $f_t$ = tensile strength of plate, $f_c$ = crushing strength of rivet or plate, all in lb. per sq. in. Then the resistance for the various joints are as follows:

**Single-riveted Lap-joint**, Fig. 7.—Resistance to shearing one rivet = $(^1/_4\ \pi\ d^2)\ f_s$.
Resistance to tearing plate between rivets = $(p - d)\ t\,f_t$.
Resistance to crushing of rivet or plate = $d\,t\,f_c$.
**Double-riveted Lap-joint**, Fig. 8.—Resistance to shearing two rivets = $2\ (^1/_4\ \pi\ d^2)\ f_s$.
Resistance to tearing between two rivets = $(p - d)\ t\,f_t$.
Resistance to crushing in front of two rivets = $2\,d\,t\,f_c$.
Strength of solid plate = $p\,t\,f_t$.

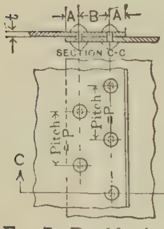


FIG. 7. Double-riveted Lap-joint

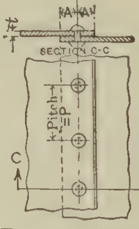


FIG. 8. Single-riveted Lap-joint

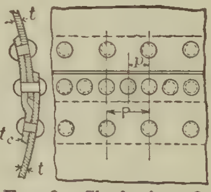


FIG. 9. Single-riveted Lap-joint with Inside Cover Plate

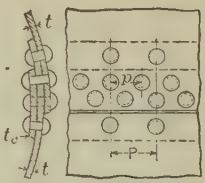


FIG. 10. Double-riveted Lap-joint with Inside Cover Plate

**Table 1.—Dimensions and Efficiency of Double- and Single-riveted Lap-joints for Longitudinal Seams.** (See Figs. 7 and 8)

| Thickness of Plate, in. | Diam. of Rivet Holes, in. | Single-riveted Joints |  |  |  | Double-riveted Joints† |  |  |  |
|---|---|---|---|---|---|---|---|---|---|
|  |  | Efficiency, % | Pitch, in. | $A$, in. | Method of Failure* | Efficiency, % | Pitch, in. | $A$, in. | $B$, in. |
| 1/4 | 11/16 | 60.7 | 1 3/4 | 1 1/16 | T.P. | 69.5 | 2 1/4 | 1 1/16 | 1 3/4 |
| 5/16 | 13/16 | 59.4 | 2 | 1 1/4 | T.P. | 69.1 | 2 5/8 | 1 1/4 | 1 7/8 |
| 3/8 | 15/16 | 58.3 | 2 1/4 | 1 7/16 | T.P. | 68.9 | 3 | 1 7/16 | 2 |
| 7/16 | 1 1/16 | 57.5 | 2 1/2 | 1 5/8 | T.P. | 68.5 | 3 3/8 | 1 5/8 | 2 1/8 |
| 1/2 | 1 1/16 | 56.7 | 2 1/2 | 1 5/8 | S.R. | 68.5 | 3 3/8 | 1 5/8 | 2 1/8 |

* T.P. = failure by tearing plate in net section between rivet holes; S.R. = failure by shearing of rivets.

† All joints fail by tearing net section of plate between rivet holes.

**Single-riveted Lap-joint, Inside Cover-plate,** Fig. 9.—Resistance to tearing between outer row of rivets = $(p - d)\,t f_t$.

Resistance to tearing between inner row of rivets, and shearing outer row of rivets $= (P - 2d)\,t f_t + (^1/_4\,\pi d^2)\,f_s$.

Resistance to shearing three rivets $= 3\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to crushing in front of three rivets $= 3\,t d f_c$.

Resistance to tearing at inner row of rivets and crushing in front of one rivet in outer row $= (P - 2d)\,t f_t + t d f_c$.

Strength of solid plate $= p t f_t$.

**Double-riveted Lap-joint with Inside Cover-plate,** Fig. 10.—Resistance to tearing at outer row of rivets $= (P - d)\,t f_t$.

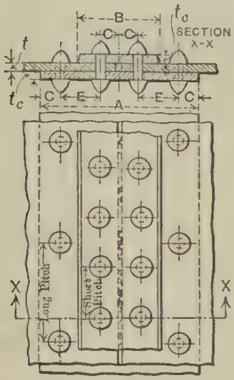


FIG. 11. Double-riveted Butt-joint

Resistance to shearing four rivets $= 4\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to tearing at inner row and shearing outer row of rivets $= (P - 1\,^1/_2\,d)\,t f_t + (^1/_4\,\pi d^2) f_s$.

Resistance to crushing in front of four rivets $= 4\,t d f_c$.

Resistance to tearing at inner row of rivets, and crushing in front of one rivet $= (P - 1\,^1/_2\,d)\,t f_t + t d f_c$.

Strength of solid plate $= P t f_t$.

**Double-riveted Butt-joint,** Fig. 11.—Resistance to tearing at outer row of rivets $= (P - d)\,t f_t$.

Resistance to shearing two rivets in double shear and one in single shear $= 5\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to tearing at inner row of rivets and shearing one of the outer row of rivets $= (P - 2d)\,t f_t + (^1/_4\,\pi d^2)\,f_s$.

Resistance to crushing in front of three rivets $= 3\,t d f_c$.

Crushing in front of two rivets and shearing one rivet $= 2\,t d f_c + (^1/_4\,\pi d^2)\,f_s$.

Strength of solid plate $= P t f_t$.

**Triple-riveted Butt-joint,** Fig. 12.—Resistance to tearing at outer row of rivets $= (P - d)\,t f_t$.

Resistance to shearing four rivets in double shear and one in single shear $= 9\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to tearing at middle row of rivets and shearing one rivet $= (P - 2d)\,t f_t + (^1/_4\,\pi d^2)\,f_s$.

Resistance to crushing in front of four rivets and shearing one rivet $= 4\,d t f_c + (^1/_4\,\pi d^2)\,f_s$.

Resistance to crushing in front of five rivets $= 5\,d t f_c$.

Strength of solid plate $= P t f_t$.

**Quadruple-riveted Butt-joint with Two Cover-plates,** Fig. 13.—Resistance to tearing at outer row of rivets $= (P - d)\,t f_t$.

Resistance to tearing at the third row and shearing one rivet in the outer row $= (P - 2d)\,t f_t + (^1/_4\,\pi d^2)\,f_s$.

Resistance to tearing at the second row of rivets and shearing three rivets $= (P - 4d)\,t f_t + 3\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to double shearing eight rivets and single shearing three rivets $= 19\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to crushing in front of eight rivets and single shearing three rivets $= 8\,d t f_c + 3\,(^1/_4\,\pi d^2)\,f_s$.

Resistance to crushing in front of eleven rivets $= 11\,d t f_c$.

Resistance to tearing at the third row of rivets and crushing in front of one rivet in the outer row $= (P - 2d)\,t f_t + d t f_c$.

Resistance to tearing at the second row of rivets and crushing in front of three rivets $= (P - 4d)\,t f_t + 3\,d t f_c$.

Strength of the solid plate $= P t f_t$.

**Table 2.—Dimensions and Efficiency of Double-riveted Butt-joints.** (See Fig. 11)

All joints fail by tearing the plate between rivet holes in the outer row.

| Thickness of Plate, in. | Thickness of Straps, in. | Diam. of Holes, in. | Efficiency, percent | Long Pitch, in. | Short Pitch, in. | *A*, in. | *B*, in. | *C*, in. | *D*, in. |
|---|---|---|---|---|---|---|---|---|---|
| 1/4 | 1/4 | 11/16 | 82.8 | 4 | 2 | 8 1/2 | 4 1/4 | 1 1/16 | 2 1/8 |
| 5/16 | 9/32 | 13/16 | 81.9 | 4 1/2 | 2 1/4 | 9 7/8 | 5 | 1 1/4 | 2 7/16 |
| 3/8 | 5/16 | 13/16 | 81.9 | 4 1/2 | 2 1/4 | 9 7/8 | 5 | 1 1/4 | 2 7/16 |
| 7/16 | 3/8 | 15/16 | 81.3 | 5 | 2 1/2 | 11 1/4 | 5 3/4 | 1 7/16 | 2 3/4 |
| 1/2 | 7/16 | 15/16 | 81.3 | 5 | 2 1/2 | 11 1/4 | 5 3/4 | 1 7/16 | 2 3/4 |

**Lap-riveted Girth Joints.**—(Boiler Book of Hartford Steam Boiler Inspection & Insurance Co.)—In horizontal tubular boilers, tanks, and similar vessels, it is customary to use the same size of rivets in both girth and longitudinal seams. If the heads are not stayed by tubes or through braces, the strength of the girth joints should be at least 50% of that required for the longitudinal joints. The joints in Table 5 are designed to meet these requirements; a higher efficiency could be obtained in some cases by using a different size of rivet. It is assumed in each case that the efficiency of the corresponding longitudinal joint is not greater than that of the standard design of quadruple-riveted butt-joints given in Table 4.

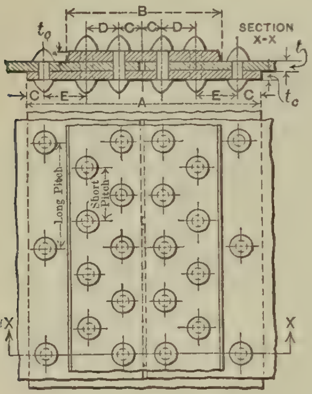


Fig. 12. Triple-riveted Butt-joint

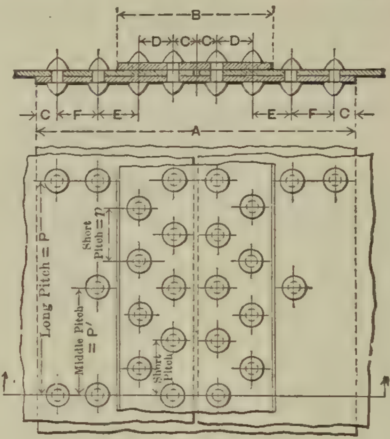


Fig. 13. Quadruple-riveted Butt-joint

**Table 3.—Dimensions and Efficiency of Triple-riveted Butt-joints.** (See Fig. 12)

Joints for plate thicknesses of 3/4 in., and 7/8 in. to 1 in. (both inclusive), fail by tearing the plate between rivet holes in the second row and shearing a rivet in the outer row. All other joints fail by tearing the plate between rivet holes in the outer row.

| Thickness of Plate, in. | Thickness of Straps, in. | Diam. of Holes, in. | Efficiency, per-cent | Long Pitch, in. | Short Pitch, in. | *A*, in. | *B*, in. | *C*, in. | *D*, in. | *E*, in. |
|---|---|---|---|---|---|---|---|---|---|---|
| 1/4 | 1/4 | 11/16 | 87.5 | 5 1/2 | 2 3/4 | 12 | 7 3/4 | 1 1/16 | 1 3/4 | 2 1/8 |
| 5/16 | 9/32 | 13/16 | 87.5 | 6 1/2 | 3 1/4 | 13 5/8 | 8 3/4 | 1 1/4 | 1 7/8 | 2 7/16 |
| 3/8 | 5/16 | 13/16 | 88.4 | 7 | 3 1/2 | 13 5/8 | 8 3/4 | 1 1/4 | 1 7/8 | 2 7/16 |
| 7/16 | 3/8 | 15/16 | 87.9 | 7 3/4 | 3 7/8 | 15 1/4 | 9 3/4 | 1 7/16 | 2 | 2 3/4 |
| 1/2 | 7/16 | 15/16 | 88.3 | 8 | 4 | 15 1/4 | 9 3/4 | 1 7/16 | 2 | 2 3/4 |
| 9/16 | 7/16 | 1 1/16 | 86.7 | 8 | 4 | 17 | 11 | 1 5/8 | 2 1/4 | 3 |
| 5/8 | 1/2 | 1 1/16 | 86.7 | 8 | 4 | 17 | 11 | 1 5/8 | 2 1/4 | 3 |
| 11/16 | 1/2 | 1 3/16 | 85.6 | 8 1/4 | 4 1/8 | 18 1/2 | 12 | 1 13/16 | 2 3/8 | 3 1/4 |
| 3/4 | 1/2 | 1 3/16 | 85.5 | 8 1/4 | 4 1/8 | 18 1/2 | 12 | 1 13/16 | 2 3/8 | 3 1/4 |
| 13/16 | 9/16 | 1 5/16 | 84.6 | 8 1/2 | 4 1/4 | 20 1/4 | 13 1/4 | 2 | 2 5/8 | 3 1/2 |
| 7/8 | 5/8 | 1 5/16 | 84.1 | 8 3/4 | 4 3/8 | 20 1/4 | 13 1/4 | 2 | 2 5/8 | 3 1/2 |
| 15/16 | 11/16 | 1 5/16 | 83.7 | 9 | 4 1/2 | 20 1/4 | 13 1/4 | 2 | 2 5/8 | 3 1/2 |
| 1 | 3/4 | 1 7/16 | 83.4 | 9 1/2 | 4 3/4 | 22 | 14 1/2 | 2 3/16 | 2 7/8 | 3 3/4 |

Where 50% of more of the load which would act on an unstayed head of the same diameter as the shell is relieved by the effect of tubes or through stays, as in tubular boilers, the strength of the girth joints should be at least 35% of that required in longitudinal joints. When the girth joints are exposed to products of combustion, the shearing strength of the rivets should be not less than 50% of the full strength of the plate corresponding to the thickness at the joint. Table 6 is designed for such conditions, using rivets of the same size as standard butt-joints. Plates thicker than 9/16 in. are not advised if the girth seams are exposed to fire or products of combustion. In those portions of the seams exposed to fire, the riveting should be started at the bottom of the boiler and carried up continuously on both sides, reaming the holes anew as the riveting is carried up to the full diameter of the boiler.

**Precautions to be Observed in Making Riveted Joints.**—Plates must be brought into close contact before the rivet is driven, either by bolts close to the rivet hole, firmly drawn up, or by a power-actuated holding device. The latter sometimes is incorporated in power-driven riveting machines.

**Table 4.—Dimensions and Efficiency of Quadruple-riveted Butt-joints.** (See Fig. 13)

Joints for 3/4 in. and 7/8 in. to 1 1/4 in. plates (both inclusive) fail by tearing the plate between rivet holes in the third row and shearing the rivets in the two outer rows. All other joints fail by tearing the plate between rivet holes in the outer row.

| Thickness of Plate, in. | Thickness of Straps, in. | Diam. of Holes, in. | Efficiency, percent | Long Pitch, in. | Middle Pitch, in. | Short Pitch, in. | *A*, in. | *B*, in. | *C*, in. | *D*, in. | *E*, in. | *F*, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/4 | 1/4 | 11/16 | 93.8 | 11 | 5 1/2 | 2 3/4 | 16 1/2 | 7 3/4 | 1 1/16 | 1 3/4 | 2 1/8 | 2 1/4 |
| 5/16 | 9/32 | 13/16 | 93.8 | 13 | 6 1/2 | 3 1/4 | 18 7/8 | 8 3/4 | 1 1/4 | 1 7/8 | 2 7/16 | 2 5/8 |
| 3/8 | 5/16 | 13/16 | 94.2 | 14 | 7 | 3 1/2 | 19 1/8 | 8 3/4 | 1 1/4 | 1 7/8 | 2 7/16 | 2 3/4 |
| 7/16 | 3/8 | 15/16 | 94.0 | 15 1/2 | 7 3/4 | 3 7/8 | 21 3/8 | 9 3/4 | 1 7/16 | 2 | 2 3/4 | 3 1/16 |
| 1/2 | 7/16 | 15/16 | 94.1 | 16 | 8 | 4 | 21 1/2 | 9 3/4 | 1 7/16 | 2 | 2 3/4 | 3 1/8 |
| 9/16 | 7/16 | 1 1/16 | 93.4 | 16 | 8 | 4 | 23 5/8 | 11 | 1 5/8 | 2 1/4 | 3 | 3 5/16 |
| 5/8 | 1/2 | 1 1/16 | 93.4 | 16 | 8 | 4 | 23 5/8 | 11 | 1 5/8 | 2 1/4 | 3 | 3 5/16 |
| 11/16 | 1/2 | 1 3/16 | 92.8 | 16 1/2 | 8 1/4 | 4 1/8 | 25 5/8 | 12 | 1 13/16 | 2 3/8 | 3 1/4 | 3 9/16 |
| 3/4 | 1/2 | 1 3/16 | 92.7 | 16 1/2 | 8 1/4 | 4 1/8 | 25 5/8 | 12 | 1 13/16 | 2 3/8 | 3 1/4 | 3 9/16 |
| 13/16 | 9/16 | 1 5/16 | 92.3 | 17 | 8 1/2 | 4 1/4 | 27 7/8 | 13 1/4 | 2 | 2 5/8 | 3 1/2 | 3 13/16 |
| 7/8 | 5/8 | 1 5/16 | 91.2 | 17 1/2 | 8 3/4 | 4 3/8 | 28 | 13 1/4 | 2 | 2 5/8 | 3 1/2 | 3 7/8 |
| 15/16 | 11/16 | 1 5/16 | 90.1 | 18 | 9 | 4 1/2 | 28 1/8 | 13 1/4 | 2 | 2 5/8 | 3 1/2 | 3 15/16 |
| 1 | 3/4 | 1 7/16 | 90.2 | 19 | 9 1/2 | 4 3/4 | 30 1/2 | 14 1/2 | 2 3/16 | 2 7/8 | 3 3/4 | 4 1/4 |
| 1 1/16 | 3/4 | 1 7/16 | 89.0 | 19 | 9 1/2 | 4 3/4 | 30 1/2 | 14 1/2 | 2 3/16 | 2 7/8 | 3 3/4 | 4 1/4 |
| 1 1/8 | 3/4 | 1 7/16 | 88.0 | 19 | 9 1/2 | 4 3/4 | 30 1/2 | 14 1/2 | 2 3/16 | 2 7/8 | 3 3/4 | 4 1/4 |
| 1 3/16 | 13/16 | 1 7/16 | 87.7 | 20 | 10 | 5 | 30 5/8 | 14 1/2 | 2 3/16 | 2 7/8 | 3 3/4 | 4 5/16 |
| 1 1/4 | 7/8 | 1 7/16 | 86.8 | 20 | 10 | 5 | 30 5/8 | 14 1/2 | 2 3/16 | 2 7/8 | 3 3/4 | 4 5/16 |

**Table 5.—Dimensions and Efficiency of Lap-riveted Girth Joints**

(For Cylindrical Vessels with Solid Unstayed Heads.)

| Thickness of Plate, in. | Diam. of Rivet Holes, in. | Single-riveted Joint | | Double-riveted Joint | | Thickness of Plate, in. | Diam. of Rivet Holes, in. | Double-riveted Joint | |
|---|---|---|---|---|---|---|---|---|---|
| | | Pitch of Rivets, in. | Efficiency, percent | Pitch of Rivets, in | Efficiency, percent | | | Pitch of Rivets, in. | Efficiency, percent |
| 1/4 | 11/16 | 1 3/4 | 60.7 | ........ | ........ | 13/16 | 1 5/16 | 4 | 66.6 |
| 5/16 | 13/16 | 1 7/8 | 56.7 | ........ | ........ | 7/8 | 1 5/16 | 4 | 61.9 |
| 3/8 | 13/16 | 1 7/8 | 56.7 | ........ | ........ | 15/16 | 1 5/16 | 4 | 57.7 |
| 7/16 | 15/16 | 2 1/8 | 55.9 | ........ | ........ | 1 | 1 7/16 | 4 1/2 | 57.7 |
| 1/2 | 15/16 | 2 1/8 | 52.0 | ........ | ........ | 1 1/16 | 1 7/16 | 4 1/2 | 54.3 |
| 9/16 | 1 1/16 | 2 3/8 | 53.3 | 3 1/4 | 67.3 | 1 1/8 | 1 7/16 | 4 1/2 | 51.3 |
| 5/8 | 1 1/16 | ........ | ........ | 3 1/4 | 67.3 | 1 3/16 | 1 7/16 | 4 1/2 | 48.6 |
| 11/16 | 1 3/16 | ........ | ........ | 3 3/4 | 68.3 | 1 1/4 | 1 7/16 | 4 1/2 | 46.2 |
| 3/4 | 1 3/16 | ........ | ........ | 3 3/4 | 63.0 | ........ | ........ | ........ | ........ |

**Table 6.—Lap-riveted Joints for Girth Seams of Horizontal Tubular Boilers**

| | Thickness of Plate, in. | | | | | |
|---|---|---|---|---|---|---|
| | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 9/16 |
| Diameter of holes, in. | 11/16 | 13/16 | 13/16 | 15/16 | 15/16 | 1 1/16 |
| Pitch of rivets, in. | 1 3/4 | 1 7/8 | 1 7/8 | 2 1/8 | 2 1/8 | 2 3/8 |
| Efficiency of joint, percent | 60.7 | 56.7 | 56.7 | 55.9 | 52.0 | 53.3 |
| Shearing strength of rivets, percent of solid plate | 67.9 | 70.8 | 59.1 | 59.4 | 52.0 | 53.3 |

Rivet holes must be perfectly aligned in the members that are to be joined. This is best insured by drilling of the members. Punched holes, usually used in structural work, are difficult of alignment, and have a stress set up around the hole which seriously impairs the strength of the plate unless it is subsequently relieved by reaming, drilling, or annealing. Punched plates or structural members in which the holes do not align never should be brought into alignment by drifting with taper pins. This practice usually causes serious injury to the plates or structural members. The best practice with punched members is to punch the holes smaller than required, and after assembly, ream or drill them to the correct size.

The rivet must completely fill the hole, and consequently must be upset throughout its entire length. Otherwise, an undue concentrated stress, either in bearing or in shearing, may be brought upon adjoining rivets. To insure complete upsetting all of the larger sizes of rivets should be power-driven. Care must be taken in machine riveting to avoid applying a riveting pressure that will set up stresses in the plate tending to crush it. It is essential in machine riveting that the pressure be so applied that the heads are concentric. Pressure should be maintained on the rivet until it has cooled sufficiently to hold the plate firmly.

## FASTENINGS FOR TIMBER JOINTS

**METHODS OF FAILURE.**—Wooden framed structures, especially roof trusses, towers, bridges and trestles tend to fail at bolted joints. Fig. 1 shows the method of failure when the wood is stronger than the bolt. In Fig. 2, where the bolt is the stronger, the joint fails by deforming and splitting the wood. Drawing the bolt up tight creates friction between the surfaces and retards slipping. Shrinkage of the wood decreases the friction and promotes failure.

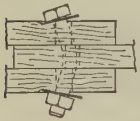


FIG. 1. Wood Stronger than Bolt

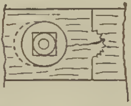


FIG. 2. Bolt Stronger than Wood

Methods of Failure of Wood Joints

**METHODS OF FASTENING WOOD JOINTS.**—Fig. 3 shows various fastenings for wood joints. At *A* is the simplest bolted joint, with a washer under the head and nut, which is adequate in well-dried timber, when there is little tension, or a large number of bolts. If the washer is made with prongs, as at *B*, tendency to slip is much reduced. The insertion of a washer of substantial thickness in counterbored holes on mating faces of the joint, as at *C*, prevents slipping, up to the point where the pressure on the end wood tends to crush the fiber. If the faces of the washer are serrated, as at *D*, the tendency to slip is further minimized.

The steel plate, with prongs at right angles, inserted between the members of the joint, as shown at *E*, covers practically the entire area of contact and makes a firm non-slipping joint. A variation of this construction is shown at *F*, in which ribbed cast-iron plates are substituted for the steel plates with prongs. A ring washer, as shown at *G*, adds largely to the strength of the joint, since it affords external resistance to slipping on the outside of the ring, as does the plain washer, but also adds internal resistance on the portion of the wood within the ring. Rings of this type should be tapered into the wood to obtain full bearing surface. A similar construction is shown at *H*, except that the washers are made of hard wood. They should be placed with the grain of wood in the washer parallel to the grain of the wood in the timber.

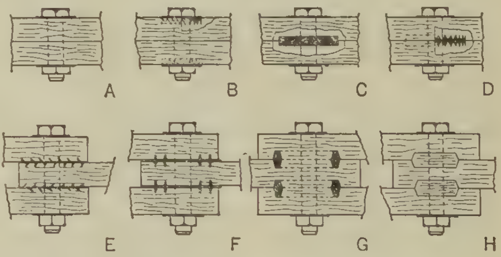


FIG. 3. Forms of Timber Joint Fastenings

# Section 10

# MECHANICAL SPRINGS

By [illegible]

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# MECHANICAL SPRINGS

By Mortimer F. Sayre

## 1. TYPES OF SPRINGS

**DEFINITIONS.**—A mechanical spring is a resilient machine member used for the absorption or redistribution of energy. Springs are made in many forms to meet a variety of conditions. The more usual types are:

**Helical Springs. Compression, Extension, or Torsion.**—A helical spring comprises a bar or wire coiled in the general form of a helix, the bar advancing like the thread of a screw.

**Cone, Double Cone, and Keg Springs.**—A cone spring is one form of helical spring, but is wound on a tapered mandrel. A double cone spring may be considered as two similar cone springs joined at the small ends. A keg spring may be considered as two similar cone springs joined at the large ends. These types usually are compression springs.

**Volute Springs** are conical springs made from a flat bar, which is so wound that each coil partially overlaps the next adjacent coil.

**Leaf or Laminated Springs** are made of flat or slightly curved bars, plates, or *leaves*, of regularly varying lengths, superposed on each other.

**Elliptical Springs; Semi-elliptical and Full Elliptical** are the commonest of the leaf springs. The latter may consist of one or more parallel units, designated as *Single Full Elliptic*, *Double Elliptic*, *Triple Elliptic*, etc. A semi-elliptic spring, or one unit of a full elliptic, comprises a number of plates of rectangular section, superposed on each other. The term elliptical spring is sometimes extended to include all forms of leaf springs.

**Spiral, Clock, or Power Springs** lie in one plane, and are wound around a fixed center, continually receding from it.

**Special Flat Springs** usually are made of a single metal strip or bar, initially flat, formed into any one of a great variety of shapes.

**Disc Springs. Ring Springs.**—The disc or " Belleville " spring comprises a series of dished or otherwise specially shaped discs, which act similarly to spring washers, in combined bending and direct stress. The ring spring is composed of a series of metal rings of alternately larger and smaller diameter, which bear against each other along wedge-shaped or conical faces. As the spring is loaded in compression, the individual units are driven together and stressed in tension or compression.

**SPRING MATERIAL.**—Table 1 covers the principal materials used for springs. The compositions given are typical, and can be modified somewhat to meet special requirements.

For hot-coiled helical compression springs, of material $^1/_2$-in. round or larger, for railroad car and locomotive suspensions and for general industrial purposes, Grade 5 or a similar steel with 0.15% minimum Si content requirement is used almost universally. Alloy steels have not proven very satisfactory. The same material generally is used for hot-wound helical extension and torsion springs, cone and keg springs, and volute springs.

Small helical and special shape coiled springs of $^3/_8$-in. round, or smaller wire, usually are wound cold from Grades 8 or 10. Grade 7 is used occasionally for severe service requirements.

For automobile valve springs (small helical compression) Grade 6 in electrically melted and cold-drawn steel has been successful. Special precautions are required throughout all fabricating processes, to eliminate surface defects in the finished wire. Straight carbon steel of Grade 8 may be used as an alternative.

Safety-valve and pop-valve springs usually are made from Grade 5. For marine work, to minimize atmospheric corrosion, they often are nickel plated.

Cone shaped upholstery springs are made of Grade 10.

For elliptic springs in railroad car and locomotive suspensions Grade 5 is popular, but Grade 6, and some special alloys, have found favor, particularly silico-manganese steel, S.A.E. 9250, S.A.E. 9260, or A.S.T.M. Specification A 59–27. Surface defects in the material used in elliptic springs are not apt to be serious; in helical springs they are fatal.

Elliptic springs for automobile suspensions are sometimes made of Grade 5, but more often of silico-manganese steel, S.A.E. 9250, S.A.E. 9260, or chrome-vanadium steel, S.A.E. 6150, or other special alloys.

### Table 1.—Physical Properties of Spring Materials

(Wallace Barnes Co., Bristol, Conn.)

| Grade | Material. Note 1 | Analysis, percent |  |  |  |  |  |  |  |  | Process of Manufacture. See Notes | Tensile Properties |  |  | Modulus of Elasticity, Millions | Hardness |  | Torsional Properties of Wire |  |  | Chief Uses. See Note |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  | C | Mn | Cr | V | Ni | Cu | Zn | Pb | Sn |  | Ultimate Strength, 1000 lb. per sq. in. | Elastic Limit, 1000 lb. per sq. in. | Elongation in 2 in., per-cent |  | Rockwell | Brinell | Ultimate Strength, 1000 lb. per sq. in. | Elastic Limit, 1000 lb. per sq. in. | Modulus in Torsion, Millions |  |
| COLD-ROLLED SPRING STEEL |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 1 | WSS | 1.10–1.19 | 0.15–0.25 | ........ | ........ | .... | .... | .... | .... | .... | 2–3 | 325–340§ | 310 330§ | ........ | 32 | C52–C55 | ...... | ........ | ...... | ........ | 23 |
| 2 | CSS | 0.90–1.05 | 0.30–0.40 | ........ | ........ | .... | .... | .... | .... | .... | 2–3 | 270–325§ | 250–310§ | ........ | 30 | C48–C52 | ...... | ........ | ...... | ........ | 24 |
| 3 | MSS | 0.80–0.90 | 0.30–0.50 | ........ | ........ | .... | .... | .... | .... | .... | 2–4 | 250–300§ | 225–270§ | ........ | 30 | C46–C50 | ...... | ........ | ...... | ........ | 25 |
| 4 | FSS | 0.70–0.80 | 0.30–0.50 | ........ | ........ | .... | .... | .... | .... | .... | 2–4 | 200–250§ | 165–210§ | ........ | 30 | C35–C45 | ...... | ........ | ...... | ........ | 26 |
| HOT-ROLLED SPRING STEEL |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 5 | HRB | 0.90–1.05 | 0.30–0.50 | ........ | ........ | ... | ... | .... | .... | .... | 5 | 180§ | 135§ | 6 | 28.5 | C40–C46 | ...... | 150 | 100 | 11.4 | 27 |
| COLD-DRAWN WIRE—SOFT PROCESS—HEAT-TREATED USUALLY AFTER FORMING |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 6 | CV | 0.45–0.55 | 0.50–0.80 | 0.80–1.10 | 0.15–0.18 | .... | .... | .... | .... | .... | 6–7–8 | 210§ | 200§ | 10 | 30 | C42–C48 | 410 | 180 | 110 | 12 | 28 |
| 7 | AHC | 0.85–0.95 | 0.30–0.40 | ........ | ........ | .... | .... | .... | .... | .... | 7–9–10–11 | 250 300§ | 210 280§ | 7 | 30 | C44–C48 | ...... | ........ | 110–130 | 12 | 29 |
| 8 | OTW | 0.60–0.70 | 0.80–1.10 | ........ | ........ | .... | .... | .... | .... | .... | 7–9–10–11 | Note 18 | Note 18 | 8 | 28.5 | C42–C46 | ...... | ........ | 110–120 | 11.4 | 30 |
| COLD-DRAWN WIRE—HARD PROCESS—COLD-DRAWN AFTER PATENTING |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 9 | MW | 0.80–0.90 | 0.30–0.40 | ........ | ........ | .... | .... | .... | .... | .... | 7–12 | Note 19 | Note 19 | 8 | 30 | ........ | ...... | ........ | 130–150 | 12 | 31–32 |
| 10 | HDS | 0.50–0.60 | 0.80–1.10 | ........ | ........ | .... | .... | .... | .... | .... | 7–13 | 200–300¶ | Note 20 | 5 | 28.5 | ........ | ...... | ........ | 110–130 | 11.4 | 30–32 |
| STAINLESS STEEL |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 11 | CW | 0.12 max. | 0.50 max. | 12.0–15.0 | ........ | .... | .... | .... | .... | .... | 14 | Note 42 | Note 21 | ........ | 28 | ........ | 140–250 | ........ | ...... | 11.2 | 33 |
| 12 | 18–8 | 0.12 max. | ........ | 18.0 | ........ | 9.0 | .... | .... | .... | .... | 14 | Note 43 | Note 21 | ........ | 28 | ........ | 140–370 | ........ | ...... | 11.2 | 33 |
| 13 | LCN | 0.12 max. | ........ | 13.0 | ........ | 2.0 | .... | .... | .... | .... | 14 | 220 max. | Note 21 | ........ | 28.5 | C40–C46 | ...... | ........ | ...... | 11.4 | 34 |
| 14 | 30C | 0.30–0.40 | 0.50 max. | 12.0–14.0 | ........ | .... | .... | .... | .... | .... | 2–9–10 | 245§ | 210§ | ........ | 30 | C54 max. | ...... | ........ | ...... | 12 | 35 |
| NON-FERROUS |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| 15 | HB | ........ | ........ | ........ | ........ | .... | 56.0 | 34.0 | .... | .... | 14–15 | 90* | 31* | 3 | 14.5 | B88‡ | ...... | ........ | ........ | 5.5 | 36 |
| 16 | SB | ........ | ........ | ........ | ........ | .... | 72.0 | 28.0 | .... | .... | 14–15 | 95† | 32† | 4.5 | 14.5 | B90‡ | ...... | ........ | 40–50 | 5.5 | 36 |
| 17 | CB | ........ | ........ | ........ | ........ | .... | 62.5 | 36.0 | 1.5 | .... | 14–16 | 73† | 31† | 9.5 | 16 | B80‡ | ...... | ........ | ........ | ........ | 37 |
| 18 | NS | ........ | ........ | ........ | ........ | 18.0 | 55.0 | 27.0 | .... | .... | 14–15 | 113† | 60† | 2 | 20 | B98‡ | ...... | ........ | ........ | 5.5 | 38 |
| 19 | PB | ........ | ........ | ........ | ........ | .... | 95.0 | .... | .... | 5.0 | 14–15 | 98† | 50† | 2.5 | 14.5 | B93‡ | ...... | ........ | 45–55 | 6 | 39 |
| 20 | PB | ........ | ........ | ........ | ........ | .... | 92.0 | .... | .... | 8.0 | 14–15 | 112† | 55† | 3 | 14 | B99‡ | ...... | ........ | ........ | 6 | 40 |
| 21 | MM | ........ | ........ | ........ | ........ | 70.0 | 30.0 | .... | .... | .... | 14–17 | 135† | 80† | Note 22 | 26 | >B100 | ...... | ........ | 50–60 | 9 | 41 |

* Sheet; wire approximately 20 to 25,000 lb. greater. † Sheet. ‡ 100 kg. § Quenched and tempered. ¶ Varies with reduction from last heat.

For special flat springs for brake beam hangers, agricultural seat springs, etc., Grade 5 or an alloy steel may be used.

Small flat springs for general purposes usually are made of Grade 4.

Music wire Grade 9 is recommended for small helical springs operating under severe conditions. It is the strongest, stiffest, toughest and most uniformly made wire on the market.

To resist corrosion, small sizes of helical springs have been made from stainless steel Grade 11. An increasing number of springs are being made from chrome-nickel-steel (18% Cr, 8% Ni) Grade 12. Other metals used are brass, phosphor-bronze, German silver, Monel metal, and alloy steels. Brass and phosphor-bronze are recommended for springs which must resist moisture; phosphor-bronze is tougher than brass and resists fatigue better. Springs of these materials must be larger than steel springs for the same

### Notes for Table 1

Note 1. 1. WSS = Watch spring steel. 2. CSS = Clock spring steel. 3. MSS = Motor spring steel. 4. FSS = Flat spring steel. 5. HRB = Hot rolled bars. 6. CV = Chrome-vanadium steel. 7. AHC = Annealed high-carbon wire. 8. OTW = Oil tempered wire. 9. MW = Music wire. 10. HDS = Hard-drawn spring wire. 11. CW = Straight chromium wire and flat strips; corrosion resistant. 12. 18–8 = Stainless steel wire and flat strip; corrosion resistant. 13. LCN = low-chromium, 2% nickel strip; corrosion resistant. 14. 30C = 0.30–0.40% carbon wire and flat strip; heat resistant. 15. HB = High brass. 16. SB = Spring brass. 17. CB = Clock brass. 18. NS = Nickel-silver. 19, 20. PB = Phosphor bronze. 21. MM = Monel metal.

Note 2. Cleaned and annealed strip is cold rolled to correct size.
Note 3. Heat treated before forming.
Note 4. Heat treated before or after forming, depending on sharpness of required bends.
Note 5. Open hearth or electric steel, rolled in rod or bar mill.
Note 6. Electric steel rod, annealed.
Note 7. Cold-drawn to size.
Note 8. Heat treated after forming.
Note 9. Annealed after drawing.
Note 10. Quenched and tempered, usually after forming.
Note 11. If formed after hardening, is blued to remove strains.
Note 12. Highest grade electric steel.
Note 13. Basic open hearth steel.
Note 14. Cold-rolled or drawn from annealed bars or strips.
Note 15. 60% reduction to spring temper.
Note 16. 35% reduction to hard temper.
Note 17. Wire reduced 70% to spring temper.

Note 18. Varies with size: 0.177 in. = 210,000 lb.; 0.054 in. = 260,000 lb.; 0.020 = 300,000 lb. Elastic Limit: 0.177 in. = 75% of ultimate strength; 0.020 in. = 85% of ultimate strength.

Note 19. Varies with size: Ultimate strength, 0.187 in. = 230,000 lb.; 0.062 in. = 300,000 lb.; 0.032 in. = 350,000 lb.; 0.015 in. = 400,000 lb. Elastic limit = 60% of ultimate strength.

Note 20. 60% of ultimate strength.

Note 21. Cold-rolled or drawn to various degrees of spring hardness; does not possess the higher elastic limits of hardened and tempered carbon steels.

Note 22. Wire, 3 to 8%; sheet, 1 to 2%.
Note 23. Mainsprings for watches.
Note 24. Clock springs; brush holder springs.
Note 25. Phonograph and motor springs; used where forming properties are important.
Note 26. Locks; clips; spring washers; flat springs generally.
Note 27. Heavy coil or flat springs; formed hot and heat treated.
Note 28. Used for special applications.
Note 29. Scale springs; highest grade wire springs, especially in coarse sizes.
Note 30. Spiral springs generally.
Note 31. High grade, small and tough springs.
Note 32. These springs have high shock resistance, but will take a small set if overloaded.
Note 33. Flat and helical springs generally to resist corrosion.
Note 34. Stiff, resilient flat springs.
Note 35. Springs to resist stress under heat. Must be polished after tempering to resist corrosion.
Note 36. Flat or coiled springs for electrical equipment and for moderate stress.
Note 37. Flat springs with sharp bends or threaded holes.
Note 38. Better class springs, and where white color is desired.
Note 39. To resist corrosion at normal temperatures.
Note 40. Stiff resilient springs, corrosion resistant.
Note 41. To resist corrosion at high temperatures (except sulphurous gases).

Note 42. Varies as size.

| | | | | | |
|---|---|---|---|---|---|
| Diam., in. = | 0.187 | .125 | .062 | .032 | .015 |
| Ult. Str. = | 140,000 | 150,000 | 175,000 | 190,000 | 200,000 |

Note 43. Varies as size.

| | | | | | |
|---|---|---|---|---|---|
| Diam., in. = | 0.187 | .125 | .062 | .032 | .015 |
| Ult. Str. = | 180,000 | 200,000 | 225,000 | 250,000 | 265,000 |

duty. This is due to the lower unit stress necessary for such materials, and also because both brass and bronze are approximately 10% heavier than steel of the same dimensions. Monel metal is useful for springs subjected to heat and moisture.

For springs to resist corrosion and fatigue, beryllium copper has been strongly recommended. This alloy of copper contains usually 2 to 2.25% beryllium, is very susceptible to heat treatment, and after heat treatment develops a tensile strength of 150,000 to 175,000 lb. per sq. in. The Johnson's elastic limit is stated to be from 75,000 to 130,000 lb. per sq. in., depending on amount of cold work and on heat treatment. Young's modulus of elasticity is 16–18,000,000. Beryllium copper increases in strength and decreases in ductility very rapidly on cold working. The resistance to fatigue in the form of springs is stated to be unusually high for a non-ferrous metal, being comparable to that of spring steel.

## 2. HELICAL COMPRESSION AND EXTENSION SPRINGS

**DESIGN FORMULAS.** **Notation.**— $d$ = diameter of (round) bar; $D$ = outside diameter of spring; $k$ = ratio $D/d$; $H$ = free length of spring; $h$ = solid length of spring; $I$ = inactive portion of length, due to flattened ends, or to loops; $n$ = number of active helical turns or coils; $P$ = compressive or extensile load on spring; $C$ = maximum allowable load; $S$ = maximum fiber stress in the section, ignoring direct shear and effect of curvature; $S'$ = true maximum fiber stress; $f$ = extension or compression of one coil with load $P$; $f_n$ = extension or compression of $n$ coils with load $P$; $G$ = torsional modulus of elasticity. All dimensions in inches; all loads in pounds.

The stress usually is assumed to be pure torsion. The standard formulas for round section are:

$$P = \frac{S\pi d^3}{8(D-d)} = 0.3927\,\frac{Sd^3}{D-d} \quad . \quad . \quad [1]; \quad f = \frac{8P(D-d)^3}{G d^4} = \frac{S\pi}{G}\cdot\frac{(D-d)^2}{d} \quad . \quad . \quad [2]$$

If $S$ be the maximum permissible stress, $f$ in the last formula is the maximum permissible deflection per coil. The deflection in the whole spring may be obtained by multiplying $f$ by the number of effective coils in the spring. In helical compression springs (see Fig. 1) it is customary practice to design the springs with sufficient space between coils so that they can be closed solid in the manufacturer's test, and return to the desired free height. Then $f$ becomes the allowable space between coils, and $C$ is the capacity or load required to close the spring solid. The free height is

$$H = n(d + f) + I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

The solid height is

$$h = nd + I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

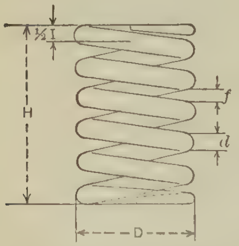


FIG. 1. Helical Compression Spring

In fabricating hot-rolled helical compression springs, the ends of the bars are tapered before coiling, so that the finished spring will have flat ends. Sometimes the ends are ground after coiling, so that the ends will be truly square with the helical axis; this is regular practice with valve springs. The inactive length added to the spring by the two tapered ends is about $I$ = 1.25 $d$ for ordinary flat ends, and somewhat less for ground ends.

**CORRECTION FOR CURVATURE (THE RÖVER CORRECTION).**—The stress in a spring due to direct shear, while neglected in the above standard formulas, often is considerable. The formulas also neglect the curvature of the bar. These errors have been examined mathematically by A. Röver, and his conclusions verified experimentally by A. M. Wahl, with very satisfactory accuracy. The correction (for the two errors combined) is as follows: If $D/d = k$, and the true maximum stress = $S'$, then:

$$\frac{S'}{S} = \frac{4k-5}{4k-8} + \frac{0.615}{k-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

| For $k$ = | 3.5 | 4.0 | 4.5 | 5.0 | 6.0 | 7.0 | 8.0 | 10.0 | 12.0 | 16.0 |
|---|---|---|---|---|---|---|---|---|---|---|
| $S'/S$ = | 1.746 | 1.580 | 1.476 | 1.404 | 1.311 | 1.253 | 1.213 | 1.162 | 1.131 | 1.095 |

This true maximum stress $S'$ is present at only a single point element in the section, the point on the surface of the bar at the extreme inside of the helix. Care must be exercised in using a localized stress such as this as the sole criterion of design. With hot-rolled springs, or other springs with moderately roughened surface, it is probable that stress concentration due to curvature will be masked, in part, by the effects of other stress concentrations due to irregularities of surface. The values of $S'/S$ given above then exaggerate the true effect. For music wire springs, or springs with smooth wire surface, carefully made so that other stress concentrations are absent, it is probable that the

Röver curvature correction should be used almost, if not quite, in full. This is particularly true if fatigue is an important factor. Engineers are not in agreement on these points and further experimental work is needed.

Note particularly however that if the Röver curvature correction factor is to be used in design, the allowable working stresses must be changed to correspond. This is necessary if the two formulas are to give concordant results over that range of values of $k$ within which the old standard formula has been satisfactory. Taking $k = 8.0$ as a mid-point in that range, the corresponding value of $S'/S$ is 1.213. If a stress of 100,000 lb. per sq. in. has given good results with a given material and the standard formula, then for the same material, using a formula containing the Röver correction, a design working stress of 121,300 lb. per sq. in. should be used, and pro rata for other stresses.

**TABLES FOR HELICAL SPRING DESIGN.**—Table 2 is taken from A.S.T.M. Tentative Specifications for Heat-Treated Carbon Steel Helical Springs (A125–33). It represents maximum test loads for typical railroad type springs. The Röver curvature correction is not included. A torsional modulus of 10,500,000 was used to compute deflections. When using this table to find total deflection, the deflection per coil should be multiplied by the number of completely free coils in the spring, treating one coil at each end, measured from tip of tapered bar, as inactive.

Table 3, based partially on data prepared for the Wallace Barnes Co., gives for smaller sizes of springs, three items: 1. Load in lb. at 100,000 lb. per sq. in. fiber stress, Röver curvature correction ignored. 2. Load in lb. at 100,000 lb. per sq. in. fiber stress, computed on basis of Röver curvature correction formula. 3. Deflection in inches per coil per 1000 lb. of load, computed for a modulus of elasticity of 11,500,000.

In these two tables, working loads and the corresponding deflections for other unit stress than 100,000 lb. per sq. in. may be obtained by changing the tabular values in the corresponding ratio of stresses. For other moduli of elasticity than those used for the tables, vary the deflection in inverse ratio to the modulus.

EXAMPLES.—1. Find safe load and deflection for a steel spring of 15 effective coils, 0.50 in. outside diam., made of 16 gage wire (0.0625 in. diam.). *Solution.* From Table 3, safe load $w_1$ at 100,000 lb. fiber stress for this spring is 21.9 lb., ignoring curvature correction, or 18.1 lb. corrected for curvature. Deflection is 0.00382 in. per lb. per coil; total deflection $= 0.00382 \times 15 \times 18.1 = 1.037$ in. under full load.

2. Design a spring to carry a load of 50 lb. with a deflection of 2.5 in. Use fiber stress of 120,000 lb., including correction for curvature. *Solution.* A load $w_2$ of 50 lb. for 120,000 lb. fiber stress is equivalent, by proportion, to a load of 41.7 lb. under 100,000 lb. fiber stress. The following choices are possible among others:

| | Wire diam., in. | Coil diam., in. | $w_2$ | $f$ | No. of coils |
|---|---|---|---|---|---|
| 1. | 0.080 | 0.4375 | 41.5 | 0.000776 | 64.5 |
| 2. | 0.0915 | 0.625 | 44.7 | 0.001507 | 33.2 |
| 3. | 0.1055 | 1.000 | 43.7 | 0.00402 | 12.5 |

The desired deflection per lb. for this spring is 2.50/50, or 0.05 in. The required number of effective coils is $0.05/f$ in each case.

3. Design a phosphor-bronze spring to carry a load of 50 lb. with deflection 2.50 in., using maximum stress of 50,000 lb. and torsional modulus of 6,000,000. *Solution.* A load of 50 lb. with a unit stress of 50,000 lb. is equivalent to 100 lb. load under 100,000 lb. stress. The following choices are possible among others:

| | Wire diam., in. | Coil diam., in. | $w_2$ | $f$ | No. of coils |
|---|---|---|---|---|---|
| 1. | 0.1205 | 0.500 | 117.0 | 0.000180 | 144 |
| 2. | 0.125 | 0.625 | 109.4 | 0.000356 | 77.8 |
| 3. | 0.135 | 0.875 | 102.0 | 0.000848 | 34.9 |
| 4. | 0.148 | 1.000 | 118.0 | 0.000897 | 29.1 |

The desired deflection per pound is 2.5/50, or 0.05 in. Since phosphor-bronze with a modulus of 6,000,000 is to be used instead of steel with a modulus of 11,500,000, the deflection per pound will be greater than those given in the table in the ratio 11.5/6, and the number of coils will equal $(0.05 \times 6)/(11.5 \times f)$.

**MODULUS OF ELASTICITY.**—The normal value of the torsional modulus of carbon steel ranges from 11,000,000 to 12,000,000, averaging 11,500,000. The actual value to be used in design, however, varies with three factors.

1. The modulus decreases with increasing temperature at the rate of approximately 0.015% per deg. F. change, this rate of change increasing at higher temperature. Except for precision instruments or for high temperature work, this change usually can be neglected.

2. By initially overloading a spring, the metal in the spring may be cold-worked, and the apparent elastic limit of the spring raised, so that higher allowable stresses may be used in design. The effect of this overstressing is to lower the torsional modulus of the spring under future loading. In heavily overstressed springs, the torsional modulus may be reduced by as much as 5 to 15%.

**Table 2.—Test Loads and Deflections for Heat-treated Helical Carbon Steel Springs**

Corresponding to a maximum fiber stress of 100,000 lb. per sq. in. (A.S.T.M. Specification A125–33)
*L* = Test load, lb.; *D* = Deflection, in. per turn, for *G* = 500,000 lb. per sq. in.

| Outside Diam. of Spring, in. | | Diameter of Bar, in. | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 1/2 | 9/16 | 5/8 | 11/16 | 3/4 | 13/16 | 7/8 | 15/16 | 1 | 1 1/16 | 1 1/8 | 1 3/16 | 1 1/4 | 1 5/16 | 1 3/8 | 1 1/2 | 1 5/8 |
| 2 | *L* | 3272 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.134 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2 1/4 | *L* | 2805 | 4140 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.183 | 0.152 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2 1/2 | *L* | 2454 | 3606 | 5113 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.239 | 0.200 | 0.168 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2 3/4 | *L* | 2181 | 3194 | 4511 | 6187 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.303 | 0.254 | 0.216 | 0.185 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 | *L* | 1963 | 2866 | 4036 | 5518 | 7363 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.374 | 0.316 | 0.270 | 0.233 | 0.202 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 1/4 | *L* | 1785 | 2600 | 3652 | 4980 | 6627 | 8641 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.452 | 0.384 | 0.330 | 0.285 | 0.249 | 0.219 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 1/2 | *L* | 1636 | 2379 | 3334 | 4537 | 6025 | 7837 | 10022 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.538 | 0.459 | 0.396 | 0.344 | 0.302 | 0.266 | 0.235 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 3/4 | *L* | 1510 | 2192 | 3065 | 4166 | 5522 | 7171 | 9150 | 11503 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.632 | 0.540 | 0.467 | 0.408 | 0.359 | 0.317 | 0.282 | 0.252 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 | *L* | 1402 | 2033 | 2840 | 3852 | 5097 | 6608 | 8418 | 10564 | 13090 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | 0.733 | 0.628 | 0.545 | 0.477 | 0.421 | 0.374 | 0.334 | 0.299 | 0.269 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 1/4 | *L* | ..... | 1895 | 2644 | 3581 | 4733 | 6127 | 7795 | 9767 | 12083 | 14777 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | ..... | 0.723 | 0.629 | 0.553 | 0.489 | 0.435 | 0.389 | 0.350 | 0.316 | 0.286 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 1/2 | *L* | ..... | 1774 | 2474 | 3347 | 4418 | 5712 | 7257 | 9081 | 11220 | 13702 | 16567 | ..... | ..... | ..... | ..... | ..... | ..... |
| | *D* | ..... | 0.824 | 0.718 | 0.632 | 0.561 | 0.500 | 0.449 | 0.405 | 0.366 | 0.333 | 0.302 | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 3/4 | *L* | ..... | ..... | 2324 | 3141 | 4142 | 5352 | 6789 | 8486 | 10472 | 12773 | 15424 | 18460 | ..... | ..... | ..... | ..... | ..... |
| | *D* | ..... | ..... | 0.815 | 0.718 | 0.638 | 0.571 | 0.513 | 0.464 | 0.420 | 0.383 | 0.349 | 0.319 | ..... | ..... | ..... | ..... | ..... |
| 5 | *L* | ..... | ..... | 2191 | 2959 | 3898 | 5030 | 6377 | 7964 | 9817 | 11962 | 14429 | 17250 | 20450 | ..... | ..... | ..... | ..... |
| | *D* | ..... | ..... | 0.916 | 0.810 | 0.720 | 0.646 | 0.582 | 0.527 | 0.479 | 0.436 | 0.399 | 0.366 | 0.336 | ..... | ..... | ..... | ..... |
| 5 1/4 | *L* | ..... | ..... | ..... | 2797 | 3681 | 4746 | 6013 | 7502 | 9240 | 11248 | 13554 | 16185 | 19175 | 22545 | ..... | ..... | ..... |
| | *D* | ..... | ..... | ..... | 0.906 | 0.808 | 0.725 | 0.654 | 0.594 | 0.540 | 0.494 | 0.452 | 0.416 | 0.383 | 0.353 | ..... | ..... | ..... |
| 5 1/2 | *L* | ..... | ..... | ..... | 2651 | 3488 | 4493 | 5688 | 7091 | 8726 | 10614 | 12780 | 15250 | 18045 | 21200 | 24750 | ..... | ..... |
| | *D* | ..... | ..... | ..... | 1.008 | 0.900 | 0.809 | 0.731 | 0.664 | 0.606 | 0.554 | 0.509 | 0.468 | 0.432 | 0.400 | 0.370 | ..... | ..... |
| 5 3/4 | *L* | ..... | ..... | ..... | ..... | 3313 | 4266 | 5396 | 6723 | 8267 | 10049 | 12089 | 14415 | 17040 | 20005 | 23335 | ..... | ..... |
| | *D* | ..... | ..... | ..... | ..... | 0.997 | 0.898 | 0.812 | 0.739 | 0.675 | 0.618 | 0.569 | 0.524 | 0.484 | 0.449 | 0.416 | ..... | ..... |
| 6 | *L* | ..... | ..... | ..... | ..... | 3155 | 4060 | 5133 | 6391 | 7854 | 9540 | 11469 | 13665 | 16145 | 18940 | 22070 | 29450 | ..... |
| | *D* | ..... | ..... | ..... | ..... | 1.099 | 0.991 | 0.898 | 0.818 | 0.748 | 0.686 | 0.632 | 0.583 | 0.540 | 0.501 | 0.465 | 0.40- | ..... |
| 6 1/4 | *L* | ..... | ..... | ..... | ..... | ..... | 3874 | 4894 | 6090 | 7480 | 9080 | 10910 | 12990 | 15340 | 17980 | 20940 | 27900 | ..... |
| | *D* | ..... | ..... | ..... | ..... | ..... | 1.088 | 0.987 | 0.901 | 0.824 | 0.758 | 0.699 | 0.645 | 0.598 | 0.556 | 0.517 | 0.450 | ..... |
| 6 1/2 | *L* | ..... | ..... | ..... | ..... | ..... | 3703 | 4676 | 5816 | 7140 | 8660 | 10402 | 12379 | 14610 | 17115 | 19920 | 26505 | 34565 |
| | *D* | ..... | ..... | ..... | ..... | ..... | 1.191 | 1.081 | 0.987 | 0.905 | 0.833 | 0.768 | 0.711 | 0.659 | 0.613 | 0.571 | 0.498 | 0.437 |
| 6 3/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | 4478 | 5566 | 6829 | 8282 | 9940 | 11823 | 13945 | 16325 | 18995 | 25245 | 32880 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | 1.180 | 1.078 | 0.989 | 0.910 | 0.841 | 0.779 | 0.724 | 0.674 | 0.628 | 0.550 | 0.483 |
| 7 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | 4295 | 5337 | 6545 | 7933 | 9517 | 11314 | 13340 | 15610 | 18150 | 24095 | 31350 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | 1.283 | 1.172 | 1.077 | 0.993 | 0.918 | 0.851 | 0.791 | 0.737 | 0.688 | 0.603 | 0.532 |
| 7 1/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5125 | 6283 | 7612 | 9129 | 10847 | 12780 | 14950 | 17375 | 23050 | 29955 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.271 | 1.168 | 1.078 | 0.998 | 0.926 | 0.861 | 0.804 | 0.751 | 0.659 | 0.582 |
| 7 1/2 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 4930 | 6041 | 7317 | 8770 | 10418 | 12270 | 14345 | 16665 | 22090 | 28680 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.374 | 1.263 | 1.167 | 1.081 | 1.004 | 0.935 | 0.873 | 0.816 | 0.718 | 0.635 |
| 7 3/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5817 | 7043 | 8440 | 10020 | 11800 | 13790 | 16015 | 21205 | 27510 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.363 | 1.259 | 1.167 | 1.085 | 1.011 | 0.945 | 0.884 | 0.779 | 0.691 |
| 8 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5610 | 6789 | 8133 | 9653 | 11360 | 13275 | 15405 | 20390 | 26430 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.466 | 1.355 | 1.256 | 1.169 | 1.090 | 1.020 | 0.955 | 0.842 | 0.748 |
| 8 1/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 7847 | 9312 | 10955 | 12795 | 14845 | 19635 | 25435 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.350 | 1.256 | 1.173 | 1.097 | 1.028 | 0.908 | 0.808 |
| 8 1/2 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 7581 | 8993 | 10575 | 12350 | 14325 | 18935 | 24510 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.447 | 1.347 | 1.258 | 1.177 | 1.104 | 0.977 | 0.870 |
| 8 3/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 8696 | 10225 | 11935 | 13840 | 18280 | 23650 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.441 | 1.346 | 1.260 | 1.183 | 1.048 | 0.934 |
| 9 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 8418 | 9895 | 11550 | 13385 | 17670 | 22845 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.538 | 1.437 | 1.347 | 1.265 | 1.122 | 1.001 |

Table 2.—*Continued*

| Outside Diam. of Spring, in. | | Diameter of Bar, in. | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 1/2 | 9/16 | 5/8 | 11/16 | 3/4 | 13/16 | 7/8 | 15/16 | 1 | 1 1/16 | 1 1/8 | 1 3/16 | 1 1/4 | 1 5/16 | 1 3/8 | 1 1/2 | 1 5/8 |
| 9 1/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 9585 | 11185 | 12960 | 17100 | 22100 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.532 | 1.436 | 1.349 | 1.198 | 1.070 |
| 9 1/2 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 9295 | 10840 | 12565 | 16565 | 21395 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.629 | 1.528 | 1.436 | 1.276 | 1.141 |
| 9 3/4 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 10520 | 12190 | 16065 | 20740 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.622 | 1.526 | 1.357 | 1.215 |
| 10 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 10220 | 11835 | 15590 | 20120 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.720 | 1.619 | 1.441 | 1.291 |
| 10 1/2 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 11185 | 14725 | 18985 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.811 | 1.615 | 1.450 |
| 11 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 13950 | 17975 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.800 | 1.618 |
| 11 1/2 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 13250 | 17065 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.994 | 1.795 |
| 12 | *L* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 16240 |
| | *D* | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1.981 |

3. In compression helical springs with flattened ends, a certain amount of twisting occurs in the metal of the so-called inactive coils, as well as in the completely free coils. The true equivalent number of coils in the spring is equal to the number of completely free coils, plus one-quarter coil for each flattened end. It has been customary to compute total deflection by multiplying deflection per coil by the number of completely free coils, ignoring deflection in the flattened ends. When this was done, the error was tacitly corrected in design by using a somewhat lowered value of torsional modulus, so that for springs of usual length, correct numerical results were obtained. This accounts for the low torsional modulus values of 10,500,000 to 11,000,000 often used in practical design.

For most extension springs a similar end effect is present. In springs with the commonly used hook ends, formed by bending up the last half coil, each hook is equivalent to 0.1 coil added effective length.

**HOT-COILED HELICAL COMPRESSION SPRINGS.**—The permissible design stress varies somewhat with the service required. For Grade 5 spring steel, in as-formed condition (commonly used for heavy railroad springs), ignoring the Röver correction, 80,000 to 95,000 lb. per sq. in. may be used. By coiling the springs to an initial height much greater than the final free height desired and then compressing the springs beyond their yield point, entrapped stresses may be introduced and the apparent maximum fiber stress appreciably increased, up to 110,000 to 115,000. At these limiting figures the life of a spring in any service, except a practically steady load, would be very short. Provided the maximum normal load in service is well below the peak capacity, and the travel or range of movement is not too great or too frequently applied, a design stress of 90,000 lb. per sq. in. based on peak capacity usually is safe.

For smaller sizes of helical springs (wire 3/8 in. diam. or smaller) used in compression, stresses of 100,000 or sometimes as high as 120,000 lb. per sq. in. may be used, if impact stresses or fatigue conditions are not severe. It is customary to reduce these allowable stresses by 20% for springs used in extension. Further reductions must be made if the springs must withstand large numbers of repetitions of loading.

**HOT-COILED HELICAL EXTENSION SPRINGS.**—The fundamental formulas are the same as for compression springs. Design stresses, however, should be lower. A maximum stress of not over 60,000 lb. per sq. in. for large springs under service conditions is recommended. Even at this stress the spring will usually take a small amount of permanent set when tested. In figuring overall length under load, allowance must be made for this set. Alternatively, the spring may be designed so that at the desired unloaded length, there will be a space between coils of 40% to 50% of the desired elongation per coil. This gives leeway for the manufacturer to coil the spring close, and then apply such test loads that when unloaded it will have the desired free length. By so doing, the apparent elastic limit is raised, and the spring should take no further permanent set under maximum stresses up to 60,000 lb. per sq. in.

**COLD-COILED HELICAL COMPRESSION SPRINGS.**—Several different grades of steel wire are in common use. Permissible design stresses vary not only with the grade, but also with the size of wire, and type of service. When not subject to severe impact or

*Continued on p.* 10–12.

**Table 3.—Safe Loads and Deflections for Smaller Sizes of Helical Extension and Compression Springs**

$w_1$ = load, lb., for fiber stress of 100,000 lb. per sq. in., curvature correction neglected.
$w_2$ = load, lb., for fiber stress of 100,000 lb. per sq. in., curvature correction considered.
$f$ = deflection, in. per 1000 coils, per lb. of load.

| A.W.G. No. & Wire Diam. | | Outside Diameter of Spring, in. | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | .125 | .156 | .187 | .219 | .250 | .312 | .375 | .437 | .500 | .625 | .750 | .875 | 1.00 | 1.25 | 1.50 | 1.75 | 2.00 |
| No. 25 | $w_1$ | 2.99 | 2.31 | 1.87 | 1.57 | 1.36 | 1.07 | 0.88 | 0.75 | 0.66 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| .020 | $w_2$ | 2.30 | 1.90 | 1.59 | 1.38 | 1.20 | 0.96 | 0.82 | 0.70 | 0.62 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| in. | $f$ | 5.03 | 11.0 | 20.4 | 34.1 | 52.9 | 108 | 194 | 317 | 481 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_1$ | 4.06 | 3.12 | 2.52 | 2.12 | 1.82 | 1.43 | 1.18 | 1.00 | 0.87 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| .022 in. | $w_2$ | 3.07 | 2.50 | 2.10 | 1.83 | 1.60 | 1.30 | 1.09 | 0.93 | 0.82 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | 3.24 | 7.19 | 13.5 | 22.7 | 35.2 | 72.8 | 131 | 213 | 324 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_1$ | 5.38 | 4.12 | 3.31 | 2.78 | 2.40 | 1.87 | 1.54 | 1.31 | 1.14 | 0.90 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| .024 in. | $w_2$ | 3.90 | 3.20 | 2.73 | 2.36 | 2.08 | 1.70 | 1.42 | 1.21 | 1.07 | 0.85 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | 2.16 | 4.85 | 9.16 | 15.5 | 24.2 | 50.3 | 90.7 | 148 | 226 | 455 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_1$ | 6.96 | 5.30 | 4.25 | 3.57 | 3.07 | 2.40 | 1.97 | 1.67 | 1.46 | 1.14 | 0.99 | ..... | ..... | ..... | ..... | ..... | ..... |
| .026 in. | $w_2$ | 4.90 | 4.05 | 3.40 | 2.94 | 2.61 | 2.13 | 1.81 | 1.54 | 1.35 | 1.07 | 0.94 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | 1.48 | 3.37 | 6.41 | 10.9 | 17.1 | 35.8 | 64.7 | 106 | 162 | 327 | 578 | ..... | ..... | ..... | ..... | ..... | ..... |
| No. 22 | $w_1$ | 9.54 | 7.21 | 5.76 | 4.81 | 4.14 | 3.22 | 2.65 | 2.24 | 1.95 | 1.53 | 1.27 | ..... | ..... | ..... | ..... | ..... | ..... |
| .0286 | $w_2$ | 6.39 | 5.36 | 4.53 | 3.97 | 3.48 | 2.85 | 2.38 | 2.03 | 1.81 | 1.44 | 1.20 | ..... | ..... | ..... | ..... | ..... | ..... |
| in. | $f$ | .932 | 2.16 | 4.17 | 7.13 | 11.3 | 23.8 | 43.2 | 71.1 | 100 | 221 | 390 | ..... | ..... | ..... | ..... | ..... | ..... |
| No. 21 | $w_1$ | ..... | 10.1 | 8.02 | 6.67 | 5.73 | 4.45 | 3.64 | 3.08 | 2.67 | 2.11 | 1.74 | 1.48 | ..... | ..... | ..... | ..... | ..... |
| .0317 | $w_2$ | ..... | 7.28 | 6.10 | 5.35 | 4.70 | 3.84 | 3.21 | 2.78 | 2.44 | 1.95 | 1.65 | 1.40 | ..... | ..... | ..... | ..... | ..... |
| in. | $f$ | ..... | 1.33 | 2.60 | 4.50 | 7.17 | 15.2 | 27.9 | 46.0 | 70.7 | 144 | 258 | 415 | ..... | ..... | ..... | ..... | ..... |
| No. 20 | $w_1$ | ..... | 13.6 | 10.8 | 8.96 | 7.68 | 5.93 | 4.86 | 4.10 | 3.55 | 2.80 | 2.31 | 1.97 | 1.72 | ..... | ..... | ..... | ..... |
| .0348 | $w_2$ | ..... | 9.28 | 7.90 | 6.94 | 6.16 | 5.04 | 4.25 | 3.68 | 3.25 | 2.56 | 2.17 | 1.87 | 1.61 | ..... | ..... | ..... | ..... |
| in. | $f$ | ..... | .841 | 1.69 | 2.95 | 4.73 | 10.2 | 18.7 | 31.0 | 47.7 | 97.5 | 174 | 281 | 425 | ..... | ..... | ..... | ..... |
| No 19 | $w_1$ | ..... | ..... | 18.4 | 15.2 | 12.9 | 9.92 | 8.08 | 6.80 | 5.88 | 4.62 | 3.81 | 3.24 | 2.82 | ..... | ..... | ..... | ..... |
| .0410 | $w_2$ | ..... | ..... | 12.6 | 11.1 | 9.93 | 8.11 | 6.88 | 5.91 | 5.21 | 4.19 | 3.57 | 3.04 | 2.67 | ..... | ..... | ..... | ..... |
| in. | $f$ | ..... | ..... | .774 | 1.38 | 2.25 | 4.93 | 9.17 | 15.3 | 23.8 | 49.0 | 87.7 | 143 | 217 | ..... | ..... | ..... | ..... |
| No. 18 | $w_1$ | ..... | ..... | 30.0 | 24.6 | 20.8 | 15.8 | 12.8 | 10.8 | 9.31 | 7.30 | 5.99 | 5.08 | 4.41 | ..... | ..... | ..... | ..... |
| .0475 | $w_2$ | ..... | ..... | 18.9 | 16.8 | 15.1 | 12.4 | 10.6 | 9.20 | 8.04 | 6.53 | 5.50 | 4.71 | 4.12 | ..... | ..... | ..... | ..... |
| in. | $f$ | ..... | ..... | .375 | .686 | 1.13 | 2.54 | 4.80 | 8.10 | 12.7 | 26.3 | 47.4 | 77.4 | 118 | ..... | ..... | ..... | ..... |
| No. 17 | $w_1$ | ..... | ..... | ..... | 37.3 | 31.5 | 23.8 | 19.2 | 16.1 | 13.8 | 10.8 | 8.86 | 7.52 | 6.52 | 5.16 | ..... | ..... | ..... |
| .054 | $w_2$ | ..... | ..... | ..... | 23.9 | 21.6 | 18.0 | 15.4 | 13.3 | 11.7 | 9.35 | 7.90 | 6.80 | 6.04 | 4.84 | ..... | ..... | ..... |
| in. | $f$ | ..... | ..... | ..... | .366 | .616 | 1.41 | 2.71 | 4.62 | 7.26 | 15.2 | 27.6 | 45.3 | 69.3 | 140 | ..... | ..... | ..... |
| No. 16 | $w_1$ | ..... | ..... | ..... | ..... | 51.2 | 38.3 | 30.7 | 25.5 | 21.9 | 17.0 | 13.9 | 11.8 | 10.2 | 8.07 | ..... | ..... | ..... |
| .0625 | $w_2$ | ..... | ..... | ..... | ..... | 32.5 | 27.3 | 23.4 | 20.8 | 18.1 | 14.6 | 12.2 | 10.5 | 9.34 | 7.49 | ..... | ..... | ..... |
| in. | $f$ | ..... | ..... | ..... | ..... | .300 | .712 | 1.39 | 2.40 | 3.82 | 8.11 | 14.8 | 24.4 | 37.6 | 76.3 | ..... | ..... | ..... |
| No. 15 | $w_1$ | ..... | ..... | ..... | ..... | 82.2 | 60.6 | 48.2 | 39.9 | 34.2 | 26.5 | 21.6 | 18.2 | 15.7 | 12.4 | 10.2 | ..... | ..... |
| .072 | $w_2$ | ..... | ..... | ..... | ..... | 46.8 | 40.3 | 35.0 | 31.2 | 27.3 | 22.2 | 18.6 | 16.0 | 14.2 | 11.4 | 9.62 | ..... | ..... |
| in. | $f$ | ..... | ..... | ..... | ..... | .146 | .360 | .720 | 1.26 | 2.03 | 4.38 | 8.07 | 13.4 | 20.7 | 42.3 | 75.4 | ..... | ..... |
| No. 14 | $w_1$ | ..... | ..... | ..... | ..... | ..... | 85.9 | 67.9 | 56.1 | 47.7 | 36.8 | 29.9 | 25.3 | 21.8 | 17.1 | 14.1 | ..... | ..... |
| .080 | $w_2$ | ..... | ..... | ..... | ..... | ..... | 53.9 | 47.1 | 41.5 | 36.8 | 30.4 | 25.4 | 21.9 | 19.4 | 15.5 | 13.1 | ..... | ..... |
| in. | $f$ | ..... | ..... | ..... | ..... | ..... | .213 | .436 | .776 | 1.26 | 2.75 | 5.11 | 8.54 | 13.4 | 27.2 | 48.6 | ..... | ..... |
| No. 13 | $w_1$ | ..... | ..... | ..... | ..... | ..... | 136 | 106 | 86.8 | 73.6 | 56.4 | 45.7 | 38.4 | 33.1 | 25.9 | 21.4 | 18.1 | ..... |
| .0915 | $w_2$ | ..... | ..... | ..... | ..... | ..... | 76.1 | 68.2 | 60.7 | 54.3 | 44.7 | 37.7 | 32.7 | 28.9 | 23.2 | 19.5 | 16.9 | ..... |
| in. | $f$ | ..... | ..... | ..... | ..... | ..... | .107 | .226 | .411 | .677 | 1.51 | 2.83 | 4.77 | 7.44 | 15.4 | 27.7 | 45.3 | ..... |
| No. 12 | $w_1$ | ..... | ..... | ..... | ..... | ..... | 222 | 171 | 139 | 117 | 88.8 | 71.5 | 59.8 | 51.5 | 40.2 | 33.0 | 28.0 | ..... |
| .1055 | $w_2$ | ..... | ..... | ..... | ..... | ..... | 107 | 99.8 | 90.0 | 81.9 | 68.0 | 57.3 | 49.3 | 43.7 | 35.3 | 29.8 | 25.6 | ..... |
| in. | $f$ | ..... | ..... | ..... | ..... | ..... | .050 | .110 | .205 | .345 | .787 | 1.50 | 2.56 | 4.02 | 8.42 | 15.2 | 25.0 | ..... |
| No. 11 | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | 270 | 216 | 181 | 136 | 109 | 91.0 | 78.0 | 60.8 | 49.7 | 42.1 | ..... |
| .1205 | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | 137 | 127 | 117 | 98.3 | 85.0 | 73.4 | 64.3 | 52.8 | 44.1 | 38.1 | ..... |
| in. | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | .054 | .105 | .180 | .424 | .823 | 1.42 | 2.24 | 4.75 | 8.66 | 14.3 | ..... |
| | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | 306 | 244 | 204 | 153 | 123 | 102 | 87.6 | 68.2 | 55.7 | 47.2 | ..... |
| .125 in. | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | 149 | 140 | 129 | 109 | 93.7 | 81.5 | 72.3 | 58.6 | 49.4 | 42.3 | ..... |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | .044 | .087 | .150 | .356 | .696 | 1.20 | 1.91 | 4.06 | 7.41 | 12.2 | ..... |
| No. 10 | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 318 | 264 | 197 | 157 | 130 | 111 | 86.4 | 70.6 | 59.7 | 51.7 |
| .135 | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 170 | 159 | 136 | 116 | 102 | 90.2 | 73.4 | 61.8 | 52.9 | 46.8 |
| in. | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .058 | .102 | .246 | .487 | .848 | 1.35 | 2.90 | 5.32 | 8.82 | 13.6 |
| No. 9 | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 442 | 364 | 268 | 213 | 176 | 150 | 116 | 94.6 | 79.8 | 69.1 |
| .148 | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 214 | 199 | 173 | 151 | 133 | 118 | 96.7 | 81.2 | 70.0 | 61.9 |
| in. | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .035 | .063 | .157 | .316 | .557 | .897 | 1.94 | 3.58 | 5.96 | 9.21 |

**Table 3** (*Continued*)

| A.W.G. No. & Wire Diam. | | Outside Diameter of Spring, in. | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | .437 | .500 | .625 | .750 | .875 | 1.00 | 1.25 | 1.50 | 1.75 | 2.00 | 2.50 | 3.00 | 3.50 | 4.00 | 4.50 | 5.00 | 6.00 |
| 5/32 in. | $w_1$ | ..... | 434 | 318 | 252 | 208 | 177 | 136 | 111 | 93 6 | 81.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | 225 | 202 | 176 | 155 | 139 | 113 | 95.2 | 82.2 | 71.8 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | .047 | .120 | .244 | .432 | .701 | 1.53 | 2.83 | 4.72 | 7.32 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| No 8 .162 in. | $w_1$ | ..... | 493 | 360 | 284 | 234 | 199 | 153 | 125 | 105 | 90.8 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | 248 | 223 | 195 | 172 | 153 | 125 | 106 | 91.6 | 80.5 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | .039 | .100 | .205 | .366 | .594 | 1.30 | 2.42 | 4.04 | 6.27 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| No. 7 .177 in. | $w_1$ | ..... | ..... | 484 | 379 | 311 | 264 | 201 | 164 | 138 | 119 | 93.4 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | 277 | 247 | 221 | 198 | 162 | 137 | 119 | 104 | 84.4 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | .064 | .133 | .241 | .395 | .876 | 1.64 | 2.76 | 4.29 | 8.88 | ..... | ..... | ..... | ..... | ..... | ..... |
| 3/16 in. | $w_1$ | ..... | ..... | 592 | 461 | 376 | 318 | 243 | 197 | 165 | 142 | 111 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | 322 | 291 | 258 | 230 | 193 | 162 | 141 | 124 | 100 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | .047 | .100 | .183 | .302 | .675 | 1.27 | 2.15 | 3.35 | 6.96 | ..... | ..... | ..... | ..... | ..... | ..... |
| No. 6 .192 in. | $w_1$ | ..... | ..... | 641 | 497 | 406 | 343 | 262 | 212 | 178 | 153 | 120 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | 339 | 308 | 277 | 246 | 204 | 171 | 150 | 133 | 106 | ..... | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | .041 | .089 | .163 | .270 | .606 | 1.15 | 1.94 | 3.02 | 6.29 | ..... | ..... | ..... | ..... | ..... | ..... |
| No. 5 .207 in. | $w_1$ | ..... | ..... | 831 | 641 | 520 | 438 | 333 | 269 | 225 | 194 | 152 | 124 | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | 408 | 378 | 341 | 308 | 255 | 215 | 187 | 166 | 134 | 111 | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | .028 | .061 | .113 | .189 | .430 | .819 | 1.39 | 2.18 | 4.57 | 8.25 | ..... | ..... | ..... | ..... | ..... |
| 7/32 in. | $w_1$ | ..... | ..... | ..... | 772 | 624 | 524 | 397 | 319 | 267 | 230 | 179 | 147 | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | ..... | 433 | 396 | 360 | 297 | 252 | 220 | 195 | 158 | 130 | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | ..... | .045 | .086 | .145 | .333 | .638 | 1.09 | 1.71 | 3.60 | 6.53 | ..... | ..... | ..... | ..... | ..... |
| No. 4 .225 in. | $w_1$ | ..... | ..... | ..... | 854 | 690 | 578 | 437 | 351 | 294 | 253 | 197 | 161 | ..... | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | ..... | 465 | 426 | 391 | 322 | 275 | 242 | 215 | 172 | 143 | ..... | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | ..... | .039 | .074 | .126 | .292 | .562 | .962 | 1.52 | 3.19 | 5.80 | ..... | ..... | ..... | ..... | ..... |
| No. 3 .244 in. | $w_1$ | ..... | ..... | ..... | 1129 | 905 | 754 | 570 | 455 | 379 | 325 | 253 | 205 | 174 | ..... | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | ..... | 564 | 529 | 484 | 407 | 349 | 303 | 267 | 219 | 183 | 158 | ..... | ..... | ..... | ..... |
| | $f$ | ..... | ..... | ..... | .025 | .049 | 085 | .200 | .389 | .671 | 1.06 | 2.25 | 4.11 | 6.78 | ..... | ..... | ..... | ..... |
| 1/4 in. | $w_1$ | ..... | ..... | ..... | 1226 | 982 | 818 | 613 | 490 | 409 | 350 | 272 | 223 | 189 | 163 | ..... | ..... | ..... |
| | $w_2$ | ..... | ..... | ..... | 597 | 562 | 518 | 437 | 376 | 327 | 287 | 235 | 197 | 170 | 149 | ..... | ..... | ..... |
| | $f$ | ..... | ..... | ..... | .022 | .043 | .075 | .178 | .348 | .601 | .954 | 2.03 | 3 70 | 6.11 | 9.39 | ..... | ..... | ..... |
| No. 2 .262 in. | $w_1$ | ..... | ..... | ..... | ..... | 1167 | 969 | 723 | 579 | 480 | 411 | 319 | 261 | 221 | 191 | 169 | ..... | ..... |
| | $w_2$ | ..... | ..... | ..... | ..... | 630 | 588 | 496 | 428 | 373 | 331 | 269 | 227 | 196 | 172 | 153 | ..... | ..... |
| | $f$ | ..... | ..... | ..... | ..... | .034 | .059 | .142 | .280 | .486 | .775 | 1.65 | 3.03 | 5.01 | 7.71 | 11.2 | ..... | ..... |
| No 1 .283 in. | $w_1$ | ..... | ..... | ..... | ..... | 1503 | 1243 | 921 | 732 | 607 | 519 | 402 | 328 | 277 | 240 | 211 | ..... | ..... |
| | $w_2$ | ..... | ..... | ..... | ..... | 755 | 715 | 615 | 533 | 467 | 415 | 338 | 285 | 245 | 216 | 192 | ..... | ..... |
| | $f$ | ..... | ..... | ..... | ..... | .022 | .040 | .098 | .195 | .342 | .549 | 1.18 | 2.17 | 3.61 | 5.58 | 8.13 | ..... | ..... |
| No 0 .306 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | 1640 | 1204 | 951 | 787 | 671 | 518 | 422 | 356 | 308 | 271 | 242 | ..... |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | 870 | 784 | 664 | 586 | 518 | 425 | 358 | 309 | 272 | 243 | 219 | ..... |
| | $f$ | ..... | ..... | ..... | ..... | ..... | .026 | .067 | .135 | .239 | .385 | .837 | 1.55 | 2.58 | 3.99 | 5.85 | 8.20 | ..... |
| 5/16 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | 1744 | 1279 | 1009 | 833 | 710 | 547 | 446 | 378 | 325 | 286 | 256 | ..... |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | 914 | 826 | 706 | 623 | 553 | 453 | 382 | 329 | 290 | 258 | 233 | ..... |
| | $f$ | ..... | ..... | ..... | ..... | ..... | .024 | .060 | .122 | .217 | .351 | .764 | 1.42 | 2.36 | 3.66 | 5 36 | 7.51 | ..... |
| 11/32 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | 1756 | 1379 | 1132 | 961 | 740 | 600 | 504 | 436 | 383 | 342 | 282 |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | 1036 | 920 | 814 | 727 | 597 | 504 | 437 | 382 | 344 | 310 | 261 |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | .037 | .077 | .138 | .226 | .499 | .934 | 1.57 | 2.43 | 3.58 | 5.03 | 9.01 |
| 3/8 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | 2361 | 1839 | 1503 | 1272 | 973 | 788 | 662 | 571 | 500 | 447 | 367 |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | 1264 | 1165 | 1037 | 931 | 766 | 650 | 563 | 494 | 444 | 401 | 336 |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | .023 | .050 | .091 | .151 | .337 | .636 | 1.07 | 1.67 | 2.47 | 3.48 | 6.26 |
| 13/32 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | 3121 | 2408 | 1955 | 1646 | 1255 | 1015 | 850 | 732 | 642 | 571 | 469 |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | 1564 | 1435 | 1290 | 1165 | 966 | 820 | 712 | 629 | 562 | 508 | 427 |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | .015 | .033 | .062 | .103 | .234 | .446 | .756 | 1.18 | 1.75 | 2 48 | 4.47 |
| 7/16 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 3094 | 2503 | 2102 | 1595 | 1280 | 1072 | 922 | 809 | 721 | 591 |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1740 | 1585 | 1436 | 1196 | 1019 | 884 | 776 | 696 | 649 | 535 |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .023 | .043 | .072 | .167 | .319 | .545 | .859 | 1.27 | 1.80 | 3.27 |
| 15/32 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 3913 | 3153 | 2632 | 1990 | 1595 | 1331 | 1144 | 1002 | 890 | 730 |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2059 | 1906 | 1735 | 1456 | 1236 | 1071 | 944 | 852 | 784 | 654 |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .016 | .030 | .051 | .120 | .233 | .400 | .633 | .941 | 1.34 | 2.43 |
| 1/2 in. | $w_1$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 4910 | 3922 | 3269 | 2450 | 1960 | 1634 | 1400 | 1225 | 1090 | 891 |
| | $w_2$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2389 | 2250 | 2070 | 1740 | 1493 | 1310 | 1154 | 1045 | 934 | 782 |
| | $f$ | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .011 | .022 | .037 | .089 | .174 | .300 | .477 | .712 | 1.01 | 1.85 |

fatigue, a maximum stress of 90,000 lb. per sq. in., when the spring is solid, is safe for all grades and sizes. With the smaller sizes of wire, 100,000 lb. per sq. in. may be used. In cases of severe impact or fatigue, stresses should be reduced to 80,000, or even 60,000 lb. per sq. in. These figures are for use with the customary design formula, which ignores correction for curvature. In case the Röver curvature correction factor is used, the allowable stresses just given may be increased by 20%. For special service, or materials, higher stresses may be used, but a reputable manufacturer then should be consulted.

Stresses for good quality brass wire may be taken at 40% of the above figures; for phosphor-bronze wire at 50%. Torsional modulus with steel may be taken at 11,000,000 to 11,500,000, if the design stress is conservative. For high stresses it will be lower.

Cold-coiled springs are coiled in multiple lengths, the ends are dressed, either automatically or by hand, and are then ground square, or have their end coils bent down flat. An allowance of 1.25 *d* should be made for the inactive height.

**COLD-COILED HELICAL EXTENSION SPRINGS** usually are coiled with initial tension, produced by partly overlapping the wire as it is fed into the coiling machine. A very considerable load must be applied before the initial tension is neutralized and the spring begins to elongate. The fundamental load formula is applicable, but, owing to the initial tension, extensions cannot be accurately predicted. Springs can be made without initial tension, and the same conditions then apply as for hot-wound extension springs.

Stresses for cold-coiled helical extension springs of steel wire, not subject to severe impact or fatigue, usually are limited to 80,000 lb. per sq. in. (96,000 to 100,000 lb. per sq. in. with Röver correction).

The ends of extension springs are commonly formed into hooks, loops, or some other shape, to permit the extensile load to be applied to the spring. See Fig. 2.

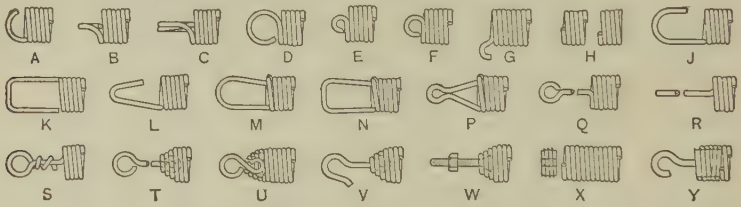


FIG. 2. Types of Spring Ends. *A*, Hook over center. *B*, Loop over center. *C*, Loop of two coils over center. *D*, Loop of one coil on side. *E*, Small eye on side. *F*, Small eye over center. *G*, Small hook on side. *H*, Plain ends—one end ground. *J*, Long hook. *K*, Square long loop. *L*, V-shaped hook. *M*, Knotted loop. *N*, Square knotted loop. *P*, Knotted eye. *Q*, Extended eye from center. *R*, Straight end from center; part of straight end usually annealed so that it can be twisted to attach to other parts or to form an eye, as shown in *S*. *T*, Extended swivel eye in tapered end spring; commonly employed when the ends must be safeguarded against breakage. *U*, Regular swivel eye in tapered end spring. *V*, Special-shaped swivel hook in tapered end spring. *W*, Swivel threaded bolt in tapered end spring. *X*, Plain end to screw on threaded plugs. *Y*, Plain end attached to special threaded plug.

**DOUBLE COIL SPRINGS,** consisting of two concentric helical compression springs, one inside the other, often are used. Triple coils, consisting of three concentric springs, also are common. A double coil spring, for a given design stress, is more flexible than a single coil of the same capacity and outside dimensions. Usually the concentric coils are of the same free height; but by making one of the springs (usually the inner) shorter than the other, a graduated effect, *i.e.*, a change in slope of the load-deflection curve, is produced. In double coils of common railroad sizes the clearance between the outer and inner springs should be $1/16$ in. all around. Thus in a spring 6 $1/4$ in. outside diameter, with an outer coil of 1 $1/4$-in. round bar, the inner coil should be 3 $5/8$ in. outside diameter. In very large or long springs, the clearance should be $3/32$ or $1/8$ in.

In a double coil spring properly designed there is a definite relation between the bar sizes of the outer and inner coils. If the design stress is the same for both coils, and if there is no inactive height, the index $D/d$ should be the same for both coils. Practically, since the inner coil always has the smaller bar diameter, its design stress can be somewhat greater, and its inactive height ($I = 1.25\,d$) somewhat less, than for the outer coil; hence its index can be a little the smaller of the two. Using commercial sizes of bars, this relation must be approximate in some cases, but should be adhered to as closely as possible.

**SQUARE WIRE.**—For helical compression or extension springs of square steel

$$P = \frac{0.4163\,Sb^3}{D - b} \quad \ldots\ldots\ldots\ldots [6]; \qquad f = \frac{5.57\,P(D - b)^3}{Gb^4} = \frac{2.32(D - b)^2 S}{Gb} \quad \ldots\ldots\ldots [7],$$

where $b$ = side of square bar, in.; $P$ = load corresponding to stress $S$; $D$ = outside diam. of spring, in.; $G$ = torsional modulus of elasticity; $f$ = deflection, in. per coil. If the Röver correction for curvature is to be included, the stress $S$ as given by these equations must be further multiplied by the curvature correction factor following formula [5].

When coiled into a spring a square bar upsets into a keystone section, with the larger side next to the coiling mandrel. The amount of this upset is difficult to predict, but a roughly approximate formula is: Upset = $0.3\,b^2/(D - 2b)$. See Fig. 3. Allowance must be made for this upset in figuring the solid height of the spring. Its effect upon stiffness and strength is relatively small, except in springs of very small spring index. Bars initially rolled to a keystone section sometimes are used, with the smaller side placed next to the coiling mandrel. In this way, after coiling, a square section is obtained.

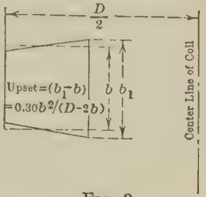


FIG. 3

For the same working stress, the safe load with square wire is 6% greater than with round wire of diameter equal to the side of the square. The deflection per pound of load is 70% as great; the deflection for a given unit stress is 74% as great for the square as for the round wire.

**RECTANGULAR SECTIONS.**—Helical compression or extension springs made of rectangular bars are not very common, but sometimes they will meet requirements that no round bar spring can fulfill. The torsional theory of rectangular bars is quite complex. The fundamental formulas for springs are:

$$P = \frac{\alpha a b^2 S}{R}. \quad . \quad . \quad [8]; \quad f = \frac{2\,\pi P R^3}{\gamma G a b^3}. \quad . \quad . \quad [9]$$

where $R$ = mean coil radius measured from center of coil to center of wire; $a$ = in all cases, the longer side, and $b$ = the shorter side of the rectangular bar, irrespective of whether the spring is made up with the bar coiled edgewise or flatwise; $f$ = deflection per coil; $\alpha$ and $\gamma$ are constants (see Table 4); $G$ = torsional modulus of elasticity.

Similar formulas for bars of elliptical section are as follows:

$$P = \frac{0.1963\, a b^2 S}{R}. \quad . \quad . \quad [10]; \quad f = \frac{2\,\pi P R^3}{\beta G a b^3}. \quad . \quad . \quad [11].$$

Values of $\beta$ are given in Table 4.

Table 4.—Values of Constants for Torsion in Rectangular and Elliptical Bars

| $a/b$ = | 1 | 1.25 | 1.5 | 1.75 | 2 | 2.5 | 3 | 4 | 10 | Infinite |
|---|---|---|---|---|---|---|---|---|---|---|
| $\alpha$ = | 0.208 | 0.221 | 0.231 | 0.239 | 0.246 | 0.258 | 0.267 | 0.282 | 0.312 | 0.333 |
| $\gamma$ = | .141 | .172 | .196 | .214 | .229 | .249 | .263 | .281 | .312 | .333 |
| $\beta$ = | .0982 | .1197 | .136 | .147 | .157 | .169 | .177 | .185 | .194 | .196 |

In many cases flattened wire, made by passing round wire through rolls, is used rather than truly rectangular wire. In these cases the moduli $\alpha$ and $\gamma$ may be multiplied by 0.85 to 0.95, depending on how nearly the shape approximates a true rectangle.

In rectangular sections, as in square, the upset of the bar in coiling must be considered in design. The allowable design stress in rectangular bar springs is greater than in rounds, and increases as the ratio $a/b$ increases. For large ratios it may be as high as 125,000 lb. per sq. in. This is because the stress is maximum at only two points in the section, *viz.*, the middle points of the long sides.

Correction for curvature in springs of rectangular or elliptical section is complicated by this same fact that maximum stress, as given by the above formulas, exists only at the middle point of the long sides, while the maximum stress increase due to curvature occurs at the midpoint of the side parallel to and nearest the axis. If the wire is coiled with the long side parallel to the axis, the stress must be multiplied by the same curvature correction factor as for round wire springs. See p. 10–06. If the bar is coiled with the long side perpendicular to the axis of the coil, the point of maximum stress in torsion comes at a point on the bar at which the curvature correction factor is unity, and if the bar is appreciably out of square the curvature correction factor can be neglected.

**CONICAL HELICAL SPRINGS.**—Conical springs have the following advantages: 1. By making the difference in diameter of successive coils equal to or slightly greater than the thickness of the wire, the coils may be made so to nest together as to give, for a given total extension, a small solid height as compared to that for cylindrical springs. 2. If made with uniform initial pitch between coils, the outside coils will bottom before the inner coils; the spring will show an increasing stiffness with load, rather than a constant stiffness. This curved load-deflection line may be desirable in special cases. 3. Conical springs have somewhat greater lateral stability than cylindrical springs. On the other hand, in conical springs as usually designed, much of the metal is only partially stressed; to carry a given load with a given deflection, from 50% to 100% more metal may be needed than in a cylindrical spring.

Volute springs, hot coiled from flat bar stock, are used to some extent in railroad work in Europe. They are not used in the U. S.

**The Safe Load on a Conical or Volute Spring** is the same as that on a cylindrical helical spring made of the same size wire or bar, and of a diameter equal to the maximum diameter of the cone. The stiffness depends on the number of coils in action at any one time, and on the minimum and maximum coil diameters. Deflection per pound of load at any instant $= (f/P) = (16\,n/Gd^4)\{(R_2^2 + R_1^2)(R_2 + R_1)\}$, where $R_1$, $R_2$ = respectively, minimum and maximum radii, measured from center of coil to center of wire; $n$ = number of coils; $d$ = wire diameter; $G$ = torsional modulus of elasticity.

## 3. TORSION SPRINGS

**HELICAL TORSION SPRINGS** act to transmit a twist instead of a direct pull. This twisting action may be used to: 1. Maintain a steady torque to hold an object in place, with only enough motion to compensate for wear or readjustment. 2. Store up energy while the spring is being coiled, and to restore the energy when the spring is uncoiled. 3. Absorb shocks when power is applied to rotate a shaft, or to act simply as a buffer.

**DESIGN FORMULAS.**—The stiffness of the spring, in terms of torque required to produce a given angle of twist, depends only on the material and the size of wire used, and on the length of wire. Number of coils, and within reasonable limits, the spacing between coils, have no effect except as they may determine the length of the wire.

Similarly, the maximum safe load in inch-pounds of torque that may be applied to the spring depends only on the size and material of the wire used. The length of coil has no influence. The radius of the coil is important only in that, if the spring is too sharply coiled, a minor correction must be applied to provide for the effect on strength of this too-sharp curvature.

For springs of round wire, the formulas used are as follows:

Torque or turning moment developed, in.-lb., is

$$M = \pi d^3 S/32\,k = d^3 S/10\,k \qquad [13];$$

$$\omega = \frac{1167\,ML}{E\,d^4} = \frac{114.6\,SL}{E\,d\,k} \qquad [14];$$

$$L = \frac{E\,d^4\,\omega}{1167\,M} \qquad [15].$$

For rectangular wire, these formulas become:

$$M = \frac{S\,b\,d^2}{6\,k} \qquad [16];$$

$$\omega = \frac{687.6\,ML}{E\,b\,d^3} = \frac{114.6\,SL}{E\,d\,k} \qquad [17];$$

$$L = \frac{E\,b\,d^3\,\omega}{687.6\,M} \qquad [18].$$

In these formulas, $b$ = breadth of rectangular wire, parallel to axis of coil; $d$ = diam. of wire, or thickness of rectangular wire, perpendicular to axis of coil; $L$ = effective length of wire in coiled portion of spring, plus a small allowance for length of wire in projection lugs; $\omega$ = angle turned through, deg.; $S$ = unit stress in wire; $E$ = modulus of elasticity in bending of spring material = 30,000,000 for steel; = 15,000,000 for phosphor-bronze or brass; $k$ = correction factor for effect of sharp curvature. See Table 5 for values of $k$ for different values of spring index, *i.e.*, ratio of mean diameter of coil to diameter of wire.

It has been general practice to omit the curvature correction factor $k$ from these formulas, tacitly caring for it by using a value of allowable stress which has been determined experimentally to work satisfactorily for springs of average spring index. An objection to this method is that an unnecessarily large amount of metal will be used for springs of large index, while springs of small index will be overstressed and may fail. More economical

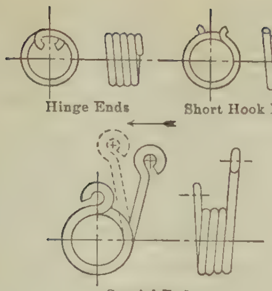


Fig. 4

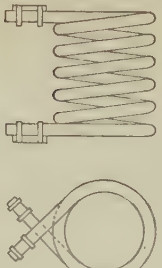


Fig. 5

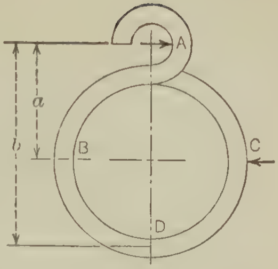


Fig. 6

results may be obtained by using an allowable stress value 8 to 10% larger, but inserting the proper curvature factor in the formulas.

Examples.—1. Design a helical torsion spring for a screen door, 30 in. wide, to develop a pull at the handle of 1 lb. with the door just opened, and of 3 lb. after the door has been turned through 180°. Use round wire with allowable unit stress of 150,000 lb. per sq. in. *Solution.*—Maximum turning moment developed = $3 \times 30 = 90$ in.-lb. From equation [13], $d^3/k = (10 \times 90)/150{,}000 = 0.006$. Assuming a spring index of 6, $k = 1.142$. Substituting and solving, $d = 0.218$. For a spring index of 6, outside diameter of coil will be $(6 + 1) \times 0.218 = 1.52$ in. From equation [15], $L = (30{,}000{,}000 \times 0.218^4 \times 180)/(1167 \times 90) = 116.1$ in. This length is equivalent to 28.4 coils of 1.30 in. mean diam.

2. Design a helical torsion spring of square wire for a window shade to develop a maximum pull of 3 lb. on the shade after 15 revolutions. Diameter of roller = 1.25 in. *Solution.*—Maximum turning moment developed = $3 \times 0.5 \times 1.25 = 1.875$ in.-lb. In this case, in the equation for rectangular wire, $b = d$ and $d^3/k = 6M/S = (6 \times 1.875)/150{,}000 = 0.000075$. As a first trial, let $k = 1$; then $d = 0.042$ in. Assuming the mean diameter of the spring inside the shade roller to be 0.625 in., the approximate value of the spring index will be $0.625/0.042 = 14.9$. From Table 5, the corresponding value of $k = 1.046$. Resubstituting $d^3 = 1.046 \times 0.000075$, $d = 0.0428$ in. $\omega = 15 \times 360 = 5400°$. $L = (30{,}000{,}000 \times 0.0428^4 \times 5400)/(687.6 \times 1.875) = 421.6$ in., equivalent to 215 coils, 0.625 in. mean diameter. Actually a slightly larger size wire, with correspondingly greater number of coils, might be used.

**DESIGN SUGGESTIONS.**—The following comments, from *The Mainspring* (Wallace Barnes Co.), will aid in the design of helical torsion springs:

1. While the spring as a whole is acting in torsion, the wire itself is being bent, not twisted. For manufacturing reasons, it may be cheaper and more expedient to make the spring of round wire. Inherently a square or a rectangular shape is better in bending than a round one; if space or weight limitations are important, square or rectangular wire often may be substituted to advantage. *a.* Flat wire, made by running round wire through flattening rolls, often is used instead of true rectangular wire. Due to its rounded corners, its stiffness and strength is 5 to 10% less than true rectangular wire, but it usually is easier to manufacture and handle. *b.* For a powerful torsion spring in small space, flat wire, coiled edgewise, may be advantageous.

2. The twisting action exerted by a torsion spring cannot be transmitted by means of a one-point contact, as the hook used with extension springs. A clamp or a firm two- or three-point contact is needed at each end of the spring. In simplest form, one of the necessary two points of contact may be furnished by a lug projecting inward or outward at the end of the coil, as in Fig. 4. A thimble placed within the coil and bearing against the inside of the wire at a point 90° away from the lug could furnish the other contact point. Alternatively, clamps may be used as in Fig. 5 to furnish these two points of contact at each end of the spring.

The shorter the lever arm between the lines of pressure through the points of contact, the greater the pressures must be to give the desired twisting moment. With stiff springs, lugs projecting outward rather than inward, will reduce the resulting bearing pressure exerted at the lug. If the end of the coil is supported externally rather than internally, the points of contact being at

**Table 5.—Curvature Correction Factors for Torsion Springs**

| Spring Index | Curvature Correction Factor $k$ | | Spring Index | Curvature Correction Factor $k$ | | Spring Index | Curvature Correction Factor $k$ | | Spring Index | Curvature Correction Factor $k$ | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Rectangular Wire | Circular Wire | | Rectangular Wire | Circular Wire | | Rectangular Wire | Circular Wire | | Rectangular Wire | Circular Wire |
| 3 | 1.286 | 1.332 | 6 | 1.124 | 1.142 | 10 | 1.072 | 1.080 | 16 | 1.043 | 1.049 |
| 4 | 1.200 | 1.229 | 7 | 1.105 | 1.119 | 12 | 1.057 | 1.066 | 20 | 1.036 | 1.039 |
| 5 | 1.153 | 1.175 | 8 | 1.091 | 1.103 | 14 | 1.049 | 1.056 | 40 | 1.017 | 1.019 |

*A* and *C*, Fig. 6, with 270° of coil between them, the useful torque is determined by the lever arm *a*; the stress in the wire at the point of contact is controlled by the greater lever arm *b*. The wire along the major portion of the length of the spring is no more heavily stressed when the external support is used than with the internal thimble; but with the external support, the stress at the one danger point is greater than the average stress in the ratio $b/a$. The useful load on the spring then must be reduced proportionately. This difficulty can be avoided by using an internal thimble, or by using clamps as in Fig. 5.

3. Always so place torsion springs that they will coil up under load. All cold-coiled helical springs have internal stresses as a result of the methods used in forming the spring. These stresses act to form a hidden safety factor, strengthening the spring against failure under a repetition of the same kind of deformation as originally caused the stresses but weakening it under stresses in the opposite direction. In general, if material in springs, or other types of construction, is to be heavily stressed in a direction opposite to that in which it was stressed in forming, the material should be heat-treated after shaping.

4. Long torsion springs, like long compression springs, but unlike extension springs, tend to be unstable laterally and to buckle on moderate provocation. This can be counteracted: *a*. By care in clamping or so holding the spring at the ends as to provide a slight degree of lateral support, and avoiding skew pressures that would tend to start buckling. Less desirable alternatives are: *b*. Providing a slight degree of lateral support along the length of the spring. *c*. Placing the spring under a small initial tension, tending to hold it straight.

5. As a torsion spring is loaded, its coil radius changes. Inside and outside clearances must be provided to permit these changes in radius. For example, a spring, 10 coils long, whose maximum twist is to be 0.5 revolution, will change by 0.5/10, or 5% in radius of coil. That much clearance, at least, must be present.

6. The longitudinal spacing between coils also changes. In the case just cited, initial spacing should be at least 5% of the wire diameter. A slight excess will prevent rubbing action. Unless frictional resistance is definitely desired, close-coiled springs should not be used.

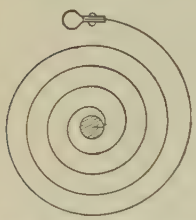


Fig. 7. Spiral or Power Spring

7. For best results, the spring should act in torsion only, with the center line of the coil coinciding with the axis of rotation. Sometimes clearance conditions prevent this, and the spring must be placed off-side. If this is done the spring will bend laterally as well as twist in operation. Successful results may be obtained under these conditions, but care is required, and lower stresses must be used.

**SPIRAL OR POWER SPRINGS,** Fig. 7, usually are made of thin cold-rolled strip steel. Under ideal conditions, the strip acts as a long curved beam subjected to a uniform bending moment throughout its entire length. The fundamental formulas are the same as for helical torsion springs of rectangular steel. The maximum torque exerted by the spring upon the central shaft is equal to the bending moment in the strip considered as a beam, and

$$M = S\,b\,h^2/6. \qquad [19]$$

Working stresses in this formula may run extremely high, 150,000 to 200,000, or even 250,000 lb. per sq. in. The higher values should be chosen only after consultation with a spring manufacturer. Rigorously, the curvature correction factor *k* should be introduced in this formula as in the equivalent formula for helical torsion springs. Practically, the radius of curvature is usually so large, compared to the thickness of the strip, that the correction can be neglected. The angle of twist $\theta$ in radians is given by the equation

$$\theta = ML/EI = 12\,ML/E\,b\,h^3. \qquad [20]$$

Dividing by $2\pi$, gives $\theta$ in terms of revolutions of the shaft. See notation on p. 10-14.

Practically, these formulas must be used with caution. It is somewhat difficult to so support the springs as to produce pure bending in the ends, rather than a combination of bending and tension. The presence of tension or shear in the metal at the ends of the spring causes a variable, rather than a constant, bending moment throughout the length of the spring. The metal then is overstressed in some places and understressed in others.

The relation between bending moment and angle of twist is even more influenced by practical conditions. If the spring shaft is turned through more than moderate angles the spring begins to coil solid on the arbor, and *L*, the working length of spring strip, gradually decreases. Also, strips in successive coils begin to touch, and the resulting friction materially affects the torque acting on the shaft. Fortunately, where power springs are to store energy, the demand usually is for the maintenance of a certain minimum torque over the maximum angle of rotation. With the spring fully wound up, available torque is much greater than the desired minimum value, and the large amount of friction then present is not particularly detrimental. When the spring is nearly run down, and torque has dropped almost to the minimum allowable value, the coils no longer touch and friction is much smaller. Friction may be reduced by lubricating the sides of the strip.

In spiral springs used in precision instruments, or for hair springs in watches or clocks, the problem is entirely different. Here the angle of twist is relatively small compared to the length of the spring. The coils are so supported that at no time do they come in contact.

Cold-rolled strip steel, properly heat treated, is capable of extremely high fiber stresses. In making spiral springs it is usual to cut off the proper length of strip, slot or form the ends as required, coil the strip, and apply it in the assembly without previous forming, except as the coiling itself serves to form the shape. Until the elastic limit is exceeded, the stress developed during forming may be computed from

$$S = Eh/2R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [21]$$

the strip being assumed to be changed from an initial straight condition to a radius of curvature $R$. If the strip has an initial radius curvature $R_1$, and is further bent to a radius $R_2$, the resulting change in stress is given by

$$S = \{(1/R_2) - (1/R_1)\}(Eh/2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [22]$$

Usually the arbor about which the strip is coiled is so small that the elastic limit is exceeded, and the strip is given a permanent radius of curvature. This introduces internal stresses which act to strengthen the material under further bending applied in the same direction as in forming, but to materially weaken it if bent in the reverse direction.

## 4. ELLIPTIC AND SEMI-ELLIPTIC SPRINGS

**NOTATION.**—$L$ = length, in., of spring between hanger or bolt centers, under working load; $n$ = number of plates in spring; $b$ = width of plates, in.; $h$ = thickness of plates, in.; $B$ = width of band, in.; $P$ = working load, lb. applied at center of spring; $S$ = maximum fiber stress in plates, at load $P$, lb. per sq. in.; $f$ = deflection, in., of spring under load $P$; $E$ = Young's modulus of elasticity.

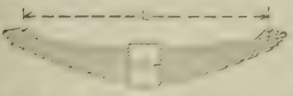


FIG. 8. Semi-elliptic Locomotive Spring

In the ideal case, the laminated semi-elliptic spring is considered as equivalent to a single flat plate supported at both ends and loaded in the middle, of uniform thickness $h$, and of width equal to $nb$ at the center and tapering regularly to zero at each end. The fundamental equations for this ideal case are:

$$P = 2Sbh^2n/\{3(L-B)\} \quad . . . [23]; \qquad f = 3PL^3/8Ebh^3n = SL^2/\{4E(L-B)h\} \quad . . . [24].$$

The case is supposed to be one of pure bending, direct shear being neglected.

In practice, a proportion of the plates are of length $L$ or longer. Hence the equivalent width of the single plate is not zero at the ends. This does not affect the load formula, but the deflection formula becomes

$$f = SKL^2/\{4E(L-B)h\}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

$$K = \{(1-r)(1-3r) - 2r^2 \log_e r\}/(1-r)^3, \quad . \quad . \quad . \quad . \quad . \quad [26]$$

where $r$ = (number of plates of length $L$)/(Total number of plates.) Values of $K$ are given in Table 6.

Table 6.—Values of $K$

| $r =$ | 0 | 0.05 | 0.10 | 0.15 | 0.20 | 0.25 | 0.30 | 0.40 | 0.50 |
|---|---|---|---|---|---|---|---|---|---|
| $K =$ | 1.000 | .950 | .927 | .901 | .876 | .855 | .836 | .804 | .776 |

Deflection formulas for elliptic springs are at best only approximate. It is customary for manufacturers to make up a trial spring before executing an order.

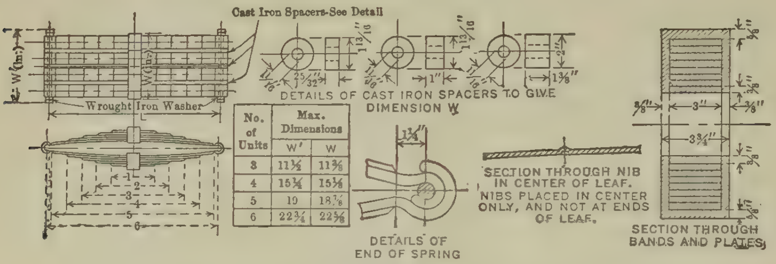
**For Automobile Springs,** former practice was to design for a static deflection, of 2 to 2 1/2 in., front axle, and 6 in. rear axle, the car containing no passengers. The present trend (1937) is toward a static deflection, both axles, of 6 in. unloaded, 7 1/2 to 9 in. loaded, with a maximum possible deflection of 12 in. This gives a car with approximately the same natural frequency, 80 to 85 cycles per min. for both front and rear springs. To obtain the higher flexibility for front springs, it has been necessary either to increase materially the size of the front leaf spring or springs, or (usually) to use helical springs for the front wheels. Design stresses run 65–85,000 lb. per sq. in., unloaded and 150,000 lb. per sq. in. maximum for leaf springs.

**For Semi-elliptic Locomotive Springs** the maximum fiber stress under working load (weight on spring with locomotive stationary on level track) is standard at 70,000 lb. per sq. in., on the assumption that the spring has adequate reserve movement. This requirement, under average working conditions, will be met if the length of spring and thickness of plates are related as shown in Table 7.

Table 7.—Length and Thickness of Semi-elliptic Locomotive Springs

| | | | | | |
|---|---|---|---|---|---|
| Length of spring, in. | 22–27 | 27–33 | 33–40 | 40–48 | 48–57 |
| Thickness, in., ideal | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 |
| Thickness, in., good practice | 1/4–5/16 | 9/32–3/8 | 5/16–7/16 | 3/8–1/2 | 7/16–9/16 |

**FULL ELLIPTIC SPRINGS** usually are so designed that they can be tested band to band. This insures against excessive deflection in service. For usual railroad service, if the maximum stress when springs are so tested does not exceed 125,000 lb. per sq. in., safe maximum stress at working load usually can be 80,000 to 85,000 lb. per sq. in. These stresses are for Grade 5 carbon steel (see Table 1). Alloy steel springs usually can be designed to somewhat higher stresses. Young's modulus of elasticity can be taken at 29,000,000 for all common grades of spring steel.


Cast Iron Spacers-See Detail
W'(in.)
W(in.)
Wrought Iron Washer
DETAILS OF CAST IRON SPACERS TO GIVE DIMENSION W
25/32"
1"
1⅜"
1¼"
SECTION THROUGH NIB IN CENTER OF LEAF. NIBS PLACED IN CENTER ONLY, AND NOT AT ENDS OF LEAF.
DETAILS OF END OF SPRING
3"
3¼"
SECTION THROUGH BANDS AND PLATES

| No. of Units | Max. Dimensions | |
|---|---|---|
| | W' | W |
| 3 | 11½ | 11⅜ |
| 4 | 15¼ | 15⅛ |
| 5 | 19 | 18⅞ |
| 6 | 22¾ | 22⅝ |

Fig. 9. Quadruple Elliptic Passenger Car Spring

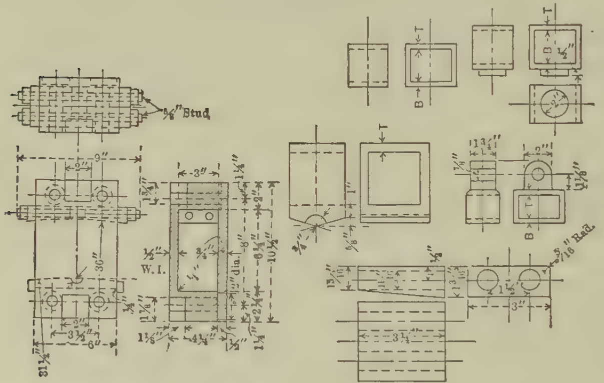


Fig. 10. Typical Bands

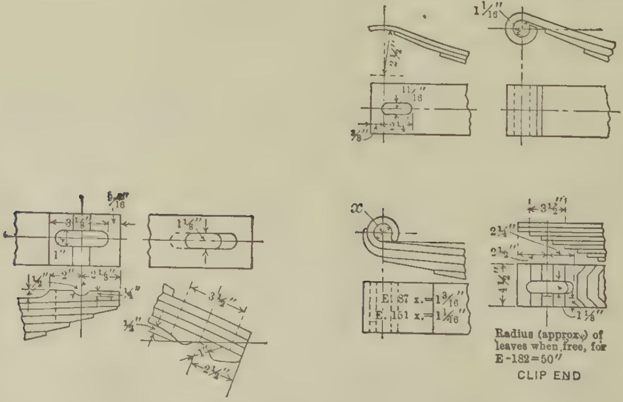


Fig. 11. Typical Elliptical Spring Ends

Fig. 8 shows a typical semi-elliptic locomotive spring. Fig. 9 is a typical quadruple-elliptic passenger car spring. In American practice all railroad elliptic springs are

" banded," or encircled with a cast steel or welded wrought iron strap. The latter has been most generally used, but cast steel bands are becoming more common. Fig. 10 shows typical styles of bands. For semi-elliptics the size of iron is usually from 3 × $^1/_2$ in. to 5 × $^3/_4$ in., depending on the size of spring. The " head " or side of the band running across the short plate, usually is heavier than the other three sides. It often bears a groove, a round dowel, a boss, or a hole, to position the spring on its seat.

The spring plates are " nibbed " at their centers, as shown in Fig. 9. This locks them together and into the band. The nib on the short plate is forced into the hot iron when the band is forged into place. Ends of the main plate on semi-elliptic springs are slotted, grooved, lipped, or turned into eyes, to provide engagement with the spring hangers. Fig. 11 shows typical ends, formed by welding steel pads on the ends of the main plate, or by attaching malleable iron end clips.

The bands of full elliptic railway springs usually are of 3 × $^3/_8$-in. iron on all four sides; sometimes a little heavier. Ends are either of eye-and-wrapper type, standard on passenger car elliptics, or are lipped to fit into end castings. The latter are known as the sectional type, and are standard for tender springs.

## 5. FLAT SPRINGS

**FLAT SPRINGS** is a term used to cover springs of a wide range of shapes, which are alike only in that they have all been made from originally flat strips. Simple flat springs are included, but the term also includes springs which have been twisted or notched out, bent or dished, each to meet special uses. Due to the wide range of shapes it is difficult to give exact design equations but the following statement of general principles taken from *The Mainspring* (Wallace Barnes Co) will be useful.

1. For highest efficiency, proportion each part of the spring to the load it must carry, no stronger and no weaker. Excess metal in the flexible part of a spring not only is useless in itself; it is undesirable, as it reduces flexibility, and so may result in a reduced, rather than an increased, safety factor. A tapering cantilever section is much more efficient than a plain rectangular strip.

2. As far as possible, place the metal so that it acts in bending rather than in torsion. A round rod is well adapted to resist twisting. Thin strip metal, no matter how dished or otherwise shaped, is weak and uneconomical under forces which tend to twist rather than bend it.

3. For parts of the spring where flexibility is desired, use a simple flat rectangular section. Added strength or stiffness, desired at any one point, can be obtained by crimping the metal longitudinally, but at a sacrifice in efficiency in absorbing energy.

4. Similarly, cupping the metal is an efficient method of obtaining local strength and stiffness at the expense of resilience.

5. Sharp bends are undesirable, both for manufacturing reasons and because of the serious localized stresses introduced in the material. A radius of curvature two or more times the thickness of the metal is advisable. Table 8 gives curvature stress factors.

**Table 8.—Curvature Stress Factor for Flat Springs**

$t$ = thickness of metal

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Inside Radius of Bend........ | 0.5$t$ | 0.75$t$ | 1.00$t$ | 1.25$t$ | 1.5$t$ | 2$t$ | 3$t$ | 4$t$ |
| Curvature Stress Factor...... | 1.523 | 1.370 | 1.286 | 1.236 | 1.200 | 1.153 | 1.105 | 1.060 |

6. For simpler cases of design, where a more or less straight rectangular strip can be used, the customary beam formulas for strength and deflection give satisfactory results. As deflection from a straight line becomes greater, or as bends are introduced (see Fig. 12-A) modifications become necessary. *a.* Correct values for strength can be obtained in all cases by using as $L$ in the beam formula the perpendicular distance from the point where stress is desired to the line of application

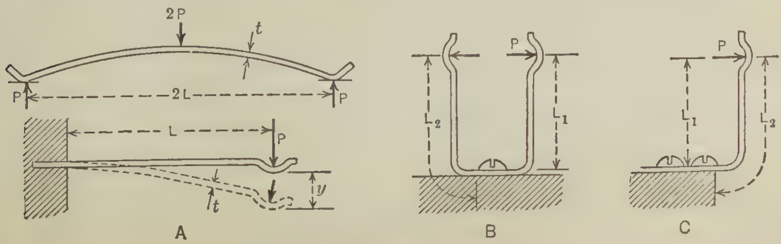


Fig. 12. Flat Springs

of the load. *b*. Correct values for flexibility can be approximated by using as $L$ a value intermediate between the length of the strip and this perpendicular distance, using judgment as to the exact value to use.

For a given maximum required spring deflection and given material, a definite relation exists between effective spring length and thickness of metal which gives the best results. If greater thickness of metal is used than this relation calls for, the metal will be overstressed and will fail. If a smaller thickness is used, the width of the spring must be increased by a far greater ratio than the thickness if reduced, in order to obtain sufficient stiffness. Where feasible, therefore, the best relation between spring length and thickness of metal should be approximated.

**DESIGN FORMULAS.**—The usual cantilever beam formulas may be used:

$$\text{Required section modulus} = b\,t^2/6 = PL/S \quad [27]$$

$$\text{Deflection} = PL^3/3\,EI = 2\,SL^2/3\,E\,t \quad [28]$$

where $b$ = width of spring; $t$ = thickness of spring; $L$ = effective cantilever length of spring; all in inches; $S$ = allowable stress = 150,000 lb. per sq. in. in good grade spring stock; $P$ = load on spring, lb.; $E$ = modulus of elasticity = 30,000,000 for steel; $I$ = moment of inertia. Using $S$ = 150,000 and $E$ = 30,000,000, equations [27] and [28] become

$$P/b = 25{,}000\,t^2/L \quad [29]; \qquad y = PL^3/7{,}500{,}000\,b\,t^3 = L^2/300\,t, \quad [30]$$

whence

$$\text{Recommended thickness} = L^2/300\,y, \quad [31]$$

where $y$ = maximum deflection, in. For a spring supported at both ends and loaded in the middle, $L = \frac{1}{2}$ (effective length between end supports); each arm carries $\frac{1}{2}$ full load.

## 6. VIBRATION IN SPRINGS

**LONGITUDINAL OR LATERAL VIBRATIONS** of helical, or lateral vibrations of flat or leaf springs carrying an attached load.—The natural frequency is given by

$$\text{Frequency} = 3.12\sqrt{k/(W_L + C_1\,W_S)} \text{ cycles per second,}$$

where $k$ = stiffness of the spring, lb. per in. of deflection; $W_S$ = weight of spring itself, and $W_L$ = weight of load carried by spring, lb.; $C_1$ = constant; for helical springs in extension or compression $C_1 = \frac{1}{3}$; for cantilever springs of uniform cross section $C_1 = \frac{1}{4}$; for semi-elliptic springs of uniform cross section with load at center, $C_1 = \frac{1}{2}$. For leaf springs the proportion of the weight of the spring which must be taken as acting with the load depends on the number of master leaves and on other variables. In a cantilever leaf spring it would be but a small proportion, possibly one-eighth.

Usually the weight of the attached load is so great, as compared to the weight of the spring, that spring weight can be ignored in the equation. The equation is inexact if the attached load equals or is less than the weight of the spring itself. Dash pots or other damping devices act to slow down this frequency slightly, but usually the change is not over 1 to 5%.

**TORSIONAL VIBRATIONS** in spiral springs or torsion helical springs.—Frequency = $1.245\sqrt{T/I}$ vibrations per second, where $T$ = stiffness of spring, in.-lb. of torque per revolution; $I$ = mass moment of inertia of the rotating body, lb.-in.$^2$, including one-third of the mass moment of inertia of the spring, in addition to the inertia of the attached mass.

**TRAVELING WAVES** moving back and forth within the helical spring itself.—Frequency = $8.85\sqrt{k/W_S}$, where $k$ and $W_S$ have the same meaning as before.

In valve springs this type of vibration has a frequency of 15,000 to 25,000 vibrations per min. If frequency is an exact multiple of operating speed, resonance may occur. In work done both in the U. S. and Great Britain, resonances have been found up to as high as the 25th harmonic. Maximum effects usually occur below the 12th harmonic. This spring surge usually is accompanied by poor valve seating, resulting in noise and loss of power. It also has caused many valve spring breakages, the fracture usually occurring at between one and two coils from the end of the spring. The exact shape of the cam greatly influences the magnitude of these waves for any given harmonic. Re-design of the spring, change of cam contour, or change of weight of valve mechanism may be necessary to overcome the difficulty.

## 7. ENDURANCE OF SPRINGS

Extensive research has been made on the endurance of springs and spring steels under repeated stress. Fatigue tests of hot-rolled spring steel (Grade 5), tested in a cycle of torsional stress between zero and a maximum, resulted in the following indications: The endurance limit, or maximum stress at which steel will withstand the test indefinitely, is close to the proportional limit in torsion, *viz.*, about 85,000 lb. per sq. in. Preliminary results from fatigue tests on full-size railroad type helical springs suggest that a somewhat lower fiber stress, possibly 75,000 lb. per sq. in. (with the Röver correction) more accurately represents the endurance limit when in spring form. Further tests are needed to confirm or disprove this tentative result.

In standard fatigue tests in bending, stress usually is completely reversed; *i.e.*, the cycle is between positive and negative stresses of the same numerical value. The endurance limit for Grade 5 steel under these conditions is about 100,000 lb. per sq. in. The ultimate strength of this steel, heat treated to a troostitic condition, is about 190,000 lb. per sq. in. According to the Goodman diagram the endurance limit for a zero-maximum-stress cycle would be about 130,000 lb. per sq. in. Therefore, an elliptic spring in which the maximum stress never exceeds the latter figure should last indefinitely. It is difficult accurately to apply the criterion to an actual case. In practical manufacturing, unavoidable departures from the ideal spring of theory occur, which may give rise to serious local variations in maximum stress, notably in the short plates.

It is well known that surface defects, such as pits, seams, guide marks, decarburization, etc., even if slight, have a marked effect on endurance limit. This is particularly true of the comparatively thin sections used in elliptics. Ground and polished specimens show endurance limits very much higher than those from an ordinary hot-rolled finish.

In large hot-coiled springs, surface defects probably are not as important as in an elliptic spring; conversely, they are harder to control. A helical spring cannot be ground or polished economically after coiling; the best that can be done is to use centerless ground and polished bars, and guard carefully against decarburization throughout the subsequent operations of fabrication and heat treatment.

## 8. COMPARATIVE EFFICIENCY OF SPRINGS

The primary function of a spring is to store energy, and later return it when needed. The efficiency of different types of springs, therefore, may be compared on the basis of the amounts of energy which they will store per cubic inch or per pound of metal in the spring. Conversely, the potential energy storage per cubic inch of metal in a given type of spring may be used to estimate the volume or weight of spring needed to perform a given task.

**THE MODULUS OF RESILIENCE,** or amount of energy that may be stored in 1 cu. in. of metal in a spring, is given by the equations

$$M_r = S^2/mG \text{ or } M_r = S^2/mE \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [32]$$

where $M_r$ = modulus of resilience; $S$ = allowable unit stress, lb. per sq. in.; $G$ = modulus of elasticity in shear; $E$ = modulus of elasticity in tension or compression. For values of $G$ and $E$ see Table 1. Values of $m$ for different types of springs are as follows:

| Shape of Spring | Relative Efficiency | $m$ | |
|---|---|---|---|
| Helical spring, extension or compression service, round wire | 50% | 4 | A |
| Square wire | 30.8% | 6.5 | |
| Straight wire used in tension | 100% | 2 | B |
| Helical spring used in torsion, round wire | 25% | 8 | |
| Square or rectangular wire | 33% | 6 | |
| Spiral power spring, round wire | 25% | 8 | |
| Flat wire | 33% | 6 | |
| Leaf spring | up to 33% | 6+ | |
| Flat spring, uniform cross section, concentrated load | 8 1/3% | 24 | |

A—Metal in shear; Modulus of Resilience = $S^2/mG$
B—Metal in tension; Modulus of Resilience = $S^2/mE$

If correction factor for curvature $k$ is to be included

$$M_r = S^2/k^2mG \text{ or } M_r = S^2/k^2mE \quad . \; . \; . \; . \; . \; . \; . \; . \quad [33]$$

For values of $k$, see pages 10–06 and 10–15.

To find the volume of active metal needed in any spring, multiply the maximum deflection by 1/2 the maximum load, and divide by the modulus of resilience for that type of spring. The result should be increased by 5 to 10% or more, to allow for the inactive or partly inactive metal in end coils or loops. By introducing a proper correction factor results can be obtained in terms of weight of spring, rather than in terms of cubic inches of material.

Examples.—1. Music wire compression spring to show maximum compression of 0.5 in. under a load of 10 lb. Unit stress, 80,000 lb. per sq. in. *Solution.*—Here $m = 4$ and $G = 12{,}000{,}000$.

$M_r$ = (80,000 × 80,000)/(4 × 12,000,000) = 133 in.-lb. per cu. in. Volume of metal needed = 0.5 × 5/133 = 0.0188 cu. in. per spring, or 0.020 cu. in. after a reasonable allowance for inactive material has been added. At 0.284 lb. per cu. in. for steel, the springs will weigh 5.68 lb. per thousand.

2. Steel torsion helical spring to develop torque of 100 in.-lb. for an angle of twist of 2 revolutions. Use round wire and unit stress of 120,000 lb. per sq. in.

*Solution.*—$M_r$ = (120,000 × 120,000)/(8 × 30,000,000) = 60 in.-lb. per cu. in. Angle of twist in radians = $4\pi$. Volume of active metal needed = $4\pi \times 50/60$ = 10.46 cu. in. net, or 11 cu. in. allowing for inactive metal. Weight per spring = 11 × 0.284 = 3.12 lb.

In each of these cases many alternative designs using different wire and coil diameters are possible, but for all of these designs the comparative efficiency is about the same. Approximately the same net weight of material will be required. The following, however, should be noted:

1. It is often commercially advisable to use stock sizes of wire, and round dimensions for coil diameter, even if these sizes result in lowered unit stresses and greater weight of metal. 2. The efficiency factors tabulated above assume the springs to be properly detailed at the ends. If load is improperly applied, the spring may be severely over-stressed locally, compelling reductions in working loads, and, hence, in the energy-storing capacity of the spring. 3. Judgment must be exercised in estimating the percentage to be allowed for inactive metal.

# Section 11

# ROTATING MEMBERS

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# ROTATING MEMBERS

## AXLES AND SHAFTS

### 1. AXLES

(See also Shafting, p. 24–03)

The terms, axle, shaft and spindle, are applied to rotating or oscillating machine parts whose motion is constrained by journals or bearings. Such members may be classified according to the predominating stress, or according to the particular purpose for which they are used as: *a*. Axles, loaded transversely and subject principally to bending, as axles of railroad cars. *b*. Shafts subjected to torsion or to combined torsion and bending, as transmission shafting or the shafts of engines. *c*. Spindles, or short shafts which carry a tool for actually doing work or which carry the work itself, and in consequence must have accurate motion. Examples are shafts or spindles of machine tools.

**MATERIALS OF AXLES AND SHAFTS.**—Bessemer steel, open-hearth steel and alloy steel are commonly used. While Bessemer steel is extensively used, open-hearth steel usually is specified for shafts for armatures and engines, and for machine tool spindles and shafts operating under similar conditions. Alloy steel is used where great strength is required. Nickel-steel, because of its ability to withstand heavy shocks and torsional stresses, is well adapted for axles and shafts. See pp. 2–21 and 2–40 for specifications of suitable steels.

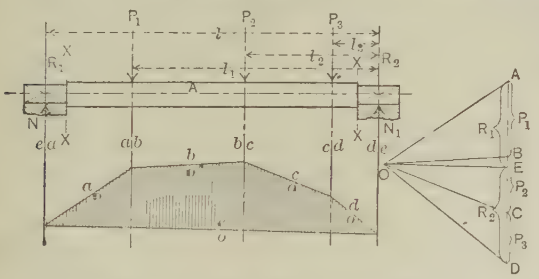


Fig. 1. Bending Moment of a Shaft

**AXLE DESIGN.**—The graphical method is best adapted to the solution of axle problems, as it gives the moments at all sections, and shows at once the location of the *maximum* bending moment. Fig. 1 shows a typical case.

Axle *A* carries the loads $P_1$, $P_2$, $P_3$. Torsional moment due to friction is neglected. The axle is considered as supported at the center of the bearings $N$ and $N_1$. The dangerous section, unless the shaft is very short, will be where the bending moment is a maximum.

In Fig. 1, denote the forces $P_1$, $P_2$, etc., as *ab*, *bc*, *cd*, etc., and draw the corresponding force diagram as shown, making $AB = P_1$, $BC = P_2$, $CD = P_3$ to any convenient scale. These forces are drawn consecutively downward, as they act in that direction; their sum *AD*, must equal the sum of the reactions, or vertical forces. From any pole, *O* draw *OA*, *OB*, *OC*, and *OD*. From any point on *ab*, in the space diagram, draw *oa*, and *ob*, parallel respectively to *OA* and *OB* in the force diagram. From the intersection of *ob* and *bc* draw *oc*, parallel to *OC*, and similarly draw *od*. Join the intersection of *oa* and *ea* with the intersection of *od* and *de*, thus locating the closing string *oe*. Draw *OE* parallel to *oe*, locating *E*. Then in the force diagram $DE = R_2$, $EA = R_1$ to the assumed scale of the force diagram.

The vertical ordinates of the space polygon are proportional to the bending moments at the point considered. The numerical value of any bending moment is the continued product of the length of the ordinate, the perpendicular distance of *O* from *AD*, the *reciprocal* of the scale of the space diagram, and the *reciprocal* of the scale of the force diagram. Thus, if a given ordinate is 2 in., the pole distance 2 $^1/_2$ in., the space scale 1 $^1/_2$ in. = 1 ft., or $^1/_8$ size, and 1 in. = 5000 lb. on the force diagram; then the bending moment at the point considered, $M = 2 \times 2\,^1/_2 \times 8 \times 5000 = 200{,}000$ in.-lb. From this moment the diameter of the shaft may be computed.

If the shaft is short, resulting in a small section to withstand the applied bending moment, the section at $XX$ should be checked for shearing. Also it is important that all forces acting be considered, and that the maximum value of each be selected. It has been found (Machine Design, Smith and Marx) that the force due to vertical oscillation caused by jar in running is about 40% of the static load for car axles. Hence, the axle should be designed for 1.4 × the static load. In addition, there is in car axles, a bending moment due to curves, switches, and wind pressures. This may amount, at a maximum, to the equivalent of a horizontal force, of 40% of the static load, applied at a height of 6 ft. from the rail. Also, due to the reversal of stress, which causes the fibers to be subjected alternately to tension and compression, the allowable unit stress is correspondingly reduced. For data on standard locomotive and tender axles see Kent's Mechanical Engineers' Handbook—Power, pp. 14–25 to 14–27, forming Vol. 2 of this series.

## 2. SHAFTS

The American Standards Assoc. has issued a code for the design of transmission shafting, in which all of the commonly used theories are discussed, and a rational procedure for the design of shafts under all conditions of loading is developed. The following notes are abstracted from the code:

**Notation.**—$A$ = area of cross-section, sq. in.; $\alpha$ = ratio of maximum to average intensity of stress, resulting from axial loading only; $D$ = outside diameter of shaft, in.; $d$ = inside diameter of hollow shaft, in.; $E$ = modulus of elasticity, approximately 30,000,000; $F$ = axial tensile or compressive load, lb.; $K = d/D$ = ratio of inside diameter to outside diameter of hollow shafts; $K_m$ = shock and fatigue factor to be applied to the computed bending moment; $K_t$ = shock and fatigue factor to be applied to the computed torsional moment; $k$ = radius of gyration of transverse cross-section of the shaft, in.; $l$ = length of shaft between centers of supports, in.; $M$ = maximum bending moment, in.-lb.; $N$ = rev. per min.; $P$ = maximum power transmitted by the shaft, horsepower; $p_t$ = maximum intensity of shearing stress in shaft, lb. per sq. in. = 8000 for commercial shafting without allowance for keyways = 6000 with allowance for keyways *; $S$ = maximum flexural stress, either tension or compression, lb. per sq. in. = 16,000 for commercial shafting without allowance for keyways = 12,000 with allowance for keyways *; $S_s$ = maximum torsional shearing stress, lb. per sq. in. = 8000 for commercial shafting without allowance for keyways = 6000 with allowance for keyways *; $T$ = maximum torsional moment, in.-lb.; $V$ = transverse shear, lb.; $W$ = transverse load supported by shaft, lb.

SOLID SHAFTS.—The formulas recommended differ from the formulas heretofore in general use in that they introduce factors $K_m$ and $K_t$ for shock and fatigue. These factors vary with the nature of the load. Values are given in Table 1.

**Shafts in Pure Torsion.**—

$$K_t\,T = 0.1963\,S_s\,D^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

$$D = \sqrt[3]{321{,}000\,K_t\,P/S_s\,N} \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

**Shafts Subjected to Bending.**—

$$D = \sqrt[3]{32 K_m\,M/\pi\,S} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

**Shafts Subjected to Combined Torsion and Bending.**—

$$D = \sqrt[3]{(16/\pi\,p_t)\sqrt{(K_m\,M)^2 + (K_t\,T)^2}} \quad . \quad . \quad . \quad . \quad [4]$$

If the diameter to transmit a given horsepower is desired, it may be found by substituting for the term $(K_t\,T)^2$ in equation [4] its value in terms of horsepower, as follows:

$$(K_t\,T)^2 = (396{,}000\,K_t\,P/2\pi\,N)^2 \quad . \quad . \quad . \quad . \quad . \quad [5]$$

**Shafts Subjected to Torsion, Flexure and Axial Loading.**—This is the general case, and all other shaft formulas may be derived from it by the omission of the factors that do not apply to the case under consideration.

$$D = \sqrt[3]{\frac{16}{\pi p_t}\sqrt{\left(K_m\,M + \frac{\alpha\,F\,D\,(1 + K^2)}{8}\right)^2 + (K_t\,T)^2}} \times \frac{1}{\sqrt[3]{1 - K^4}} \quad . \quad . \quad [6]$$

To determine the diameter of a shaft in terms of horsepower transmitted, substitute in equation [6] the value of $(K_t\,T)^2$ as found by equation [5].

**Shafts in Bending over Short Spans with Heavy Transverse Shear.**—

$$D = \sqrt{16\,V/3\,\pi\,S_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

---

* $S$ may be taken as 60% of the elastic limit in tension, but not more than 36% of the ultimate tensile strength of steel purchased under definite physical specifications. Similar values for $S_s$ and $p_t$ are 30% of elastic limit and 18% of ultimate strength.

**Table 1.—Shock and Fatigue Factors for Shafts**

| | $K_m$ | $K_t$ |
|---|---|---|
| **Stationary Shafts** | | |
| Gradually applied loads | 1.0 | 1.0 |
| Suddenly applied loads | 1.5–2.0 | 1.5–2.0 |
| **Rotating Shafts** | | |
| Gradually applied or steady loads | 1.5 | 1.0 |
| Suddenly applied loads, minor shocks only | 1.5–2.0 | 1.0–1.5 |
| Suddenly applied loads, heavy shocks | 2.0–3.0 | 1.5–3.0 |

**Shafts Subjected to Combined Torsion and Heavy Transverse Shear.—**

$$p_t = (16\,T/\pi\,D^3) + (16\,V/3\,\pi\,D^2) \quad . \; . \; . \; . \; . \; . \; . \quad [8]$$

**Diagrams for Solution of Shafting Problems.**—All of the equations given above except equations [7] and [8] can best be solved by the diagrams Figs. 2 and 3. In these diagrams, the heavy arcs represent standard transmission shafting sizes, and the lighter arcs the standard machinery shafting sizes. The scale of abscissas represents values of $K_m\,T$; the scale of ordinates at the right represents values of $K_t\,T$; the scale of ordinates at the left represents the values of $K_t\,P$, where $P$ = horsepower at 100 r.p.m. The diagrams are applicable to steady, fluctuating, reversing, and gradually or suddenly applied loads, with or without minor or heavy shocks, provided that correct values of $K_m$ and $K_t$ are used. They are not applicable to shafts in bending over short spans with heavy transverse shear, or to shafts subjected to combined torsion and heavy transverse shear.

The diagrams are based on a maximum permissible shearing stress of 6000 lb. per sq. in., but can be used for other stresses by multiplying the shaft diameter found from the diagram by an appropriate factor as follows:

| | | | | | | |
|---|---|---|---|---|---|---|
| Shearing stress, lb. per sq. in. | 4000 | 5000 | 6000 | 7000 | 8000 | 9000 |
| Factor | 1.14 | 1.06 | 1.00 | 0.95 | 0.91 | 0.87 |

EXAMPLE 1.—Required the size of shaft to withstand a maximum bending moment $M$ of 9350 lb., and a maximum torsional stress $T$ of 9600 lb., the allowable unit shearing stress being 8000 lb. per sq. in., and the load suddenly applied with minor shock. From Table 1, values of $K_m$ and $K_t$ may be taken as 1.5 and 1.25 respectively. Then $K_m\,M = 1.5 \times 9350 = 13{,}975$ and $K_t\,T = 9600 \times 1.25 = 12{,}000$. In Fig. 2, the intersection of $K_m\,M = 13{,}975$, and of $K_t\,T = 12{,}000$ lies on the arc representing a shaft 2 $1/2$ in. diameter, with a unit stress of $p_t$ of 6000 lb. per sq. in. Since the allowable unit torsional stress in the present case is 8000 lb., the correct diameter for the conditions given is found by applying the factor for the higher stress. $2.5 \times 0.91 = 2.275$. The next higher standard size of shaft is 2 $5/16$ in.

EXAMPLE 2.—Required the size of an alloy steel shaft to transmit 210 Hp. at 150 r.p.m., and at the same time sustain a bending moment of 90,000 in.-lb. The unit allowable shearing stress may be taken as 9000 lb. per sq. in. At 150 r.p.m., 210 Hp. is equivalent to $210 \times (100/150) = 140$ Hp. at 100 r.p.m. The intersection of 140 Hp. and $K_m\,M = 90{,}000$ in Fig. 3 shows that a shaft 4 $3/4$ in. diameter would be required at a unit shearing stress of 6000 lb. per sq. in. Applying the factor 0.87 for the higher unit stress, $4.75 \times 0.87 = 4.1325$ in. The next larger standard shaft is 4 $1/4$ in. diameter.

EXAMPLE 3.—Required the size of a non-rotating shaft to support a steady load of 4000 lb., the supports being spaced 3 ft. apart. From the equation for simple beams (see p. 7–22), $M = Wl/4 = 4000 \times (3 \times 12)/4 = 48{,}000$ in.-lb. From Table 1, $K_m = 1$ and $K_m M = 48{,}000$. Since there is no rotation $T = 0$, and $K_t\,T = 0$. In Fig. 3, on the abscissa of $K_t\,T = 0$, at a value of $K_m\,M = 48{,}000$, the nearest size of shaft that will be suitable at a unit shearing stress of 6000 lb. per sq. in., is 3 $1/2$ in.

**HOLLOW SHAFTS.**—All of the foregoing equations are applicable to hollow shafts, when multiplied by a factor equivalent to $(1/\sqrt[3]{1 - K^4})$, in which $K = d/D$, that is, the ratio of the inside and outside diameters. Values of $(1/\sqrt[3]{1 - K^4})$ are as follows:

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $K$ | 0.95 | 0.90 | 0.85 | 0.80 | 0.75 | 0.70 | 0.65 | 0.60 | 0.55 | 0.50 |
| $\sqrt[3]{1 - K^4}$ | 0.57 | 0.70 | 0.78 | 0.84 | 0.88 | 0.91 | 0.93 | 0.95 | 0.97 | 0.98 |
| $1/\sqrt[3]{1 - K^4}$ | 1.75 | 1.43 | 1.28 | 1.19 | 1.14 | 1.10 | 1.07 | 1.05 | 1.03 | 1.02 |

The procedure is to determine the diameter of a solid shaft for the given conditions, and to then multiply the diameter so found by the factor corresponding to the desired value of $d/D$.

EXAMPLE 4.—Determine the size of a hollow shaft to be used under the conditions given in Example 1, assuming the value of $K$ to 0.70. As determined in Example 1, the diameter of a solid shaft required would be 2.275 in. The factor for $K = 0.70$ is 1.10. Therefore the outside diameter of an equivalent hollow shaft would be $2.275 \times 1.10 = 2.250$ in. The inside diameter would be $2.25 \times 0.70 = 1.75$ in.

**SHAFTS UNDER AXIAL LOADING.**—The maximum intensity of stress resulting from a force $F$ applied axially is $S = \alpha\,F/A$, where $\alpha$ = ratio of maximum to average

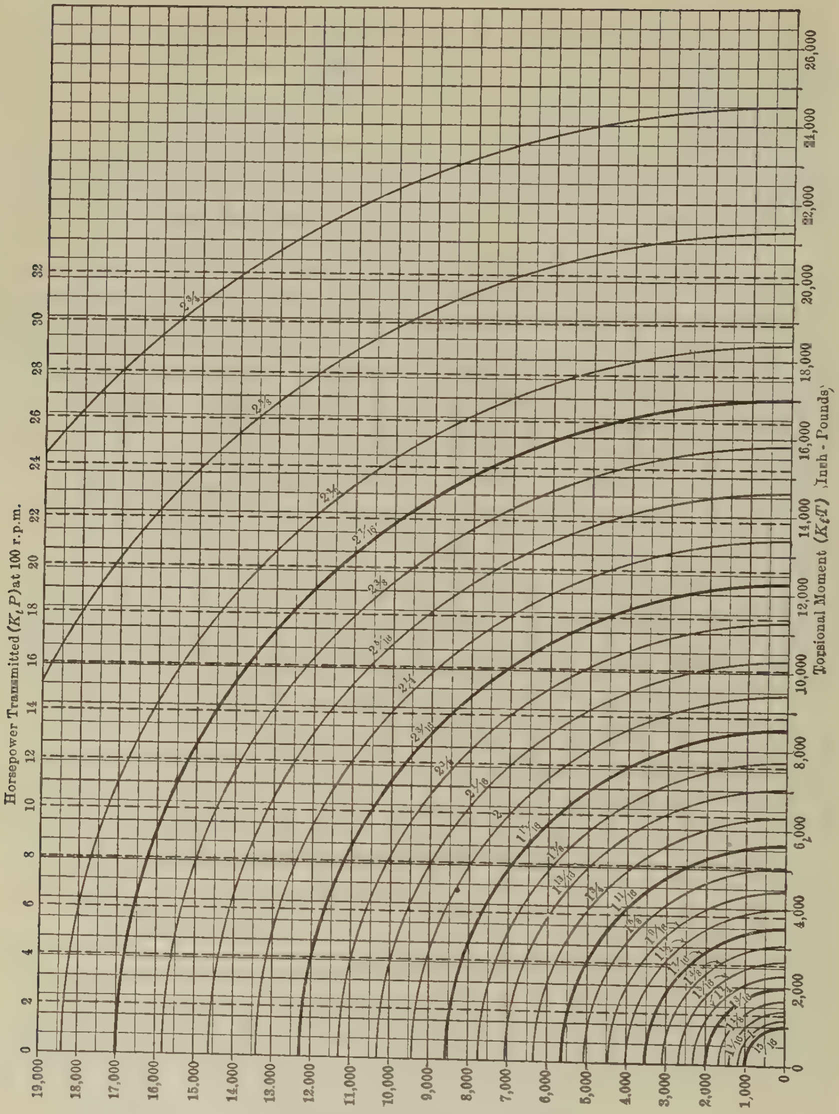


Fig. 2. Shaft Selection Chart

intensity of stress resulting from the axial load only, and $A$ = cross-sectional area of the shaft, sq. in. If the shaft is rigidly supported at short intervals and virtually acts as a column under compressive loads, $\alpha = 1$. Under all tensile axial loads $\alpha$ also is 1. If the shaft is not rigidly supported, it acts as a long column, and $\alpha$ should be computed by one of the long column formulas. Values of $\alpha$, for ratios of $l/k$ up to 115, computed according to the formula

$$\alpha = 1/\{1 - 0.0044\ (l/k)\} \qquad [9]$$

are given below. For ratios of $l/k$ greater than 115, use Euler's equation (see p 7–35).

| | | | | | | |
|---|---|---|---|---|---|---|
| Ratio $l/k$ .................. | 0 | 25 | 50 | 75 | 100 | 115 |
| $\alpha$ ......................... | 1.00 | 1.12 | 1.28 | 1.49 | 1.78 | 2.02 |

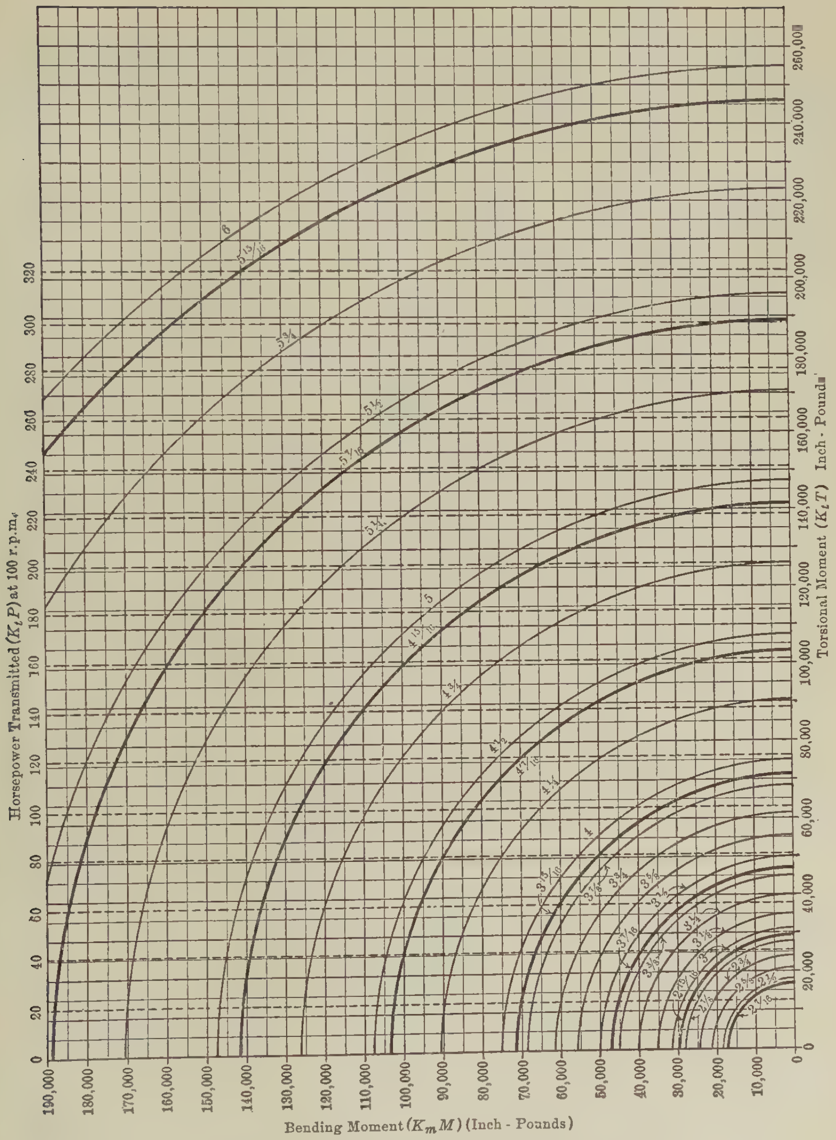


FIG. 3. Shaft Selection Chart

The value of $\alpha$ so determined then should be used in equation [6] to determine the size of shaft requied.

**SHAFTS UNDER ALTERNATING STRESS.**—Shafts subjected to torsion and bending encounter reversal of stress at each revolution, and should be calculated by means of equation [4]. The value to be ascribed to the shock and fatigue factors, $K_t$ and $K_m$, which take into account the stress reversals, depend on the conditions under which the shaft operates. No general statement can be made concerning the values to be used, since the shock factor, in particular, depends on the resiliency of the shaft system as a whole. For complete stress reversal $K_t$ and $K_m$ should be 1.5.

## Table 2.—Critical Speed Equations

NOTATION.—$N_1$, $N_2$, $N_3$ = critical speeds, r.p.m.; $\Delta_1$ $\Delta_2$ = static defection, in., at $W_1$ and $W_2$ (shaft horizontal); $d$ = diam. of shaft, in.; $I$ = moment of inertia; $E$ = 29,000,000. Weights taken in pounds; vertical shafts considered as horizontal.

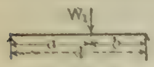
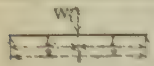
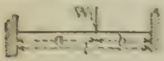
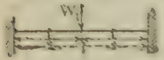
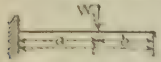
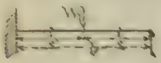
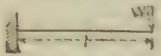
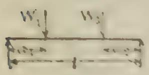
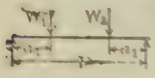
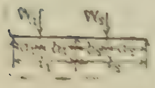
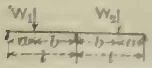
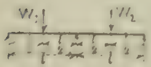
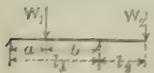
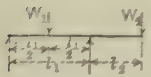
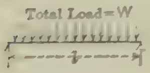
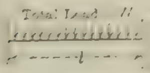
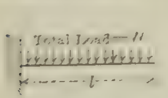
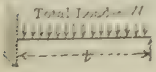
| | | |
|---|---|---|
| Single Concentrated Load | General Formula: $N_1 = \frac{187.7}{\sqrt{\Delta_1}}$. | [10] |
|  | $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}} = 387{,}000\frac{d^2}{ab}\sqrt{\frac{l}{W_1}};\ \Delta_1 = \frac{W_1 a^2 b^2}{3EIl}$. | [11] |
|  | $N_1 = 1{,}550{,}500\, d^2\sqrt{\frac{1}{W_1 l^3}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{W_1 l^3}{48EI}$. | [12] |
|  | $N_1 = 387{,}000\, d^2\sqrt{\frac{l^3}{W_1 a^3 b^3}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{W_1 a^3 b^3}{3EIl^3}$. | [13] |
|  | $N_1 = 3{,}100{,}850\, d^2\sqrt{\frac{1}{W_1 l^3}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{W_1 l^3}{192EI}$. | [14] |
|  | $N_1 = 775{,}200\frac{d^2}{ab}\sqrt{\frac{l}{W_1 a(3l+b)}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{W_1 a^3 b^2}{12EIl^3}(3l+b)$. | [15] |
|  | $N_1 = 2{,}337{,}000\frac{d^2}{l}\sqrt{\frac{1}{W_1 l}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{7}{768}\frac{W_1 l^3}{EI}$. | [16] |
|  | $N_1 = 387{,}000\, d^2\sqrt{\frac{1}{W_1 l^3}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{W_1 l^3}{3EI}$. | [17] |
| Two Concentrated Loads | General Formula:<br>$\left.\begin{matrix}N_1\\N_2\end{matrix}\right\} = 132.7\sqrt{\left(\frac{K_1}{W_1}+\frac{K_2}{W_2}\right) \pm \sqrt{\left(\frac{K_1}{W_1}-\frac{K_2}{W_2}\right)^2+\frac{4K_3^2}{W_1 W_2}}}$. | [18] |
|  | $C = \frac{6EI}{(l-a_1-a_2)^2[l(3l-2a_1-2a_2)-(a_2-a_1)^2]}$.<br>$K_1 = C\frac{2}{a_1^3}l(l-a_2)^2;\ K_2 = C\frac{2}{a_2^3}l(l-a_1)^2$.<br>$K_3 = C\frac{l(l^2-a_2^2-a_1^2)-a_1a_2(a_1-a_2)}{a_1a_2}$.<br>$N_1$, $N_2$, use general formula. | [19] |
|  | $N_1 = 548{,}400\frac{d^2}{a_1(3l-2a_1)}\sqrt{\frac{l}{W_1}}$;<br>$N_2 = 548{,}400\frac{d^2}{a_1}\sqrt{\frac{1}{W_1(3l-4a_1)}} = 187.7\sqrt{\frac{1}{\Delta_1}};\ \Delta_1 = \frac{W_1 a_1^2}{6EI}(3l-4a_1)$. | [20] |
|  | $C = \frac{3EI}{4l_1l_2(l_1+l_2)-l_1(l_2+a_2)^2-l_2(l_1+a_1)^2}$.<br>$K_1 = \left(\frac{l_1}{a_1b_1}\right)^2 C[4l_2(l_1+l_2)-(l_2+a_2)^2]$;<br>$K_2 = \left(\frac{l_2}{a_2b_2}\right)^2 C[4l_1(l_1+l_2)-(l_1+a_1)^2]$;<br>$K_3 = \frac{l_1l_2}{a_1b_1a_2b_2}C(l_1+a_1)(l_2+a_2)$;<br>$N_1$, $N_2$, use general formula. | [21] |

**Table 2.—Critical Speed Equations—*Continued***

| | | |
|---|---|---|
| Two Concentrated Loads (*Continued*) | | |
|  | $C = \frac{3EIl}{2a^2b^2[4l^2 - (l+a)^2]}$;<br>$K_1 = K_2 = C[8l^2 - (l+a)^2]$; $K_3 = C(l+a)^2$.<br>$N_1$, $N_2$, use general formula | [22] |
|  | $N_1 = 1{,}547{,}000\,d^2\sqrt{\frac{1}{W_1 l^3}} = 124.3\sqrt{\frac{1}{\Delta_1}}$.<br>$N_2 = 2{,}337{,}000\,d^2\sqrt{\frac{1}{W_1 l^3}} = 187.7\sqrt{\frac{1}{\Delta_1}}$; $\Delta_1 = \frac{7}{768}\sqrt{\frac{Wl^3}{EI}}$. | [23] |
|  | $C = \frac{12\,EI}{a^2l_2^2[4b^2l_1(l_1+l_2) - (l_1^2 - a^2)^2]}$.<br>$K_1 = Cl_1^2b^2(l_1+l_2)$; $K_2 = Ca^2b^2l_1$; $K_3 = 1/2\,Cal_1l_2(l_1^2 - a^2)$.<br>$N_1$, $N_2$, use general formula<br>$\Delta_1 = \frac{W_1a^2b^2}{3EIl_1} - \frac{W_2al_2}{6EIl_1}(l_1^2 - a^2)$;<br>$\Delta_2 = \frac{W_2l_2^2}{3EI}(l_1+l_2) - W_1\frac{al_2}{6EIl_1}(l_1^2 - a^2)$. | [24] |
|  | $C = \frac{3EI}{l_1^3l_2^2(l_1+l_2) - 9/16\,l_1^4l_2^2}$;<br>$K_1 = 16\,Cl_2^2(l_1+l_2)$; $K_2 = Cl_1^3$; $K_3 = 3\,Cl_1^2l_2$;<br>$N_1$, $N_2$, use general formula<br>$\Delta_1 = \frac{W_1l_1^3}{48\,EI} - \frac{W_2l_2l_1^2}{16\,EI}$; $\Delta_2 = \frac{W_2l_2^2(l_1+l_2)}{3\,EI} - \frac{W_1l_2l_1^2}{16\,EI}$. | [25] |
| Distributed Loads | $\Delta$ = maximum static deflection | |
|  | $N_1 = 2{,}232{,}510\,d^2\sqrt{\frac{1}{Wl^3}} = 211.4\sqrt{\frac{1}{\Delta}}$;<br>$N_1$ (shaft alone) $= 4{,}760{,}000\,d/l^2$, $N = [1, 4, 9, 16, \text{etc.}]N_1$.<br>$\Delta = \frac{5}{384}\frac{Wl^3}{EI}$. | [26] |
|  | $N_1 = 4{,}979{,}250\,d^2\sqrt{\frac{1}{Wl^3}} = 245\sqrt{\frac{1}{\Delta}}$;<br>$N_1$ (shaft alone) $= 10{,}616{,}740\,d/l^2$; $N = [1, 2.78, 5.45, 9, \text{etc.}]N_1$;<br>$\Delta = (Wl^3)/(384EI)$. | [27] |
|  | $N_1 = 795{,}196\,d^2\sqrt{\frac{1}{Wl^3}} = 167.6\sqrt{\frac{1}{\Delta}}$;<br>$N_1$ (shaft alone) $= 1{,}695{,}514\,d/l^2$; $N = [1, 6.34, 17.6, 43.6, \text{etc.}]N_1$.<br>$\Delta = (Wl^3)/(8EI)$. | [28] |
|  | $N_1 = 3{,}482{,}715\,d^2\sqrt{\frac{1}{Wl^3}} = 209.7\sqrt{\frac{1}{\Delta}}$;<br>$N_1$ (shaft alone) $= 7{,}021{,}600\,d/l^2$; $N = [1, 3.24, 6.8, 11.6, \text{etc.}]N_1$;<br>$\Delta = (Wl^3)/(185EI)$. | [29] |

## 3. CRITICAL SPEED OF SHAFTS

The center of mass of a rotating shaft never lies in its mechanical axis of rotation; hence, in a horizontal or inclined shaft, even if additional loads are neglected, there is a deflection caused by the weight of the shaft itself, which causes the center of mass to lie below the true axis of rotation. When rotation begins the shaft remains bent downward, but as the speed increases the centrifugal force increases until a state of equilibrium is established at what is called the critical speed. At this speed the centrifugal force is just sufficient to counteract the force tending to deflect the shaft downward, and the shaft rotates in a bowed form, and it is said to be in a "whirl." As vertical shafts cannot be in perfect balance the same applies to them. At the critical speed the maximum vibration of the revolving mass, and consequently of its support, occur, and for this reason, determination of critical speed is of prime importance in design. Above the critical speed the center of mass revolves inside the bow of the shaft, the tendency being to cause the rotating mass to revolve about its own center of gravity and not about the mechanical axis. This condition is met by providing special bearings which will accommodate themselves to the conditions, or by providing a flexible shaft which is just strong enough to withstand the deflection stresses in passing through the critical speed.

This refers to the first or lowest critical speed, $N_1$. There is a series of secondary critical speeds, $N$, of higher value with diminishing amplitudes of vibration. See paper by S. H. Weaver in *Trans.* A.S.M.E., 1910, from which Table 2, p. 11–08, is taken.

In addition to the equations in Table 2, equation [30] (E. A. Lof, *Machy.*, Feb., 1909) is useful for a multiple-loaded shaft,

$$N_1 = \frac{N_a N_b N_c}{\sqrt{N_b^2 N_c^2 + N_c^2 N_a^2 + N_a^2 N_b^2}} \quad . \; . \; . \; . \; . \; . \; . \quad [30]$$

where $N_1$ = first critical speed of the system, $N_a$ = first critical speed of unloaded shaft, $N_b$, $N_c$ = first critical speed of each separate load.

## CRANKS AND CRANK-PINS

**OVERHUNG CRANKS.**—Works on machine design give the proportions shown in Figs. 1 and 2 for cast-iron and forged steel cranks. While the drawings show different methods of securing the pin to the crank, the best practice is to use either press or shrink fits, both for the pin and for attaching the crank to the shaft.

Let $F$ = greatest load on the crank-pin, when crank and connecting-rod are at right angles; $S$ = maximum allowable stress in the crank, lb. per sq. in.; $L$ = distance from center of pin to section at which stress is to be computed, in. Then dimensions may be checked for bending action at any section, as C-D, Fig. 2, by the following equations:

$$\text{Bending Moment} = FL = t\,W^2\,S/6 \quad . \; . \; . \; . \; . \; . \; . \quad [1]$$

$$t = 6\,FL/W^2 S \quad . \; . \; . \quad [2]; \quad W = \sqrt{6\,FL/t\,S} \quad . \; . \; . \; . \; . \; . \quad [3]$$

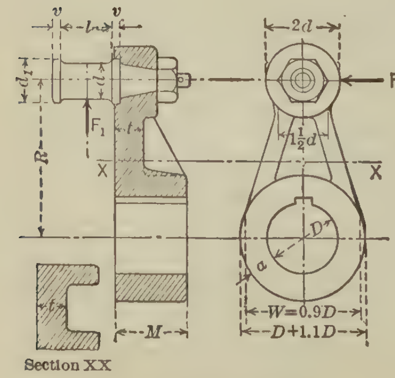


Fig. 1. Cast-iron Crank

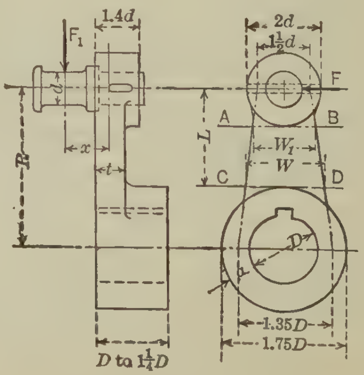


Fig. 2. Forged Steel Crank

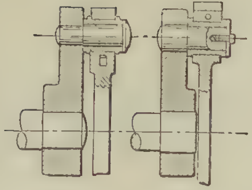


FIG. 3 FIG. 4
Methods of Reducing Crank Overhang

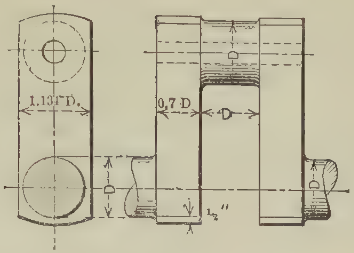


FIG. 5. Forged Crank-shaft

For cast-iron cranks $t$ may be taken as 0.6 $D$. The diameter of the crank-pin bosses $d_1$ may be taken as $d_1 = 1.16\,d + 0.02$ in. to $1.24\,d + 0.02$ in. The thickness of the bosses, $v$, may be taken as $0.75\,(d_1 - d)$.

Let $F_1$ = greatest force acting on the crank-pin when the crank is on dead centers. The bending moment at the section A–B, Fig. 2, then is

$$F_1\,x = W_1\,t^2\,S/6 \qquad [4]$$

$$W_1 = 6\,F_1\,x/t^2\,S \qquad [5]; \qquad t = \sqrt{6F_1\,x/W_1\,S} \qquad [6]$$

For steel cranks, $S$ may be taken as 4500 to 5500 lb. per sq. in.

The thickness of metal in the hub or boss of the crank (dimension $a$, Fig. 1), varies with the length of the boss. Values of $a$ given by Seaton are as follows, $D$ being the diameter of the crank-shaft:

| $M$ = | 1.0 $D$ | 0.9 $D$ | 0.8 $D$ | 0.7 $D$ |
|---|---|---|---|---|
| $a$, iron = | 0.35 $D$ | 0.38 $D$ | 0.40 $D$ | 0.41 $D$ |
| $a$, steel = | 0.30 $D$ | 0.32 $D$ | 0.33 $D$ | 0.34 $D$ |

**FORGED CRANK-SHAFTS.**—Low and Bevis (Manual of Machine Drawing and Design) give the dimensions shown in Fig. 5, based on a large number of examples chiefly taken from marine practice. Sometimes a hole $^1/_3$ the diameter of the pin is drilled through the crank-pin as shown.

**CRANK-DISCS,** Fig. 6, are often used in small engines which run at high speed. The weight $W$, opposite the crank-pin is for the purpose of balancing the crank-pin, and the part of the connecting-rod which revolves with it. The following proportions may be used in designing crank-discs: $D_1 = D$ to $1.2\,D$; $B = 0.7\,D$ to $D$; $A = 0.9\,B$; $C = 0.25\,D$; $E = 0.4\,D$; $F = 2d$; $H = d$ to $1.5\,d$; $L = B$ to $1.25\,B$.

**BALANCED CRANKS.**—Fig. 7 shows a method of balancing cranks. When the size of the weight is determined by experiment the counterweight must be detachable as shown in Fig. 8. Fig. 9 shows a case where the center of mass of the balance weight does not fall opposite the crank-pin.

**STRENGTH AND PROPORTIONS OF OVERHUNG CRANK-PINS,** Fig. 10.—The dimensions of crank-pins are based on the strength required to resist all the forces to which it will be subjected, and on the intensity of pressure permissible between pin and journal. This pressure must be kept low enough to avoid forcing lubricant out of the bearing, in which event heating and excessive wear would occur. Bending action on the pin is $^1/_2\,F\,L$, where $F$ = load on the pin, lb., and $L$ = length of pin, in. If $S$ = allowable maximum stress, lb. per sq. in., and $d$ = diameter of pin, in., the resisting moment $M_r$, in.-lb., is $M_r = \pi\,d^3\,S/32$. Hence

$$FL = d^3\,S/5.1 \qquad [7]$$

and

$$d = 1.721\,\sqrt[3]{F\,L/S} \qquad [8]$$

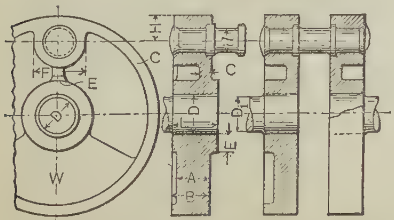


FIG. 6. Crank-discs

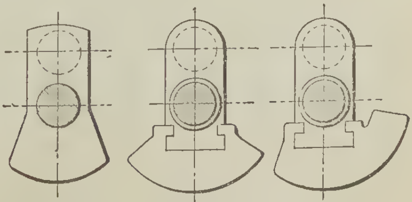


FIG. 7 FIG. 8 FIG. 9
Methods of Balancing Cranks

The stress in the pin is alternately compression and tension. The value of $S$, therefore should be taken as not over 5500 for steel. With this value of $S$, for a given ratio of $L/d = J$, equation [8] becomes

$$d = \sqrt[4]{J\,F/107.9} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

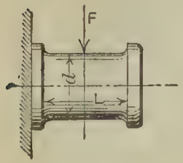


FIG. 10. Crank-pin

To avoid heating, pins usually are made of greater dimensions than are called for by equation [9]. Heating depends on the bearing pressure, which in turn depends on the projected area of the bearing, that is, on $d \times L$. If $P$ be the bearing pressure, lb. per sq. in., $F = P\,L\,d$, and $L = F/P\,d$. Substituting this value in equation [8], we have

$$d = \sqrt[4]{5.1\,F^2/PS} \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

or, for a value of $S = 5500$,

$$d = \sqrt[4]{F^2/107.9\,P} \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

For data on allowable bearing pressures see pp. 13–31 to 13–33.

**FORCES ACTING ON CRANKS.—Tangential Forces.**—In engines, if the center-line of the crank-shaft lies in the same plane as center of the piston and crosshead, the effective force of the crank acts in a line tangent to the crank circle at any given position of the crank. This effective force is

$$F_t = P \sec \beta \sin (\beta + \alpha) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

where $F_t$ = effective tangential force on the crank-pin, lb.; $P$ = total force acting on the

**Table 1.—Factors for Determining Tangential Effort**

Factor = $\sec \beta \sin (\beta + \alpha)$

Forward stroke is toward crank-shaft. Velocity of crosshead = velocity of crank-pin × tabular value

| Crank Angle, deg., Forward Stroke | Ratio (Length of connecting-rod/length of crank) | | | | | | | | | | | Crank Angle, deg., Return Stroke |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 7 | 8 | |
| 5 | 0.13061 | 0.12191 | 0.11611 | 0.11197 | 0.10887 | 0.10645 | 0.10452 | 0.10295 | 0.10162 | 0.09956 | 0.09801 | 355 |
| 10 | .25948 | .24222 | .23075 | .22257 | .21644 | .21168 | .20787 | .20476 | .20216 | .19808 | .19503 | 350 |
| 15 | .38488 | .35936 | .34246 | .33044 | .32145 | .31447 | .30889 | .30432 | .30052 | .29456 | .29008 | 345 |
| 20 | .50512 | .47180 | .44986 | .43429 | .42266 | .41366 | .40644 | .40055 | .39567 | .38799 | .38223 | 340 |
| 25 | .61855 | .57807 | .55158 | .53286 | .51891 | .50811 | .49949 | .49247 | .48661 | .47743 | .47056 | 335 |
| 30 | .72361 | .67678 | .64639 | .62500 | .60911 | .59683 | .58704 | .57905 | .57242 | .56202 | .55423 | 330 |
| 35 | .81880 | .76659 | .73313 | .70966 | .69226 | .67884 | .66817 | .65947 | .65224 | .64092 | .63232 | 325 |
| 40 | .90278 | .84660 | .81082 | .78591 | .76751 | .75334 | .74209 | .73293 | .72533 | .71343 | .70454 | 320 |
| 45 | .97437 | .91562 | .87861 | .85291 | .83411 | .81961 | .80812 | .79878 | .79102 | .77890 | .76985 | 315 |
| 50 | 1.03257 | .97296 | .93581 | .91023 | .89147 | .87709 | .86570 | .85645 | .84879 | .83681 | .82788 | 310 |
| 55 | 1.07666 | 1.01807 | .98196 | .95723 | .93915 | .92534 | .91441 | .90554 | .89820 | .88674 | .87819 | 305 |
| 60 | 1.10622 | 1.05066 | 1.01678 | .99371 | .97691 | .96408 | .95396 | .94575 | .93896 | .92836 | .82047 | 300 |
| 65 | 1.12114 | 1.07070 | 1.04024 | 1.01961 | 1.00462 | .99320 | .98420 | .97691 | .97089 | .96147 | .95450 | 295 |
| 70 | 1.12174 | 1.07842 | 1.05250 | 1.03502 | 1.02235 | 1.01272 | 1.00514 | .99900 | .99393 | .98603 | .98015 | 290 |
| 75 | 1.10868 | 1.07434 | 1.05395 | 1.04024 | 1.03033 | 1.02281 | 1.01689 | 1.01210 | 1.00814 | 1.00199 | .99741 | 285 |
| 80 | 1.08305 | 1.05923 | 1.04516 | 1.03573 | 1.02890 | 1.02375 | 1.01969 | 1.01641 | 1.01370 | 1.00948 | 1.00635 | 280 |
| 85 | 1.04626 | 1.03406 | 1.02688 | 1.02207 | 1.01861 | 1.01598 | 1.01391 | 1.01225 | 1.01087 | 1.00873 | 1.00713 | 275 |
| 90 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 270 |
| 95 | 0.94613 | 0.95833 | 0.96551 | 0.97032 | 0.97378 | 0.97641 | 0.97848 | 0.98014 | 0.98152 | 0.98366 | 0.98526 | 265 |
| 100 | .88657 | .91039 | .92446 | .93389 | .94070 | .94586 | .94994 | .95320 | .95591 | .96013 | .96327 | 260 |
| 105 | .82317 | .85751 | .87790 | .89161 | .90152 | .90904 | .91497 | .91975 | .92371 | .92987 | .93444 | 255 |
| 110 | .75765 | .80096 | .82688 | .84437 | .85703 | .86666 | .87425 | .88038 | .88546 | .89336 | .89924 | 250 |
| 115 | .69147 | .74192 | .77238 | .79301 | .80800 | .81941 | .82841 | .83570 | .84173 | .85113 | .85812 | 245 |
| 120 | .62583 | .68139 | .71527 | .73834 | .75514 | .76797 | .77809 | .78630 | .79309 | .80369 | .81158 | 240 |
| 125 | .56164 | .62023 | .65635 | .68107 | .69915 | .71297 | .72391 | .73276 | .74011 | .75157 | .76011 | 235 |
| 130 | .49952 | .55913 | .59628 | .62186 | .64062 | .65500 | .66639 | .67563 | .68330 | .69528 | .70421 | 230 |
| 135 | .43985 | .49859 | .53561 | .56124 | .58010 | .59460 | .60609 | .61544 | .62319 | .63531 | .64436 | 225 |
| 140 | .38279 | .43897 | .47475 | .49966 | .51804 | .53223 | .54348 | .55264 | .56025 | .57214 | .58104 | 220 |
| 145 | .32835 | .38048 | .41402 | .43740 | .45489 | .46831 | .47898 | .48751 | .49490 | .50623 | .51470 | 215 |
| 150 | .27639 | .32322 | .35361 | .37500 | .39089 | .40317 | .41296 | .42096 | .42758 | .43798 | .44577 | 210 |
| 155 | .22668 | .26717 | .29366 | .31238 | .32632 | .33712 | .34574 | .35277 | .35863 | .36780 | .37467 | 205 |
| 160 | .17892 | .21224 | .23418 | .24975 | .26138 | .27038 | .27759 | .28347 | .28837 | .29606 | .30181 | 200 |
| 165 | .13276 | .15828 | .17517 | .18719 | .19619 | .20317 | .20875 | .21331 | .21712 | .22308 | .22756 | 195 |
| 170 | .08782 | .10508 | .11655 | .12472 | .13086 | .13562 | .13943 | .14254 | .14514 | .14921 | .15223 | 190 |
| 175 | .04370 | .05241 | .05820 | .06234 | .06544 | .06786 | .06979 | .07137 | .07268 | .07475 | .07630 | 185 |
| 180 | .00000 | .00000 | .00000 | .00000 | .00000 | .00000 | .00000 | .00000 | .00000 | .00000 | .00000 | 180 |

piston, lb. = area of piston, sq. in. × pressure on piston, lb. per sq. in.; $\alpha$ = angle between crank and the line between center of crank and center of piston; $\beta$ = angle between connecting-rod and the line between center of crank and center of piston. Table 1 gives values of sec $\beta$ sin($\beta + \alpha$) for various ratios of $l/r$, where $l$ = length of connecting-rod, in., and $r$ = radius of crank, in. If the crank is used to impart motion to a crosshead, as in a punch press, the resistance encountered by the crosshead, *i.e.*, the resistance of the metal to shearing or deformation, is taken as the value of $P$. The tangential force to be applied to the crank then can be computed, from which the horsepower required is determined. Force acting along the center line of the connecting-rod is

$$F_c = \frac{P}{\cos\beta} \qquad [13]$$

The force acting on the guides is

$$F_g = P \tan\beta \qquad [14]$$

Notation as before.

**Inertia Forces.**—The velocity of the crosshead or piston at a given position of the crank, when center line of piston and crank-shaft are in the same plane, is, approximately

$$V_p = V_c\,(\sin\alpha + r \sin 2\alpha/2\,l) \qquad [15]$$

where $V_p$, $V_c$ = respectively, velocity of piston and crank-pin, ft. per sec.; other notation as before.

The acceleration of the reciprocating parts is

$$a = V_c^2\,(\cos\alpha \pm r\cos 2\alpha/l)r \qquad [16]$$

and the inertia forces, lb. per sq. in. of piston, is

$$F_i = (WV_c^2/g)\,(\cos\alpha \pm r\cos 2\alpha/l)/Ar \qquad [17]$$

In these equations $a$ = linear acceleration, ft. per sec. per sec.; $W$ = total weight of

**Table 2.—Inertia Factors**

Values of $\cos\alpha \pm (r/l)\cos 2\alpha$. Algebraic signs relate to forward stroke; use opposite signs for return stroke

| Crank Angle, Forward Stroke, deg. | Ratio (Length of connecting-rod/length of crank) | | | | | | | | | | | Crank Angle, Return Stroke, deg. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 7 | 8 | |
| 0 | 1.5000 | 1.4000 | 1.3333 | 1.2857 | 1.2500 | 1.2222 | 1.2000 | 1.1818 | 1.1667 | 1.1429 | 1.1250 | 360 |
| 5 | 1.4886 | 1.3901 | 1.3244 | 1.2776 | 1.2424 | 1.2148 | 1.1932 | 1.1752 | 1.1604 | 1.1369 | 1.1193 | 355 |
| 10 | 1.4547 | 1.3607 | 1.2980 | 1.2533 | 1.2197 | 1.1934 | 1.1728 | 1.1556 | 1.1415 | 1.1191 | 1.1022 | 350 |
| 15 | 1.3990 | 1.3123 | 1.2546 | 1.2134 | 1.1824 | 1.1582 | 1.1391 | 1.1234 | 1.1103 | 1.0897 | 1.0742 | 345 |
| 20 | 1.3227 | 1.2461 | 1.1950 | 1.1586 | 1.1312 | 1.1098 | 1.0929 | 1.0790 | 1.0674 | 1.0492 | 1.0355 | 340 |
| 25 | 1.2277 | 1.1634 | 1.1206 | 1.0900 | 1.0670 | 1.0490 | 1.0349 | 1 0232 | 1.0135 | 0.9982 | 0.9867 | 335 |
| 30 | 1.1160 | 1.0660 | 1.0327 | 1.0089 | 0.9910 | 0.9770 | 0.9660 | 0.9569 | 0.9494 | .9375 | .9285 | 330 |
| 35 | 0.9902 | 0.9560 | 0.9332 | 0.9169 | .9047 | .8951 | .8876 | .8813 | .8762 | .8680 | .8619 | 325 |
| 40 | .8529 | .8355 | .8239 | .8157 | .8095 | .8046 | .8008 | .7976 | .7950 | .7909 | .7878 | 320 |
| 45 | .7071 | .7071 | .7071 | .7071 | .7071 | .7071 | .7071 | .7071 | .7071 | .7071 | .7071 | 315 |
| 50 | .5560 | .5733 | .5849 | .5932 | .5994 | .6042 | .6081 | .6112 | .6138 | .6180 | .6211 | 310 |
| 55 | .4026 | .4368 | .4596 | .4759 | .4881 | .4977 | .5052 | .5113 | .5166 | .5247 | .5308 | 305 |
| 60 | .2500 | .3000 | .3334 | .3572 | .3750 | .3890 | .4000 | .4091 | .4167 | .4286 | .4375 | 300 |
| 65 | .1012 | .1655 | .2084 | .2390 | .2619 | .2799 | .2941 | .3058 | .3155 | .3308 | .3423 | 295 |
| 70 | −.0410 | .0356 | .0867 | .1232 | .1505 | .1720 | .1888 | .2028 | .2143 | .2326 | .2463 | 290 |
| 75 | −.1742 | −.0876 | −.0298 | .0114 | .0423 | .0666 | .0856 | .1014 | .1115 | .1351 | .1506 | 285 |
| 80 | −.2962 | −.2022 | −.1396 | −.0948 | −.0613 | −.0350 | −.0143 | .0028 | .1700 | .0394 | .0562 | 280 |
| 85 | −.4071 | −.3068 | −.2411 | −.1942 | −.1590 | −.1315 | −.1098 | −.0919 | −.0770 | −.0536 | −.0359 | 275 |
| 90 | −.5000 | −.4000 | −.3333 | −.2857 | −.2500 | −.2222 | −.2000 | −.1818 | −.1667 | −.1429 | −.1250 | 270 |
| 95 | −.5814 | −.4811 | −.4154 | −.3685 | −.3333 | −.3058 | −.2841 | −.2661 | −.2513 | −.2279 | −.2103 | 265 |
| 100 | −.6435 | −.5495 | −.4869 | −.4421 | −.4090 | −.3823 | −.3620 | −.3445 | −.3303 | −.3079 | −.2911 | 260 |
| 105 | −.6918 | −.6052 | −.5475 | −.5062 | −.4753 | −.4511 | −.4320 | −.4163 | −.4032 | −.3826 | −.3671 | 255 |
| 110 | −.7350 | −.6484 | −.5973 | −.5609 | −.5335 | −.5121 | −.4952 | −.4813 | −.4697 | −.4515 | −.4378 | 250 |
| 115 | −.7440 | −.6797 | −.6369 | −.6063 | −.5833 | −.5653 | −.5512 | −.5395 | −.5297 | −.5145 | −.5030 | 245 |
| 120 | −.7500 | −.7000 | −.6667 | −.6429 | −.6250 | −.6110 | −.6000 | −.5909 | −.5834 | −.5715 | −.5625 | 240 |
| 125 | −.7446 | −.7104 | −.6876 | −.6713 | −.6591 | −.6495 | −.6420 | −.6358 | −.6306 | −.6225 | −.6163 | 235 |
| 130 | −.7396 | −.7123 | −.7007 | −.6924 | −.6862 | −.6813 | −.6775 | −.6741 | −.6717 | −.6676 | −.6645 | 230 |
| 135 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | −.7071 | 225 |
| 140 | −.6792 | −.6966 | −.7032 | −.7164 | −.7226 | −.7275 | −.7313 | −.7345 | −.7371 | −.7412 | −.7443 | 220 |
| 145 | −.6481 | − 6823 | −.7052 | −.7214 | −.7336 | −.7432 | −.7508 | −.7570 | −.7621 | −.7703 | −.7764 | 215 |
| 150 | −.6160 | −.6660 | −.6994 | −.7232 | −.7410 | −.7550 | −.7660 | −.7751 | −.7827 | −.7946 | −.8035 | 210 |
| 155 | −.5849 | −.6492 | −.6921 | −.7227 | −.7456 | −.7636 | −.7778 | −.7895 | −.7992 | −.8145 | −.8260 | 205 |
| 160 | −.5567 | −.6333 | −.6844 | −.7208 | −.7481 | −.7696 | −.7865 | −.8004 | −.8120 | −.8302 | −.8439 | 200 |
| 165 | −.5329 | −.6159 | −.6773 | −.7185 | −.7494 | −.7737 | −.7927 | −.8085 | −.8216 | −.8422 | −.8577 | 195 |
| 170 | −.5150 | −.6089 | −.6716 | −.7163 | −.7499 | −.7762 | −.7969 | −.8140 | −.8282 | −.8505 | −.8674 | 190 |
| 175 | −.5038 | −.6023 | −.6680 | −.7148 | −.7500 | −.7778 | −.7992 | −.8172 | −.8320 | −.8555 | −.8731 | 185 |
| 180 | −.5000 | −.6000 | −.6667 | −.7143 | −.7500 | −.7778 | −.8000 | −.8182 | −.8333 | −.8571 | −.8750 | 180 |

reciprocating parts, lb.; $A$ = area of piston, sq. in.; $r$ = crank radius, ft.; $g$ = acceleration due to gravity = 32.2. Table 2, p. 11–13, gives values of $(\cos\alpha \pm r\cos 2\alpha/l)$ for various ratios of $l/r$.

**POSITION OF PISTON OR CROSSHEAD FOR VARIOUS CRANK ANGLES** is given in Table 3 for various ratios of $l/r$. If $L_p$ = distance of piston or crosshead from the dead center position, at any given crank angle,

$$L_p = r\,(1 - \cos\alpha \pm r^2\sin^2\alpha/2l), \text{ approx.} \quad \ldots\ldots \quad [18]$$

If the center line of motion of the reciprocating parts is not in the same plane as the center line of the crank-shafts, but is at a perpendicular distance $d$ away from it, the stroke is greater than $2r$, and the dead centers are not symmetrical. In this case the length of stroke is

$$S = \sqrt{(1+r)^2 - d^2} - \sqrt{(1-r)^2 - d^2} \quad \ldots\ldots \quad [19]$$

**Graphical Determination of Velocity and Acceleration of Reciprocating Parts.**— Fig. 11 shows a graphical method of determining the velocity and acceleration of the piston or crosshead at any given position of the crank. This method is known as the Klein construction. From the center $A$ of the crank, draw $AC$ to a scale representing the linear velocity of the crank-pin, at the desired angle to the line $AE$ joining the center line of the crank and crosshead. With $C$ as the center and $AC$ as the radius, describe a circle. Draw the connecting-rod $CE$ to the same scale as $AC$ and describe on it a circle of diameter $CE$. Draw the chord $FG$ between the intersection of the two circles, intersecting $AE$ at $D$. At $A$, erect a perpendicular to $AE$ intersecting the prolongation of $CE$ at $B$. Then the length $AD$ represents the acceleration of the crosshead and the length $AB$ the velocity of the crosshead at the position of the crank as drawn.

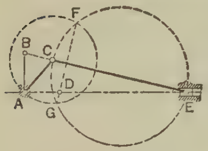


Fig. 11. Graphical Determination of Velocity and Acceleration

**Table 3.—Piston Positions for Various Crank Angles**

From beginning of stroke toward crank-shaft. Distance of piston from beginning of stroke = tabular value × length of stroke.

| Crank Angle, deg. | Ratio (Length of connecting-rod/length of crank) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 7 | 8 |
| 5 | 0.00285 | 0.00266 | 0.00254 | 0.00245 | 0.00238 | 0.00233 | 0.00228 | 0.00224 | 0.00222 | 0.00217 | 0.00214 |
| 10 | .01137 | .01061 | 0.01011 | 0.00975 | .00948 | .00927 | .00910 | .00896 | .00885 | .00867 | .00854 |
| 15 | .02545 | .02375 | .02263 | .02183 | .02123 | .02075 | .02040 | .02008 | .01983 | .01942 | .01913 |
| 20 | .04488 | .04191 | .03993 | .03852 | .03748 | .03667 | .03601 | .03548 | .03500 | .03433 | .03381 |
| 25 | .06943 | .06484 | .06151 | .05965 | .05799 | .05680 | .05579 | .05499 | .05430 | .05323 | .05244 |
| 30 | .09873 | .09224 | .08797 | .08494 | .08268 | .08092 | .07952 | .07838 | .07742 | .07593 | .07481 |
| 35 | .13243 | .12377 | .11810 | .11408 | .11110 | .10881 | .10693 | .10542 | .10416 | .10219 | .10072 |
| 40 | .17003 | .15900 | .15181 | .14674 | .14297 | .14005 | .13772 | .13582 | .13424 | .13177 | .12991 |
| 45 | .21103 | .19749 | .18871 | .18253 | .17795 | .17440 | .17158 | .16927 | .16735 | .16434 | .16210 |
| 50 | .25487 | .23874 | .22833 | .22102 | .21563 | .21145 | .20812 | .20542 | .20316 | .19964 | .19699 |
| 55 | .30094 | .28222 | .26962 | .26182 | .25556 | .25080 | .24699 | .24388 | .24130 | .23726 | .23423 |
| 60 | .34861 | .32739 | .31386 | .30442 | .29744 | .29195 | .28778 | .28431 | .28142 | .27689 | .27343 |
| 65 | .39726 | .37372 | .35878 | .34838 | .34071 | .33480 | .33011 | .32628 | .32311 | .31815 | .31444 |
| 70 | .44624 | .42066 | .40448 | .39324 | .38496 | .37859 | .37353 | .36942 | .36602 | .36067 | .35667 |
| 75 | .49495 | .46766 | .45047 | .43856 | .42978 | .42304 | .41767 | .41334 | .40972 | .40407 | .39986 |
| 80 | .54281 | .51425 | .49630 | .48388 | .47474 | .46772 | .46215 | .45763 | .45386 | .44799 | .44360 |
| 85 | .58930 | .55995 | .54155 | .52881 | .51944 | .51225 | .50654 | .50191 | .49806 | .49205 | .48756 |
| 90 | .63398 | .60436 | .58678 | .57295 | .56351 | .55627 | .55048 | .54583 | .54196 | .53591 | .53138 |
| 95 | .67646 | .64711 | .62870 | .61596 | .60659 | .59940 | .59370 | .58906 | .58522 | .57921 | .57471 |
| 100 | .71646 | .68790 | .66995 | .65753 | .64839 | .64137 | .63580 | .63127 | .62751 | .62163 | .61725 |
| 105 | .75376 | .72648 | .70929 | .69737 | .68860 | .68186 | .67650 | .67216 | .66854 | .66288 | .65867 |
| 110 | .78826 | .76268 | .74650 | .73526 | .72698 | .72062 | .71555 | .71144 | .70804 | .70269 | .69869 |
| 115 | .81987 | .79634 | .78140 | .77100 | .76332 | .75741 | .75272 | .74890 | .74573 | .74077 | .73706 |
| 120 | .84862 | .82739 | .81386 | .80442 | .79744 | .79196 | .78778 | .78431 | .78142 | .77689 | .77343 |
| 125 | .87455 | .85580 | .84319 | .83539 | .82917 | .82438 | .82057 | .81746 | .81487 | .81083 | .80781 |
| 130 | .89766 | .88152 | .87112 | .86381 | .85842 | .85424 | .85091 | .84820 | .84595 | .84242 | .83977 |
| 135 | .91814 | .90460 | .89581 | .88964 | .88506 | .88151 | .87868 | .87637 | .87446 | .87145 | .86921 |
| 140 | .93608 | .92504 | .91786 | .91279 | .90902 | .90610 | .90377 | .90186 | .90029 | .89781 | .89595 |
| 145 | .95158 | .94292 | .93725 | .93323 | .93025 | .92796 | .92608 | .92457 | .92332 | .92135 | .91987 |
| 150 | .96476 | .95827 | .95399 | .95097 | .94870 | .94695 | .94554 | .94440 | .94345 | .94196 | .94083 |
| 155 | .97574 | .97114 | .96782 | .96596 | .96430 | .96310 | .96210 | .96128 | .96061 | .95954 | .95875 |
| 160 | .98457 | .98160 | .97963 | .97822 | .97717 | .97636 | .97570 | .97517 | .97469 | .97403 | .97350 |
| 165 | .99137 | .98968 | .98855 | .98776 | .98715 | .98669 | .98632 | .98601 | .98576 | .98536 | .98505 |
| 170 | .99618 | .99542 | .99492 | .99457 | .99429 | .99408 | .99391 | .99377 | .99366 | .99348 | .99335 |
| 175 | .99905 | .99885 | .99873 | .99864 | .99858 | .99852 | .99847 | .99844 | .99841 | .99836 | .99833 |
| 180 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 | 1.00000 |

# FLY-WHEELS

## 1. ENERGY OF FLY-WHEELS

**FLUCTUATIONS OF ENERGY.**—The function of a fly-wheel is to store up and to restore the periodic fluctuations of energy given to or taken from an engine or machine, and thus to keep approximately constant the velocity of rotation. The quantity $\Delta E/2E_0$ is the coefficient of the fluctuation of speed or of unsteadiness, where $E_0$ = mean actual energy, and $\Delta E$ = excess of energy received or of work performed, above the mean, during a given interval. The ratio of the periodical excess or deficiency of energy $\Delta E$ to the whole energy exerted in one period ranges in single-cylinder engines worked expansively, from $^1/_6$ to $^1/_4$; the shorter the cut-off the higher the value. For a pair of engines with cranks at 90 deg. the value of the ratio is about $^1/_4$ its value for single-cylinder engines; for three engines with cranks at 120 deg., $^1/_{12}$ its value for single-cylinder engines. For tools working at intervals, such as punching, slotting, shearing, and forging machines, $\Delta E$ is nearly equal to the whole work performed at each operation. A fly-wheel reduces the coefficient $\Delta E/2E_0$ to a fixed amount, which is about $^1/_{32}$ for ordinary machinery, and $^1/_{50}$ or $^1/_{60}$ for machinery for fine purposes.

If $b$ = reciprocal of the intended value of the coefficient of fluctuation of speed, $\Delta E$ = fluctuation of energy, $I$ = moment of inertia of the fly-wheel alone, $a_0$ = its mean angular velocity, and $g$ = acceleration due to gravity = 32.2,

$$I = b\,g\,\Delta E/a_0^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

As the rim of a fly-wheel usually is heavy in comparison to the arms, $I$ may be taken as equal to $W\,r^2$, where $W$ = weight of rim, lb. and $r$ = radius of wheel. Then

$$W = b\,g\,\Delta E/a_0^2\,r^2 = b\,g\,\Delta E/v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

where $v$ = velocity of the rim, ft. per sec. The usual mean radius of the fly-wheel in steam engines is from 3 to 5 times the length of the crank. Ordinary values of the product $b\,g$, the unit of time being 1 second, lie between 1000 and 2000. Values of $b$ are given in Table 1.

**Table 1.—Values of Coefficient of Steadiness, *b*, for Fly-wheels**

| Engines Operating | $b$ | Authority |
|---|---|---|
| Crushing machinery | 5 | Mayer |
| Electrical " | 300 | Unwin |
| " " direct driven | 150 | " |
| Engines with belt transmission | 35 | Mayer |
| Flour milling machinery | 50 | " |
| Gear wheel transmission | 50 | " |
| Hammering machinery | 5 | " |
| Machine tools | 35 | Unwin |
| Paper-making machinery | 40 | Mayer |
| Pumping machinery | 20– 30 | " |
| Shearing machinery | 20– 30 | " |
| Spinning machinery | 50–100 | Mayer; Unwin |
| Textile machinery | 40 | Unwin |

**ENERGY OF FLY-WHEELS IN INTERMITTENT WORK.**—The method developed by O. S. Beyer (*Am. Mach.*, Oct. 17, 24, 1912) assumes that the weight of the rim of the fly-wheel is 68.6% and that of the hub and arm, 31.4% of the total weight of the wheel. These percentages were found to be practically constant in a large number of fly-wheels examined. Table 2 gives the velocity, ft. per sec., at 1 r.p.m. of the center of gyration and of the rim, for the cross-sections of rim shown in Fig. 1, with different diameters of fly-wheels. The velocity factor in the table multiplied by the r.p.m. of the wheel gives the actual velocity. Energy of the wheel then can be determined from the formula

$$E = {}^1/_2\,(Wv_1^2/g) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where $E$ = energy, ft.-lb., $W$ = weight, lb., $v_1$ = velocity of radius of gyration, ft. per sec. Energy also can be determined from Fig. 2 when the value on the energy scale corresponding to the given velocity is multiplied by the total weight in pounds of the fly-wheel.

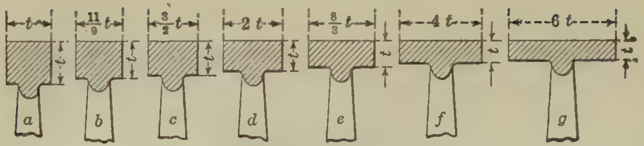


FIG. 1. Fly-wheel Rim Cross-sections

**Table 2.—Velocity of Center of Gyration of Fly-wheel Rim Sections**

| Outer Diameter of Wheel, in. | Type of Rim Cross-section. See Fig. 1 | | | | | | | Velocity of Outer Circumference, ft. per sec. |
|---|---|---|---|---|---|---|---|---|
| | a | b | c | d | e | f | g | |
| | Velocity, ft. per sec., at 1 r.p.m. | | | | | | | |
| 12 | 0.0406 | 0.0411 | 0.0417 | 0.0423 | 0.0428 | 0.0434 | 0.0440 | 0.0524 |
| 18 | .0609 | .0617 | .0625 | .0634 | .0642 | .0651 | .0660 | .0785 |
| 24 | .0812 | .0822 | .0834 | .0845 | .0857 | .0868 | .0880 | .1047 |
| 30 | .1014 | .1028 | .1042 | .1056 | .1071 | .1085 | .1100 | .1309 |
| 36 | .1217 | .1233 | .1250 | .1268 | .1285 | .1302 | .1319 | .1571 |
| 42 | .1420 | .1439 | .1459 | .1479 | .1499 | .1519 | .1539 | .1833 |
| 48 | .1623 | .1644 | .1667 | .1690 | .1713 | .1736 | .1759 | .2094 |
| 54 | .1826 | .1850 | .1876 | .1901 | .1927 | .1953 | .1979 | .2356 |
| 60 | .2029 | .2055 | .2084 | .2113 | .2141 | .2170 | .2199 | .2618 |
| 66 | .2232 | .2261 | .2292 | .2334 | .2356 | .2387 | .2419 | .2880 |
| 72 | .2435 | .2466 | .2501 | .2535 | .2570 | .2604 | .2639 | .3142 |
| 78 | .2638 | .2672 | .2709 | .2747 | .2784 | .2821 | .2859 | .3403 |
| 84 | .2841 | .2877 | .2917 | .2958 | .2998 | .3038 | .3079 | .3665 |
| 90 | .3043 | .3083 | .3126 | .3169 | .3212 | .3255 | .3299 | .3927 |
| 96 | .3246 | .3288 | .3334 | .3380 | .3426 | .3473 | .3519 | .4189 |
| 102 | .3449 | .3494 | .3543 | .3592 | .3641 | .3690 | .3739 | .4451 |
| 108 | .3652 | .3699 | .3751 | .3803 | .3855 | .3907 | .3958 | .4712 |
| 114 | .3855 | .3905 | .3959 | .4014 | .4069 | .4124 | .4178 | .4974 |
| 120 | .4058 | .4110 | .4168 | .4225 | .4283 | .4341 | .4398 | .5236 |

Fig. 3 gives relation between loss of velocity in a fly-wheel and energy expended. Fig. 4 shows the relation of total and expended energy to loss of velocity. By means of Table 2 and these three curves practically all problems relating to fly-wheels on intermittent work may be solved.

EXAMPLE 1.—Find the energy in a cast-iron fly-wheel 60 in. outside diameter with a rim 6 × 6 in. of cross-section *a*, Fig. 1, running at 100 r.p.m. *Solution.*—Taking weight of cast iron as 0.26 lb. per cu. in., weight of the rim, representing 68% of total weight of wheel, is

$$(60 - 6) \times 3.1416 \times 6 \times 6 \times 0.26 = 1588 \text{ lb.}$$

The wheel complete weighs 1588 ′0.686 = 2315 lb. From Table 2 the velocity of radius of gyration for a 60-in. wheel is 0.2029 ft. per sec. at 1 r.p.m., or 20.29 ft. per sec. at 100 r.p.m. The energy in the wheel is

$$E = \tfrac{1}{2}(Wv_1^2/g) = \tfrac{1}{2}\{(2315 \times 20.29^2)/32.2\} = 14{,}800 \text{ ft.-lb.}$$

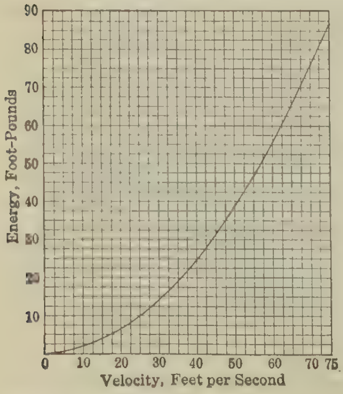


FIG. 2. Relation of Energy and Velocity

The same result may be obtained by finding in Fig. 2 the energy corresponding to 20.29 ft. per sec. to be 6.40 ft.-lb., whence 2315 × 6.40 = 14,816 ft.-lb.

EXAMPLE 2.—What energy will be expended by the wheel in Example 1 if the velocity is reduced to 85 r.p.m.? *Solution.*—Reduction in velocity is (100–85) = 15 r.p.m. = 15%. Below the point on the curve corresponding to a diminution of velocity of 15% read the percentage of energy expended as 28. The total energy in the wheel at 100 r.p.m. being 14,800 ft.-lb., the energy expended by reducing the speed to 85 r.p.m. will be 14,800 × 0.28 = 4144 ft.-lb.

Fig. 4 gives information as to the energy that may be expended when the diminution of velocity is to be kept within certain limits, and also the diminution of velocity that will occur when the energy expenditure is limited to a certain percentage of the total energy.

EXAMPLE 3.—What energy must be contained in a fly-wheel that must deliver up to 500 ft.-lb. during each cycle with a diminution of speed not to exceed 8%?

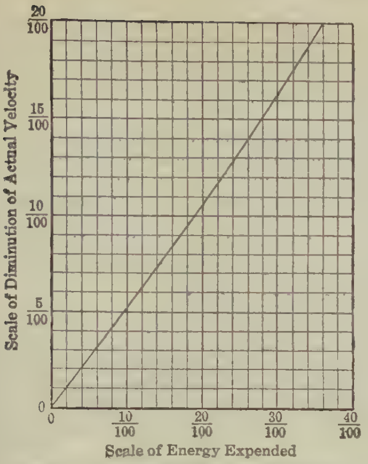


FIG. 3. Relation of Energy Expended and Velocity

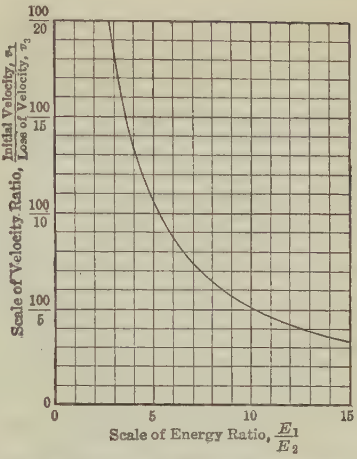


FIG. 4. Relation of Energy Expended and Diminution of Velocity

*Solution.*—Below the point on the curve, Fig. 4, corresponding to the ratio $v_1/v_3 = 100/8$, read on the scale of energy ratio $E_1$ $E_2$ the quantity 6.5. This is the ratio of total energy of the wheel to expended energy. Hence, the total energy in the wheel will be $6.5 \times 500 = 3250$ ft.-lb.

EXAMPLE 4.—Required the normal velocity of a fly-wheel that is to expend a maximum of $33^1/_3$% of its total energy, with a minimum speed of 65 r.p.m. Then $v_1$ = normal velocity, $v_2$ = minimum velocity = 65 r.p.m., and $v_3$ is the loss of velocity = $(v_1 - v_2)$. $E_1$ = energy in the wheel at velocity $v_1$, and $E_2$ = energy expended. Then $E_1/E_2 = 100/33^1/_3 = 3$. *Solution.*—Opposite the point on the curve, Fig. 4, corresponding to $E_1/E_2 = 3$ find 100/18 on scale of $v_1/v_3$. Then $v_1/v_3 = 100/18$, and $v_3 = 0.18\ v_1$. $v_3 = (v_1 - v_2) = (v_1 - 65)$, whence $0.18\ v_1 = (v_1 - 65)$, and $v_1 = 79.2$ r.p.m.

**STRESSES IN FLY-WHEELS.**—Let $W$ = weight of rim, lb. $D$ = diameter of wheel, ft. $R$ = mean radius of rim, ft. $N$ = rev. per min. $V$ = velocity of rim, ft. per min. $v$ = velocity of rim, ft. per sec. = $2\,\pi\,R\,N/60$. $g$ = acceleration due to gravity = 32.2. Then centrifugal force of whole rim is

$$F = Wv^2/gR = 4\,W\pi^2\,RN^2/3600\,g = 0.000\,341\,WRN^2 \quad . \quad . \quad . \quad . \quad [4]$$

The resultant of half the centrifugal force acting at right angles to a diameter tends to disrupt one-half of the wheel from the other half. It is resisted by the section of the rim at each end of the diameter. The ratio of the resultant of half the radial forces, taken at right angles to the diameter, to the sum of the radial forces is $1/(1/2\pi) = 2/\pi$. Hence the force tending to rupture the fly-wheel is

$$F_1 = (F/2) \times (2/\pi) = (0.000341\,WRN^2/2) \times (2/\pi) = 0.000\,108\,54\,WRN^2 \quad . \quad [5]$$

Since the force is resisted by the section at each end of the diameter the stress $S$ at any cross-section of the rim is $^1/_2$ of this, and $S = 0.000\,054\,27\,W\,R\,N^2$. The weight $W_1$ of a rim of cast iron of 1 sq. in. cross-section is $2\pi\,R \times 3.125 = 19{,}635\,R$ lb., whence stress, lb. per sq. in. of cross-sectional area of rim, is

$$S_1 = 0.001\,065\,6\,R^2\,N^2 = 0.000\,266\,4\,D^2\,N^2 = 0.000\,027\,V^2 \quad . \quad . \quad . \quad . \quad [6]$$

For wrought iron

$$S_1 = 0.001\,136\,6\,R^2\,N^2 = 0.000\,284\,2\,D^2\,N^2 = 0.000\,028\,8\,V^2$$

For steel,

$$S_1 = 0.001\,159\,3\,R^2\,N^2 = 0.000\,290\,1\,D^2\,N^2 = 0.000\,029\,4\,V^2$$

For wood,

$$S_1 = 0.000\,088\,8\,R^2\,N^2 = 0.000\,022\,2\,D^2\,N^2 = 0.\,000\,002\,25\,V^2$$

The specific gravity of wood is taken as 0.6 = 37.5 lb. per cu. ft. or $^1/_{12}$ the weight of cast iron.

## 2. WEIGHT OF FLY-WHEELS

**WEIGHT OF FLY-WHEELS FOR ENGINES.**—The weight of a fly-wheel for a given engine can be determined by means of equation [2]. The determination of $\Delta E$, however, involves a tedious graphical analysis of the performance of the engine from the probable indicator card in order to ascertain the excess or deficiency of energy at various points of the stroke. Various empirical equations of the form $W = C \times \text{Hp.}/N^2\,D^2$ have been devised to determine the weight without this analysis. Several of these are given in Table 3. In the application of the equations, the weight of the hub and arms should be subtracted from the total weight of the wheel. While in large wheels the weight of these parts may amount to from 30 to 35% of the total weight, their fly-wheel effect is only 7 to 10% of that of the rim.

**FLY-WHEELS FOR PRESSES, PUNCHES, SHEARS, ETC.**—The function of the fly-wheel on punching and similar machinery, in which action is intermittent, is to store up energy during the portion of the stroke when no work is being done and to deliver it during the period of actual working. The delivery of energy is accompanied by a reduction in the velocity of the fly-wheel.

**Notation.**—$E$ = total energy in the wheel at maximum velocity, ft.-lb. $E_1$ = energy delivered by the wheel during speed reduction, ft.-lb. $v_1$ = initial velocity of center of

### Table 3.—Equations for Weight of Fly-wheels

**Notation.**—$W$ = weight of fly-wheel rim, lb. $A$ = area of cylinders, sq. in. $s$ = length of stroke, in. $S$ = length of stroke, ft. $p$ = mean steam pressure, lb. per sq. in. $K$ = variation of speed, percent of mean speed. $D$ = diameter of fly-wheel, ft. $N$ = rev. per min. I.Hp. = indicated horsepower of engine. $d$ = diam. of engine cylinder, in. $j$ = a coefficient, values given in Table 3. $b$ = coefficient of steadiness, values given in Table 1. $C$ = constant. $V$ = mean velocity of fly-wheel rim, ft. per sec.

| Authority | $W$ = | Value of $C$ | | Reference | Equation No. |
|---|---|---|---|---|---|
| Rankine | $\frac{CASp}{KD^2N^2}$ | 475,000 | $K$ = 1 1/2 to 2% for good governing;<br>$K$ = 3% in portable engines, ordinary load;<br>$K$ = 4% in portable engines, heavy load;<br>$K$ = 2 1/2 to 3% for fixed engines for ordinary purposes. | Rankine, Useful Rules and Tables.<br>Hartnell (*Proc.* Inst. M. E., 1882) | [7] |
| Thurston | $\frac{CASp}{D^2N^2}$ | 250,000 | Corresponds to Rankine's formulas with $K$ = 1.9. For automatic valve-gear engines. | | [8] |
| Thurston | $\frac{CAS}{D^2N^2}$ | 10,000,000 to 15,000,000 | For ordinary non-condensing engines with ratio of expansion of from 3 to 5. | | [9] |
| Rites | $\frac{C \times \text{I.Hp.}}{N^3D^2}$ | 10,000,000 to 20,000,000 | | *Trans.* A.S.M.E., xiv, p. 100 | [10] |
| Stanwood | $\frac{Cd^2s}{D^2N^2}$ | 350,000*<br>700,000†<br>1,000,000‡ | For engines in which 480 ft. per min. is the lowest probable piston speed.<br>$n$ = r.p.m. corresponding to 480 ft. piston speed. | *Engg.*, June 12, 1891 | [11] |
| Theiss | $\frac{jb \times \text{I.Hp.}}{V^2N}$ | | Values of $j$ are given in Table 4.<br>Values of $b$ are given in Table 1. | *Am. Mach.*, Sept. 7 and 14, 1893 | [12] |

* For slide-valve engines, ordinary duty.
† Slide-valve engines for electric lighting and Corliss engines.
‡ For automatic high-speed engines and for Corliss engines.

### Table 4.—Values of $j$ for Use in the Theiss Formula for Weight of Fly-wheel

| Cut-off | Compression (See Note) | Single-Cylinder Engines | | | | | | 2-Cylinder Engines† | | | 3-Cyl. Engines† | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Non-Condensing | | | | Condensing | | Cranks at 90 deg. | | | Cranks at 120 deg. | |
| | | Piston Speed, ft. per min. | | | | | | | | | | |
| | | 200 | 400 | 600 | 800 | 200 | 400 | 200 | 400 | 800 | 200 | 800 |
| 1/6 | $p$ | 272,690 | 240,810 | 194,670 | 158,200 | 234,160 | 174,380 | 71,980 | 70,160 | 70,040 | 33,810 | 30,190 |
| | $0.7p$ | 241,530 | 209,890 | 165,450 | 132,020 | 206,030 | 151,680 | ...... | ...... | ...... | ...... | ...... |
| | 0 | 218,580 | 187,430 | 145,400 | 108,690 | 173,660 | 118,350 | 60,140* | 60,140* | 60,140* | 32,240 | 31,570 |
| 1/4 | $p$ | 242,010 | 208,200 | 168,590 | 162,070 | 204,210 | 164,720 | 59,420 | 57,000 | 60,140 | 33,810 | 35,140 |
| | $0.7p$ | 220,280 | 188,880 | 151,440 | 148,540 | 185,250 | 148,780 | ...... | ...... | ...... | ...... | ...... |
| | 0 | 209,170 | 179,460 | 136,460 | 135,260 | 167,140 | 133,080 | 54,340* | 54,340* | 54,340* | 35,500 | 33,810 |
| 1/3 | $p$ | 220,760 | 188,510 | 165,210 | ...... | 189,600 | 174,630 | 49,272 | 49,150 | ...... | 34,540 | 36,470 |
| | $0.7p$ | 207,230 | 176,080 | 150,710 | ...... | 173,900 | 164,970 | ...... | ...... | ...... | ...... | ...... |
| | 0 | 201,920 | 170,040 | 146,610 | ...... | 161,830 | 151,680 | 50,000* | 50,000* | ...... | 33,450 | 32,850 |
| 1/2 | $p$ | 193,340 | 174,630 | ...... | ...... | 172,690 | ...... | 37,920 | 35,500 | ...... | 35,260 | 33,810 |
| | $0.7p$ | 187,670 | 167,860 | ...... | ...... | 165,930 | ...... | ...... | ...... | ...... | ...... | ...... |
| | 0 | 182,840 | 167,860 | ...... | ...... | 156,990 | ...... | 36,950* | 36,950 | ...... | 32,370 | 32,370 |

NOTE.—$p$ denotes compression to initial steam pressure; $0.7p$ denotes compression to 0.7 × initial steam pressure; 0 denotes no compression.

* Mean values. † Values useful for comparison; fly-wheels dimensioned from these values probably would be so light as to be structurally weak.

gravity of fly-wheel rim, ft. per sec. $v_2$ = velocity of center of gravity of fly-wheel rim, at end of period during which energy is delivered. Hp. = horsepower required. $N$ = strokes of press or shear per min. $T$ = time required for stroke, sec. $t$ = time required for actual cutting of metal per stroke, sec. $W$ = weight of fly-wheel rim, lb. $d$ = diameter of rim at center of gravity. $R$ = r.p.m. of fly-wheel at initial velocity. $a$ = width of fly-wheel rim, in. $h$ = depth of fly-wheel rim, in. $y$ = ratio of depth to width of rim. $g$ = acceleration due to gravity = 32.2. $c$, and $c_1$ = constants whose values are as follows:

| Percent speed reduction = | 2 1/2 | 5 | 7 1/2 | 10 | 15 | 20 |
|---|---|---|---|---|---|---|
| $c$ = | 0.000 002 13 | .000 004 26 | .000 006 17 | .000 008 10 | .000 011 80 | .000 015 35 |
| $c_1$ = | 0.1250 | .0625 | .0432 | .0328 | .0225 | .0173 |

$$E = Wv_1^2/2g = Wv_1^2/64.4 \qquad [13]$$

$$E_1 = W(v_1^2 - v_2^2)/64.4 \qquad [14]$$

$$W = (E_1 \times 64.4)/(v_1^2 - v_2^2) \qquad [15]$$

$$v = 2\pi RN/60 \qquad [16]$$

A simplified method of calculating fly-wheels for punches and shears is given in *Machinery's Handbook*, page 316. Using the above notation,

$$\text{Hp.} = EN/33{,}000 = E/(T \times 550) \qquad [17]$$

$$E_1 = E\{1 - (t/T)\} \qquad [18]$$

$$W = E_1/cD^2R^2 \qquad [19]$$

$$a = \sqrt{1.22\,W/12\,Dy} \qquad [20]$$

$$h = ay \qquad [21]$$

For cast-iron fly-wheels with maximum stresses of 1000 lb. per sq. in., $W = c_1 E_1$; $R = 1940/D$.

Wilfred Lewis shows (*Trans.* A.S.M.E., vii, p. 549, 1886) that in belt-driven machines the belt will run off the pulley when slip exceeds 20% of the belt speed. This is the limiting low velocity, $v_2$, and for belt-driven punches and shears gives a limiting condition of

$$W = 180\,(E/v_1^2) \qquad [22]$$

**FLY-WHEELS FOR AIR COMPRESSORS.**—H. R. Goss and H. V. Putnam in *Trans.* A.S.M.E. APM-51-12, 1929, present an analysis of the determination of fly-wheels for air compressors. The paper is highly mathematical and presents curves from which the energy of the fly-wheel can be determined. Lack of space precludes presentation of the curves and formulas and the reader is referred to the original reference.

## 3. SAFE SPEEDS OF FLY-WHEELS

**DIAMETERS OF FLY-WHEELS FOR VARIOUS SPEEDS.**—If 6000 ft. per minute be the maximum rim velocity allowable, then $6000 = \pi RD$, where $R$ = rev. per min. and $D$ = diameter of wheel, ft., whence $D = 6000 \div \pi R = 1910 \div R$.

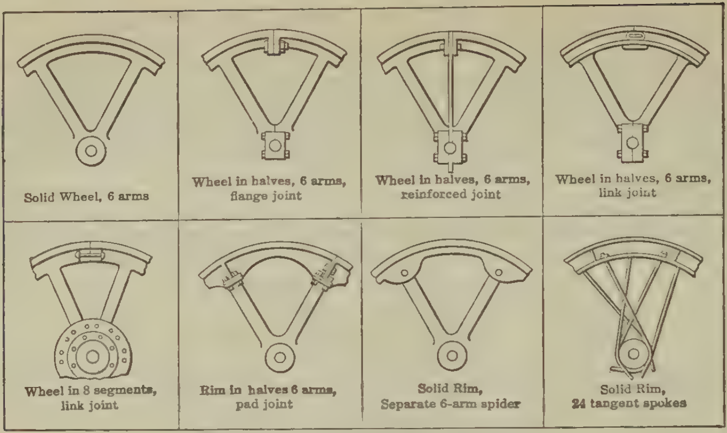
W. H. Boehm (*Eng. News*, Oct. 2, 1902), says: For a given material there is a definite speed at which rupture will occur, regardless of the amount of material used. This is expressed by

$$V = 1.6\sqrt{S/W}, \qquad [23]$$

where $V$ = rim velocity of wheel, ft. per sec. at which rupture will occur; $W$ = weight of 1 cu. in. of the material used; $S$ = tensile strength of 1 sq. in. of the material.

For cast-iron wheels made in one piece, assuming 10,000 lb. per sq. in. as the strength of large castings, and applying a factor of safety of 10, $V = 1.6\sqrt{1000/0.26} = 100$ ft. per sec. for the safe speed. For cast steel of 60,000 lb. per sq. in., $V = 1.6\sqrt{6000 \div 0.28} = 233$ ft. per sec. This is for wheels made in one piece. If the wheel is made in halves, or sections, the efficiency of the rim joint must be taken into consideration. For belt wheels with flanged and bolted rim joints located between the arms, the joints average only one-fifth the strength of the rim, and no such joint can be designed having a strength greater than one-fourth the strength of the rim. If the rim is thick enough to allow the joint to be reinforced by steel links shrunk on, as in heavy balance wheels, one-third the strength of the rim may be obtained in the joint. This construction cannot be applied to belt wheels with thin rims.

Table 5 is given by Mr. Boehm (*Am. Mach.*, Nov. 17, 1904) as the safe revolutions per minute of cast-iron wheels of different diameters. If $s$ = strength of a wheel with no joint, the strength of the various types of joint is: Flange joint, $0.25\,s$; pad joint, $0.50\,s$;



Solid Wheel, 6 arms

Wheel in halves, 6 arms, flange joint

Wheel in halves, 6 arms, reinforced joint

Wheel in halves, 6 arms, link joint

Wheel in 8 segments, link joint

Rim in halves 6 arms, pad joint

Solid Rim, Separate 6-arm spider

Solid Rim, 24 tangent spokes

Fig. 5. Types of Fly-wheel Joints

link joint, 0.60 *s*. The construction of these various types of joint is shown in Fig. 5. The table is based on a margin of safety on speed of approximately 3, which is equivalent to a margin on stress developed, or factor of safety in the usual sense of 9. The table assumes the solid wheel to have an efficiency of construction *e* of unity, whereas Prof. Benjamin's tests (see below) give it an efficiency of only 0.85. The table probably overestimates the strength of the wheels.

**Wooden Wheels.**—For hard maple, having a tensile strength of 10,500 lb. per sq. in., and weighing 0.0283 lb. per cu. in., we have, using a factor of safety of 20, and remembering that the strength is reduced one-half because the wheel is built up of segments, $V = 1.6\sqrt{262.5 \div 0.0283} = 154$ ft. per sec. The stress in a wheel varies as the square of the speed, and the factor of safety on speed is the square root of the factor of safety on strength.

**DESTRUCTIVE SPEEDS OF FLY-WHEELS.**—Various types of fly-wheel construction (see Fig. 5) were tested to destruction by Prof. C. H. Benjamin (*Trans.* A.S.M.E., xx, p. 209, 1899, xxiii, p. 168, 1902), by revolving them in a bomb-proof casing. The speed was increased gradually until the wheel burst. The averages of the results of the experiments are given in Table 6. From two to four wheels of each type were tested, except Type 5, of which only one wheel was tested.

Table 5.—Safe Revolutions per Minute of Cast-iron Fly-wheels

| Diam., ft. | Type of Joint. (See Fig. 5) | | | | Diam., ft. | Type of Joint. (See Fig. 5) | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Solid | Flange | Pad | Link | | Solid | Flange | Pad | Link |
| 1 | 1910 | 955 | 1350 | 1480 | 16 | 120 | 60 | 84 | 92 |
| 2 | 955 | 478 | 675 | 740 | 17 | 112 | 56 | 79 | 87 |
| 3 | 637 | 318 | 450 | 493 | 18 | 106 | 53 | 75 | 82 |
| 4 | 478 | 239 | 338 | 370 | 19 | 100 | 50 | 71 | 78 |
| 5 | 382 | 191 | 270 | 296 | 20 | 95 | 48 | 68 | 74 |
| 6 | 318 | 159 | 225 | 247 | 21 | 91 | 46 | 65 | 70 |
| 7 | 273 | 136 | 193 | 212 | 22 | 87 | 44 | 62 | 67 |
| 8 | 239 | 119 | 169 | 185 | 23 | 84 | 42 | 59 | 64 |
| 9 | 212 | 106 | 150 | 164 | 24 | 80 | 40 | 56 | 62 |
| 10 | 191 | 96 | 135 | 148 | 25 | 76 | 38 | 54 | 59 |
| 11 | 174 | 87 | 123 | 135 | 26 | 74 | 37 | 52 | 57 |
| 12 | 159 | 80 | 113 | 124 | 27 | 71 | 35 | 50 | 55 |
| 13 | 147 | 73 | 104 | 114 | 28 | 68 | 34 | 48 | 53 |
| 14 | 136 | 68 | 96 | 106 | 29 | 66 | 33 | 47 | 51 |
| 15 | 128 | 64 | 90 | 99 | 30 | 64 | 32 | 45 | 49 |

Table 6.—Results of Tests to Destruction of Various Types of Fly-wheels

| Type of wheel (see Fig. 5) | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| Rim Speed at Failure, ft. per sec. | 395 | 194 | 225 | 305 | 256 | 223 | 393 | 424 |
| Rim Speed at Failure, ft. per min. | 23,700 | 11,640 | 13,500 | 18,300 | 15,360 | 13,380 | 23,580 | 25,440 |
| Apparent rim stress at failure, lb. per sq. in. | 15,625 | 3,764 | 5,062 | 9,302 | 6,502 | 4,973 | 15,445 | 17,978 |
| Comparative rim speeds at failure | 100 | 49 | 57 | 77 | 65 | 56.5 | 100 | 107 |
| Comparative rim stress at failure | 100 | 24 | 32.5 | 60 | 42 | 32 | 99 | 115 |
| Efficiency of construction (*e*)* | 0.85 | 0.19 | 0.265 | 0.49 | 0.34 | 0.26 | 0.84 | 0.94 |

* By equation [24]. See p. 11–21.

## 4. DIMENSIONS AND CONSTRUCTION OF FLY-WHEELS

**THICKNESS OF RIM.**—F. A. Halsey (Handbook for Machine Designers) gives the following equations for rim velocity at which bursting of the fly-wheel may be expected. Let $v$ = bursting velocity of rim, ft. per sec.; $t$ = tensile strength of material, lb. per sq. in.; $w$ = weight of material, lb. per cu. in.; $e$ = efficiency of construction, values being given in Table 6.

Then

$$v = 1.64 \sqrt{(t\,e/w)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [24]$$

For cast iron with $w = 0.26$ and $t = 19{,}000$, $v = 443 \sqrt{e}$.

For steel, with $w = 0.28$ and $t = 60{,}000$, $v = 757 \sqrt{e}$.

These equations should be used with caution in designing fly-wheels, as stresses exist in the rim other than simple tension due to centrifugal force. The arms restrain the free expansion of the rim, with the result that the free section of the rim between arms acts as a uniformly-loaded beam, the load being due to the centrifugal force of the material in the section. The stress due to this action is to be added to the simple tension stress, and the resulting stress is always greater than that given by the equations. This beam action is exceptionally serious in built-up wheels with joints between the arms, the joint being equivalent to a joint in the middle of a beam and not to a simple splice in a tension member. The beam action may be reduced by increasing the number of arms, and the fiber stress due to beam action, other things being equal, is inversely as the square of the number of arms.

An equation for the thickness of rim based on the number of arms is given by J. B. Standwood in *Trans.* A.S.M.E., xiv, p. 251, 1893. Let $T$ = thickness of rim, in.; $d$ = wheel diameter, in.; $A$ = number of arms; $v$ = rim velocity, ft. per sec.; $F$ = greatest stress, lb. per sq. in. to which any fiber is subjected. Then

$$T = 0.475\, d/A^2 \{(F/d) - 0.1\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

If $F$ is taken as 6000 lb. per sq. in. the following values of $T$ are found for solid rim wheels:

| | | | | | |
|---|---|---|---|---|---|
| Diameter of wheel, in. | 24 | 24 | 48 | 108 | 108 |
| Rim velocity, ft. per sec. | 50 | 88 | 88 | 184 | 184 |
| " " ft. per min. | 3,000 | 5,280 | 5,280 | 11,040 | 11,040 |
| No. of arms | 6 | 6 | 6 | 16 | 36 |
| Thickness of rim, in. | 2/10 | 15/32 | 15/16 | 2 1/2 | 1/2 |

If the limiting velocity be assumed at 88 ft. per sec. (1 mile per min.), with $F = 6000$, equation [25] becomes

$$T = 0.475\, d/0.67\, A^2 = 0.7\, d/A^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [26]$$

If the wheels are made in halves or sections, the bending strain must be taken into account and $T$ will be greater than given above. For a joint half-way between the arms, equation [25] becomes

$$T = 0.712\, d/A^2 \{(F/v^2) - 0.1\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [27]$$

With a maximum rim velocity of 88 ft. per sec., and $F = 6000$ $T = 1.05\, d/A^2$.

**ARMS OF FLY-WHEELS AND PULLEYS.**—Equations for the arms of wheels cast in one piece can be at best only approximations, as the shrinkage stresses introduce an uncertain factor.

Prof. Torrey (*Am. Mach.*, July 30, 1891) gives for arms of elliptical cross-section

$$b = WL \div 30\, d^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [28]$$

where $b$ = breadth (minor axis of ellipse) of arm at hub, in.; $L$ = length of arm, ft.; $W$ = load acting on one arm, lb., and $d$ = depth of arm at hub, in. (major axis of ellipse). The value of $W$ is given by $W = Sw/n$, where $S$ = stress on belt, lb. per in. of width; $w$ = width of belt, in., and $n$ = number of arms. The formula is based on a factor of safety of 10. In using it, a value is assumed for $d$ and the value of $b$ calculated. If the resulting cross-section is too round, a larger value of $d$ is taken and $b$ is recalculated. More than two trials are seldom necessary.

Unwin gives:

$$d = 0.6337 \sqrt[3]{BD/n} \text{ for single belts,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [29]$$

$$d = 0.798 \sqrt[3]{BD/n} \text{ for double belts,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [30]$$

where $D$ = diam. of wheel, in. and $B$ = breadth of the rim, in., other notation as before. These formulas are based on an elliptical section of arm in which $b = 0.4\, d$ or $d = 2.5\, b$, on a width of belt = 4/5 the width of the wheel rim, a maximum driving force transmitted by the belt of 56 lb. per inch of width for a single belt and 112 lb. for a double belt, and a safe working stress of cast iron of 2250 lb. per square inch.

Using in Torrey's formula, the same value of $d$ as found by Unwin's formula for a given set of conditions, the value of $b$ is approximately only one-half that given by Unwin; that is, the arm is but one-half as strong.

F. A. Halsey (*Am. Mach.*, April 23, 1896) gives

$$x = {}^{7}/_{8}\text{ in.} + 0.04\,d + 0.153\,c, \text{ and } y = {}^{1}/_{2}\,x, \quad . \quad . \quad . \quad . \quad . \quad [31]$$

where $x$ = depth of arm and $y$ = breadth of arm, $d$ = outside diameter of rim, $c = \sqrt{a \times b}$, $a$ and $b$ being the thickness and width of rim, respectively; $b = {}^{2}/_{3}\,a$. All dimensions are in inches. Halsey recommends a rectangular cross-section with its greatest dimension radial. The formulas provide for other sections by considering all sections of the same area as equivalents and taking the side of a square of equal area as the base of the factor for the section.

The arms should taper both in breadth and depth from the hub to the rim. Halsey recommends a taper on each side of the center line of $^{1}/_{8}$ to $^{3}/_{16}$ in. per ft. in breadth and $^{1}/_{4}$ to $^{3}/_{8}$ in. per ft. in depth, depending on the size of the hub. Torrey gives a thickness at the rim of $^{2}/_{3}$ the breadth and depth at the hub. The Halsey and Unwin formulas for size of arms give results which agree well with ordinary practice.

A simple rule for the proportions of arms is given in *Machinery's Handbook* which makes the width of the arm at the hub equal to 1.3 × shaft diameter for a wheel with six arms, and 1.2 × shaft diameter for a wheel with eight arms. The minimum cross-section at the rim is two-thirds the area at the hub. The strength of the arms should be at least three-quarters of the strength of the shaft in torsion.

**Hollow Arms.**—Hollow arms may be desirable in large fly-wheels. The design procedure is first to design the wheel with solid arms and then determine the equivalent cored section. Assume an arm of elliptical cross-section in which the major axis is 2 $B$ and the minor axis is $B$; the major axis of the cored section is 2 $b$ and the minor axis is $b$. Let $t$ be the thickness of the equivalent solid elliptical arm at the hub, and 2 $t$ be its width at the hub. Then by equating the moduli of resistance for solid and hollow elliptical sections, the relation is established

$$t^3 = (B^4 - b^4)/B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [32]$$

from which the dimensions of the arm and core can be calculated. A core box for an elliptical cross-section is difficult to make and is expensive. A core whose cross-section is a polygon of six sides closely approximating the outline of the ellipse will be equally satisfactory and will add but little to the weight of the arm. The cross-section of such a core can be determined by drawing lines at an angle of 60 deg. to the major axis of the core ellipse, at its extremities, to intersect the circumference of the ellipse. The points of intersection then are joined to the extremities of the minor axis to complete the polygon.

**CONSTRUCTIONAL DETAILS.**—The solid rim wheel, as shown by Prof. Benjamin's tests (see p. 11–20), is structurally the strongest. For fly-wheels built in sections, the common type of joint between the arms, *i.e.*, flanges, at the end of the joints with through bolts joining the flanges, is probably the worst construction that could be devised. (See Stresses in Fly-Wheels, p. 11–17.) Joints with an efficiency of 100% or more, that is, connections which are stronger than the rim itself, are obtained by means of links shrunk over lugs on the rim or tees sunk into the recess of the rim. Fly-wheel joints having an efficiency of over 100% are described by John Fritz (*Trans.* A.S.M.E., xx, p. 237, 1899) and by H. V. Haight (*Am. Mach.*, Feb. 28, 1907). The Haight construction is shown in Fig. 6. The rim section carrying the section necessary for the links joining the parts of the wheel, is cut away around the entire circumference of the rim. The rim proper, therefore, is continuous and is not weakened by the drilling of holes or otherwise cutting it away for the reception of fastening members. The net result is to give a rim of I-section which is better fitted to resist stresses than the ordinary shape of rim. Furthermore, by having the joints come at the ends of the arms, which are made double at the joints, as shown in Fig. 6, the rim is continuous between arms and the tendency to failure at the joint is minimized. Table 7 and Figs. 7–9 give the dimensions of shrink links and keyways for connections of this character.

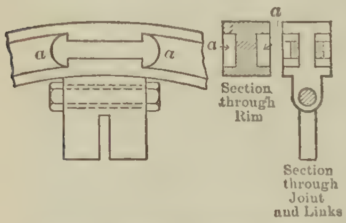


FIG. 6. Haight Fly-wheel Joint

Various designs of fly-wheel have been made for different purposes. In one type of engine fly-wheel, the rim section is cast integral with the arms. The rim section is divided into two parts on a plane perpendicular to the axis of the wheel, each part having its own set of arms. The hub is cast separately and the two halves of the rim section and the hub are fastened together by through bolts through the rim and the hub. Another type, for

**Table 7.—Dimensions of Shrink Links and Keyways**

(General Electric Company Practice)

All dimensions in inches

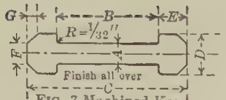


FIG. 7 Machined Key

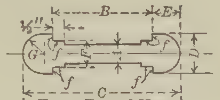


FIG. 8 Forged Key
Finished at *f*

| Machined Key, Fig. 7 | | | | | | | Forged Key, Fig. 8 | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| *A* | *B* | *C* | *D* | *E* | *F* | *G* | *A* | *B* | *C* | *D* | *E* | *F* | *G* |
| 31/32 | 4.995 | 7 1/2 | 1 7/8 | 1.252 | 1 7/16 | 3/8 | 15/16 | 4.995 | 7 1/2 | 1 7/8 | 1.252 | 31/32 | 15/16 |
| 1 15/32 | 5.994 | 9 3/4 | 2 3/4 | 1.878 | 2 1/4 | 5/8 | 1 7/16 | 5.994 | 9 3/4 | 2 3/4 | 1.878 | 1 15/32 | 1 3/8 |
| 1 31/32 | 7.992 | 13 | 3 5/8 | 2.504 | 2 7/8 | 15/16 | 1 15/16 | 7.992 | 13 | 3 5/8 | 2.504 | 1 31/32 | 1 13/16 |
| 2 7/16 | 9.990 | 16 1/4 | 4 1/2 | 3.130 | 3 9/16 | 1 3/16 | 2 7/16 | 9.990 | 16 1/4 | 4 1/2 | 3.130 | 2 15/32 | 2 1/4 |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | 2 7/8 | 9.875 | 18 | 5 3/8 | 4.062 | 2 29/32 | 2 11/16 |

| Keyway, Fig. 9 | | | | | |
|---|---|---|---|---|---|
| *H* | *J* | *K* | *L* | *M* | *N* |
| 1 | 5.000 | 8 3/4 | 2 | 6 3/4 | 1 |
| 1 1/2 | 6.000 | 11 5/16 | 2 7/8 | 8 7/16 | 1 7/16 |
| 2 | 8.000 | 14 7/8 | 3 3/4 | 11 1/8 | 1 7/8 |
| 2 1/2 | 10.000 | 18 1/2 | 4 5/8 | 13 7/8 | 2 5/16 |
| 2 15/16 | 9.885 | 20 1/4 | 5 1/2 | 14 3/4 | 2 3/4 |

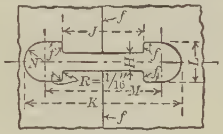


FIG. 9 Keyway
Finished at *f*

a rim speed of 15,000 ft. per min., has been built with a rim formed of laminated plates. A cast-steel spider with 12 double arms had dovetail notches cut in its periphery. The laminated plates, cut in arcs of 30 deg., were machined on the inside circumference to fit the notches on the spider, and assembled with broken joints. Side cover plates, machined with a groove into which the edge of the spider rim fits, were then placed outside of the laminated plates, and the entire assembly bolted together with 10 turned and fitted through bolts in each pair of side plates. The wheel was 13 ft. outside diameter, 2 ft. 2 in. face, and weighed about 100,000 lb.

Band saw fly-wheels usually run at rim velocities of 10,000 ft. per min. and may at times run even higher. The Allis-Chalmers Co. uses a construction in which the rim is of V-section and the arms are arranged diagonally, passing from one side of the hub to the opposite side of the rim, alternate arms being staggered with respect to each other. Another construction, largely used in band saw wheels, uses spokes composed of steel bars, either round, or square and set diagonal to the plane of rotation of the wheel, which are cast into the wheel and hub. Two rows of spokes are used, alternate spokes extending from the central plane of the rim to the opposite sides of the hub. The rim is poured first, and after it has cooled the hub is poured. Shrinkage strains in the rim that might pull the hub out of line thus are avoided.

For descriptions and illustrations of the construction details of a great variety of fly-wheels, including the above, for many different services, see Halsey's Handbook for Machine Designers.

**WIRE-WOUND FLY-WHEELS** have been used where very high rim speeds were necessary. Fly-wheels used at the Mannesmann Tube Works in Hungary, designed for a tangential velocity of over 250 ft. per sec., consisted of a cast-iron hub to which two steel discs, 20 ft. diameter, were bolted. Around the circumference of the wheel thus formed 70 tons of No. 5 wire were wound under a tension of 50 lb. In the Mannesmann Works at Landore, Wales, such a wheel makes 240 r.p.m. corresponding to a tangential velocity of 15,080 ft. or 2.85 miles per minute.

A wire-wound fly-wheel designed by George Smith Morison of Melbourne, Victoria, Australia, was built for the mine of the Great Southern Consols Gold Mining Co. at Rutherglen, Victoria, in 1910. The wheel was the center unit of a direct-coupled motor-generator set supplying current to the two 100-Hp. motors of the mine hoisting plant. The fly-wheel comprised a cast-iron center hub, 2 ft. diam. by 10 in. face, to each cheek of which were bolted circular boiler steel plates 6 ft. 3 in. diam., and 3/4 in. thick. The plates further were bolted together about 14 in. inside of their periphery by a circular gallery of bolts over which were fitted 10 in. separator tubes. A ribbon of steel 10 in. wide, 1/16 in. thick was fitted over the bolts. On this steel flooring, 13 miles of No. 8 gage steel wire was wound under a tension of 120 lb. The wheel weighed about 5 3/4 tons, and it ran at a normal peripheral velocity of about 20,000 ft. per min.

**ROTATING DISCS.**—Fly-wheels in the form of a disc, or those in which the depth of the rim is so great as compared to the rim that they approximate a disc form, cannot be designed by the methods used for the conventional type of fly-wheel. The complete determination of the stresses requires an involved mathematical analysis, which is complicated by the lack of experimental data. Kimball and Barr (Elements of Machine Design) give the following equations for the radial and tangential stresses in a rapidly rotating plain disc: Let $R$, $r$ = respectively, outer and inner radii of disc, in.; $r_1$ = radius at any point, in.; $S_t$, $S_r$ = respectively, tangential and radial stresses at any radius $r_1$; $w$ = weight of material, lb. per cu. in.; $n$ = rev. per min.; $\lambda$ = Poisson's ratio = 0.3 for steel = 0.25 for cast iron. $C$ = a constant = 0.000 003 55. Then for a solid disc,

$$S_t = C\,w\,n^2\{(3+\lambda)R^2 - (1+3\lambda)r_1^2\} \quad [33]$$

$$S_r = C\,w\,n^2\{(3+\lambda)(R^2 - r_1^2)\} \quad [34]$$

For a disc with a hole at the center,

$$S_t = C\,w\,n^2[(3+\lambda)\{R^2 + r^2 + (R^2r^2/r_1^2)\} - (1+3\lambda)r_1^2] \quad [35]$$

$$S_r = C\,w\,n^2[(3+\lambda)\{R^2 + r^2 - (R^2r^2/r_1^2)\}] \quad [36]$$

By taking $r$ so small that $r^2$ is a negligible value, it can be shown that the tangential stress at the surface of the bore is twice that obtained by putting $r = 0$ in equation [33]. This shows that even a small hole at the center of a rotating disc will greatly increase the tangential stress in the disc.

Rotating discs in which the stresses are to be uniform at all points, as steam turbine discs, will be of varying cross-section and thicker at the hub than at the periphery. The design of such discs involves complicated mathematical analysis. For one method of analysis, as used for steam turbine discs, see Kent's Mechanical Engineers' Handbook—Power, p. 8–26, forming Vol. 2 of this series.

## Section 12

# KEYS, COTTERS, PINS, TAPERS AND FITS

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KEYS

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# KEYS, COTTERS, PINS, TAPERS AND FITS

## KEYS

Keys are used to prevent relative movement of rotating members and the shafts or spindles on which they are mounted. They are of various forms, depending on the class of service in which they are to be used.

### 1. SUNK KEYS

The sunk key, *i.e.*, a key which is fitted in a groove or keyseat in the shaft, and which projects into a keyway in the bore of the other member, is the key in most common use. Sunk keys are made in various forms, as square, flat, tapered, etc. Square and flat keys preferably should be fitted on all four sides.

A key fitting tight on top and bottom of the keyway drives partly by friction. If fitted only on the sides of the keyway it exerts a prying action on the hub and shaft, and is subjected to severe bending and shearing stresses.

Taper keys (Fig. 4) are tapered on the upper surface only, *i.e.*, on the surface projecting into the hub. A corresponding taper is cut in the keyway in the hub. When the key is driven, hub and shaft are drawn together on the side opposite the key. A frictional resistance thus is established which tends to prevent axial movement of the hub on the shaft.

If the key is more tightly fitted at one end than at the other, or if the bore of the hub is tapering, the hub may be so drawn up that its axis is at an angle to the axis of the shaft. The Woodruff key (Fig. 5) adjusts itself to taper in the hub and thus avoids misalignment of shaft and hub.

**PROPORTIONS OF KEYS.**—A large number of formulas have been devised for the proportions of flat and square keys. The more important of them are the following, in which $W$ = width of key; $T$ = thickness of key; $L$ = length of key; $D$ = shaft diameter; all in inches. $t$ = taper, in. per ft.

*Machinery's Handbook:* $W = 1/4\,D$; $T = 1/6\,D$; $L = 1\tfrac{1}{2}\,D$ . . . . [1]

" " $W = 3/16\,D + 1/16$; $T = 1/8\,D + 1/16$; $L = 0.3\,D^2/T$. [2]

F. A. Halsey ($D$ = up to 4 in.),

$W = 1/4\,D$; $T = 5/8\,W$ to $3/8\,W$; $L = 1.5\,D$; $t = 1/8$ . . . [3]

" ($D$ = 4 to 6 in.),

$W = 1/4\,D - 1/8$; $T = 5/8\,W$ to $3/4\,W$; $L = 1.5\,D$; $t = 1/8$ . [4]

" ($D$ = over 6 in.),

$W = 1/4\,D - 1/4$; $T = 5/8\,W$ to $3/4\,W$; $L = 1.5\,D$; $t = 1/8$ . [5]

W. C. Unwin ($D$ over 6 in.), $W = 1/4\,D + 1/8$; $T = 1/8\,D + 1/8$ . . . . . . [6]

Formula [6] gives excessive dimensions for pulleys or shafts keyed to large shafts and transmitting small amounts of power. For such cases take $D = \sqrt[3]{100\ \text{Hp.}/N}$ or $D = \sqrt{PR/630}$, where Hp. = horsepower transmitted by pulley, $N$ = rev. per min., $P$ = force acting at circumference, lb., $R$ = radius of pulley, in.

Carl G. Barth developed in 1922 a formula based on mathematical reasoning which gives a more consistent relation of the key to the size of the shaft than previous formulas, and which is recommended for general use. This formula (see Fig. 1 for notation) is

$$t = 0.15\,D\,(D + 2)/(D + 1) \qquad [7]$$

This is represented graphically in Fig. 1, which shows the size of key for each size of shaft. The upper and lower curves show the limiting size of shaft for each commercial size of key. Barth's values are given in Table 7.

**Depth of Key Seats.**—The depth of a flat or square key is equally divided between the shaft and the hub. The depth to which a milling cutter is sunk into the shaft in milling a keyway is equal to one-half the depth of the key plus the height of the arc projecting above the intersection of the side of the keyway with the circumference of the shaft. This height can be calculated from the formula

$$H = R - \sqrt{R^2 - (1/2\,W)^2} \qquad [8]$$

where $R$ = radius of shaft, $H$ = height of arc and $W$ = width of key.

**FORMS OF KEYS.—Standard Stock Keys.**—The American Standards Assoc. has established standard sizes for parallel and flat stock keys, taper, square and flat stock keys, and square and flat taper gib-head stock keys. See A.S.A. Standard B17.1–1934. Dimensions of these keys are given in Tables 1 to 4.

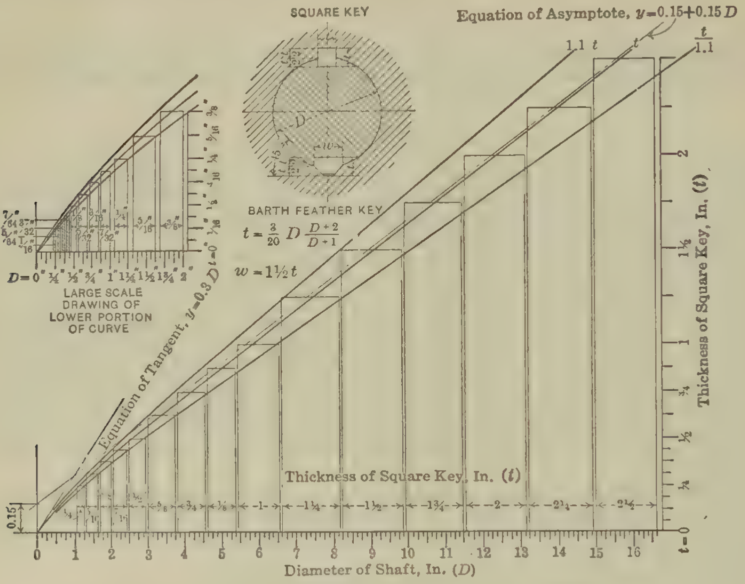


FIG. 1. Graphic Representation of Barth Key Formula

### Table 1.—Dimensions of Plain Parallel Flat Stock Keys

All dimensions in inches. See Fig. 2

| Shaft Diameter | Width $W$* | Thickness $H$* | Bottom of Keyseat to Opposite Side of Shaft $T$ | Shaft Diameter | Width $W$* | Thickness $H$* | Bottom of Keyseat to Opposite Side of Shaft $T$ | Shaft Diameter | Width $W$* | Thickness $H$* | Bottom of Keyseat to Opposite Side of Shaft $T$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | 1/8 | 3/32 | 0.455 | 1 3/4 | 3/8 | 1/4 | 1.605 | 3 3/8 | 7/8 | 5/8 | 3.005 |
| 9/16 | 1/8 | 3/32 | .509 | 1 13/16 | 1/2 | 3/8 | 1.590 | 3 7/16 | 7/8 | 5/8 | 3.069 |
| 5/8 | 3/16 | 1/8 | .548 | 1 7/8 | 1/2 | 3/8 | 1.654 | 3 1/2 | 7/8 | 5/8 | 3.132 |
| 11/16 | 3/16 | 1/8 | .612 | 1 15/16 | 1/2 | 3/8 | 1.717 | 3 5/8 | 7/8 | 5/8 | 3.259 |
| 3/4 | 3/16 | 1/8 | .676 | 2 | 1/2 | 3/8 | 1.781 | 3 3/4 | 7/8 | 5/8 | 3.386 |
| 13/16 | 3/16 | 1/8 | .739 | 2 1/16 | 1/2 | 3/8 | 1.843 | 3 7/8 | 1 | 3/4 | 3.434 |
| 7/8 | 3/16 | 1/8 | .802 | 2 1/8 | 1/2 | 3/8 | 1.908 | 3 15/16 | 1 | 3/4 | 3.498 |
| 15/16 | 1/4 | 3/16 | .827 | 2 3/16 | 1/2 | 3/8 | 1.971 | 4 | 1 | 3/4 | 3.562 |
| 1 | 1/4 | 3/16 | .890 | 2 1/4 | 1/2 | 3/8 | 2.034 | 4 1/4 | 1 | 3/4 | 3.815 |
| 1 1/16 | 1/4 | 3/16 | .954 | 2 5/16 | 5/8 | 7/16 | 2.051 | 4 7/16 | 1 | 3/4 | 4.006 |
| 1 1/8 | 1/4 | 3/16 | 1.017 | 2 3/8 | 5/8 | 7/16 | 2.114 | 4 1/2 | 1 | 3/4 | 4.069 |
| 1 3/16 | 1/4 | 3/16 | 1.081 | 2 7/16 | 5/8 | 7/16 | 2.178 | 4 3/4 | 1 1/4 | 7/8 | 4.229 |
| 1 1/4 | 1/4 | 3/16 | 1.144 | 2 1/2 | 5/8 | 7/16 | 2.242 | 4 15/16 | 1 1/4 | 7/8 | 4.420 |
| 1 5/16 | 5/16 | 1/4 | 1.169 | 2 5/8 | 5/8 | 7/16 | 2.368 | 5 | 1 1/4 | 7/8 | 4.483 |
| 1 3/8 | 5/16 | 1/4 | 1.232 | 2 3/4 | 5/8 | 7/16 | 2.495 | 5 1/4 | 1 1/4 | 7/8 | 4.733 |
| 1 7/16 | 3/8 | 1/4 | 1.288 | 2 7/8 | 3/4 | 1/2 | 2.575 | 5 7/16 | 1 1/4 | 7/8 | 4.927 |
| 1 1/2 | 3/8 | 1/4 | 1.351 | 2 15/16 | 3/4 | 1/2 | 2.639 | 5 1/2 | 1 1/4 | 7/8 | 4.991 |
| 1 9/16 | 3/8 | 1/4 | 1.415 | 3 | 3/4 | 1/2 | 2.702 | 5 3/4 | 1 1/2 | 1 | 5.150 |
| 1 5/8 | 3/8 | 1/4 | 1.478 | 3 1/8 | 3/4 | 1/2 | 2.829 | 5 15/16 | 1 1/2 | 1 | 5.341 |
| 1 11/16 | 3/8 | 1/4 | 1.542 | 3 1/4 | 3/4 | 1/2 | 2.956 | 6 | 1 1/2 | 1 | 5.405 |

* Tolerance on $W$ and $H$ = − 0.0020 in. for shafts 1/2 to 1 3/4 in. inclusive; −0.0025 in. for shafts 1 13/16 to 3 1/4 in. inclusive; −0.0030 in. for shafts 3 3/8 to 6 in. inclusive.

**Table 2.—Dimensions of Plain Parallel Square Stock Keys**

All dimensions in inches. See Fig. 2

| Shaft Diameter | Width and Thickness of Key $W$* | Bottom of Keyseat to Opposite Side of Shaft $S$ | Shaft Diameter | Width and Thickness of Key $W$* | Bottom of Keyseat to Opposite Side of Shaft $S$ | Shaft Diameter | Width and Thickness of Key $W$* | Bottom of Keyseat to Opposite Side of Shaft $S$ | Shaft Diameter | Width and Thickness of Key $W$* | Bottom of Keyseat to Opposite Side of Shaft $S$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | 1/8 | 0.430 | 1 7/16 | 3/8 | 1.225 | 2 3/8 | 5/8 | 2.021 | 3 7/8 | 1 | 3.309 |
| 9/16 | 1/8 | .493 | 1 1/2 | 3/8 | 1.289 | 2 7/16 | 5/8 | 2.084 | 3 15/16 | 1 | 3.373 |
| 5/8 | 3/16 | .517 | 1 9/16 | 3/8 | 1.352 | 2 1/2 | 5/8 | 2.148 | 4 | 1 | 3.437 |
| 11/16 | 3/16 | .581 | 1 5/8 | 3/8 | 1.416 | 2 5/8 | 5/8 | 2.275 | 4 1/4 | 1 | 3.690 |
| 3/4 | 3/16 | .644 | 1 11/16 | 3/8 | 1.479 | 2 3/4 | 5/8 | 2.402 | 4 7/16 | 1 | 3.881 |
| 13/16 | 3/16 | .708 | 1 3/4 | 3/8 | 1.542 | 2 7/8 | 3/4 | 2.450 | 4 1/2 | 1 | 3.944 |
| 7/8 | 3/16 | .771 | 1 13/16 | 1/2 | 1.527 | 2 15/16 | 3/4 | 2.514 | 4 3/4 | 1 1/4 | 4.042 |
| 15/16 | 1/4 | .796 | 1 7/8 | 1/2 | 1.591 | 3 | 3/4 | 2.577 | 4 15/16 | 1 1/4 | 4.232 |
| 1 | 1/4 | .859 | 1 15/16 | 1/2 | 1.655 | 3 1/8 | 3/4 | 2.704 | 5 | 1 1/4 | 4.296 |
| 1 1/16 | 1/4 | .923 | 2 | 1/2 | 1.718 | 3 1/4 | 3/4 | 2.831 | 5 1/4 | 1 1/4 | 4.550 |
| 1 1/8 | 1/4 | .986 | 2 1/16 | 1/2 | 1.782 | 3 3/8 | 7/8 | 2.880 | 5 7/16 | 1 1/4 | 4.740 |
| 1 3/16 | 1/4 | 1.049 | 2 1/8 | 1/2 | 1.845 | 3 7/16 | 7/8 | 2.944 | 5 1/2 | 1 1/4 | 4.803 |
| 1 1/4 | 1/4 | 1.112 | 2 3/16 | 1/2 | 1.909 | 3 1/2 | 7/8 | 3.007 | 5 3/4 | 1 1/2 | 4.900 |
| 1 5/16 | 5/16 | 1.137 | 2 1/4 | 1/2 | 1.972 | 3 5/8 | 7/8 | 3.140 | 5 15/16 | 1 1/2 | 5.091 |
| 1 3/8 | 5/16 | 1.201 | 2 5/16 | 5/8 | 1.957 | 3 3/4 | 7/8 | 3.261 | 6 | 1 1/2 | 5.155 |

* Tolerance on $W = -$ 0.0020 in. for shafts 1/2 to 1 3/4 in. inclusive; $-$0.0025 in. for shafts 1 13/16 to 3 1/4 in. inclusive; 0.0030 in. for shafts 3 3/8 to 6 in. inclusive.

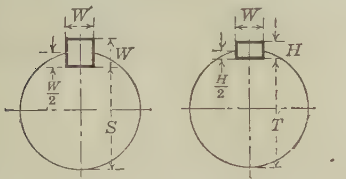


FIG. 2. Square and Flat Parallel Stock Key

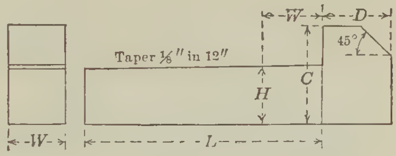


FIG. 3. Gib-head Taper Stock Key

**Table 3.—Dimensions of Square and Flat Gib-head Taper Stock Keys**

All dimensions in inches. See Fig. 3

| Shaft Diameter (Inclusive) | Square Type | | | | | Flat Type | | | | | Stock Length, $L$ | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Key | | Gib Head | | | Key | | Gib Head | | | | | |
| | Maximum Width $W$* | Height at Large End $H$* | Height, $C$† | Length, $D$ | Height, Edge of Chamfer $E$ | Maximum Width $W$* | Height at Large End $H$* | Height, $C$† | Length, $D$ | Height, Edge of Chamfer $E$ | Min. | Max. | Advancing by Increments of |
| 1/2 to 9/16 | 1/8 | 1/8 | 1/4 | 7/32 | 5/32 | 1/8 | 3/32 | 3/16 | 1/8 | 1/8 | 1/2 | 2 | 1/4 |
| 5/8 to 7/8 | 3/16 | 3/16 | 5/16 | 9/32 | 7/32 | 3/16 | 1/8 | 1/4 | 3/16 | 5/32 | 3/4 | 3 | 3/8 |
| 15/16 to 1 1/4 | 1/4 | 1/4 | 7/16 | 11/32 | 11/32 | 1/4 | 3/16 | 5/16 | 1/4 | 3/16 | 1 | 4 | 1/2 |
| 1 5/16 to 1 3/8 | 5/16 | 5/16 | 9/16 | 13/32 | 13/32 | 5/16 | 1/4 | 3/8 | 5/16 | 1/4 | 1 1/4 | 5 1/2 | 5/8‡ |
| 1 7/16 to 1 3/4 | 3/8 | 3/8 | 11/16 | 15/32 | 15/32 | 3/8 | 1/4 | 7/16 | 3/8 | 5/16 | 1 1/2 | 6 | 3/4 |
| 1 13/16 to 2 1/4 | 1/2 | 1/2 | 7/8 | 19/32 | 5/8 | 1/2 | 3/8 | 5/8 | 1/2 | 7/16 | 2 | 8 | 1 |
| 2 5/16 to 2 3/4 | 5/8 | 5/8 | 1 1/16 | 23/32 | 3/4 | 5/8 | 7/16 | 3/4 | 5/8 | 1/2 | 2 1/2 | 10 | 1 1/4 |
| 2 7/8 to 3 1/4 | 3/4 | 3/4 | 1 1/4 | 7/8 | 7/8 | 3/4 | 1/2 | 7/8 | 3/4 | 5/8 | 3 | 12 | 1 1/2 |
| 3 3/8 to 3 3/4 | 7/8 | 7/8 | 1 1/2 | 1 | 1 | 7/8 | 5/8 | 1 1/16 | 7/8 | 3/4 | 3 1/2 | 14 | 1 3/4 |
| 3 7/8 to 4 1/2 | 1 | 1 | 1 3/4 | 1 3/16 | 1 3/16 | 1 | 3/4 | 1 1/4 | 1 | 13/16 | 4 | 16 | 2 |
| 4 3/4 to 5 1/2 | 1 1/4 | 1 1/4 | 2 | 1 7/16 | 1 7/16 | 1 1/4 | 7/8 | 1 1/2 | 1 1/4 | 1 | 5 | 20 | 2 1/2 |
| 5 3/4 to 6 | 1 1/2 | 1 1/2 | 2 1/2 | 1 3/4 | 1 3/4 | 1 1/2 | 1 | 1 3/4 | 1 1/2 | 1 1/4 | 6 | 24 | 3 |

* Tolerance: $-$0.0020 on width, and $+$0.0020 on height for shafts 1/2 to 1 3/4 in. diam: $-$0.0025 on width and $+$0.0025 on height for shafts 1 13/16 to 3 1/4 in. diam.; $-$0.0030 on width and $+$0.0030 on height for shafts 3 3/8 to 6 in. diam.

† This height of the key is measured at the distance $W$, equal to the width of key, from the gib head.

‡ Final increment 1 1/8 in.

**The Woodruff Key** is extensively used in machine tools, and in the automotive industry. The keyslot is cut in the shaft with a milling cutter. The hub is forced on the shaft over

**Table 4.—Dimensions of Square and Flat Taper Stock Keys**

All dimensions in inches. See Fig. 4

| Shaft Diameter (Inclusive) | Square Type | | Flat Type | | Tolerance | | Stock Length, $L$ | | |
|---|---|---|---|---|---|---|---|---|---|
| | Maximum Width $W$ | Height at Large End $H$* | Maximum Width $W$ | Height at Large End $H$* | On Width | On Height | Minimum | Maximum | Advancing by Increments of |
| 1/2 to 9/16 | 1/8 | 1/8 | 1/8 | 3/32 | −0.0020 | +0.0020 | 1/2 | 2 | 1/4 |
| 5/8 to 7/8 | 3/16 | 3/16 | 3/16 | 1/8 | −0.0020 | +0.0020 | 3/4 | 3 | 3/8 |
| 15/16 to 1 1/4 | 1/4 | 1/4 | 1/4 | 3/16 | −0.0020 | +0.0020 | 1 | 4 | 1/2 |
| 1 5/16 to 1 3/8 | 5/16 | 5/16 | 5/16 | 1/4 | −0.0020 | +0.0020 | 1 1/4 | 5 1/4 | 5/8† |
| 1 7/16 to 1 3/4 | 3/8 | 3/8 | 3/8 | 1/4 | −0.0020 | +0.0020 | 1 1/2 | 6 | 3/4 |
| 1 13/16 to 2 1/4 | 1/2 | 1/2 | 1/2 | 3/8 | −0.0025 | +0.0025 | 2 | 8 | 1 |
| 2 5/16 to 2 3/4 | 5/8 | 5/8 | 5/8 | 7/16 | −0.0025 | +0.0025 | 2 1/2 | 10 | 1 1/4 |
| 2 7/8 to 3 1/4 | 3/4 | 3/4 | 3/4 | 1/2 | −0.0025 | +0.0025 | 3 | 12 | 1 1/2 |
| 3 3/8 to 3 3/4 | 7/8 | 7/8 | 7/8 | 5/8 | −0.0030 | +0.0030 | 3 1/2 | 14 | 1 3/4 |
| 3 7/8 to 4 1/2 | 1 | 1 | 1 | 3/4 | −0.0030 | +0.0030 | 4 | 16 | 2 |
| 4 3/4 to 5 1/2 | 1 1/4 | 1 1/4 | 1 1/4 | 7/8 | −0.0030 | +0.0030 | 5 | 20 | 2 1/2 |
| 5 3/4 to 6 | 1 1/2 | 1 1/2 | 1 1/2 | 1 | −0.0030 | +0.0030 | 6 | 24 | 3 |

* This height of the key is measured at the distance $W$, equal to the width of the key, from the large end.

† 4 1/2 in. length instead of 4 3/8 in.

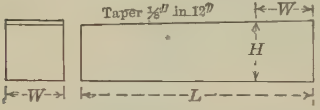


FIG. 4. Plain Taper Stock Key

the key. Dimensions of standard Woodruff keys are given in Table 5. The particular size of key to be used for various sizes of shaft may be obtained from Table 6. The dimensions of $J$, $K$, $L$ and $M$, Fig. 5, which are not given in Table 5, can be computed by means of the following formulas, in which $d$ = diameter of the shaft, other factors being as shown in Fig. 5:

Versed sine $J = \frac{1}{2} d - \sqrt{(d^2 - A^2)/4}$ . . . . . . . . . . . . . [9]

Bottom of keyseat to opposite side of shaft, $K = (d - J - E)$ . . . . . . [10]

Top of key to opposite side of shaft, $L = (F + E + K)$. . . . . . . . [11]

Bottom of keyway to opposite side of bore, $M = (d - J + G)$ . . . . . [12]

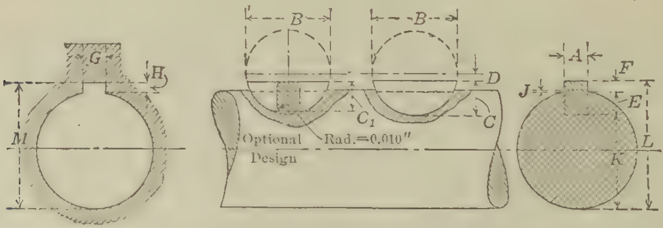


FIG. 5. Standard Woodruff Key and Keyseat

**The Barth Key.**—The key shown in Fig. 6 was devised by Carl G. Barth to combine the advantages of the Lewis key with those of the ordinary rectangular key. The Barth key is rectangular, with one-half of both sides bevelled at 45 deg. The key does not need to fit tightly, as pressure tends to drive it into its seat. There is no tendency to turn it, and the only stress to which it is subject is compression. This key has been used in many cases as a feather to replace rectangular feather keys which have given trouble. It has found wide application as a feather key in drill sockets and drill shanks, reamers, etc., which are commonly driven with a tang. Dimensions are given in Table 7.

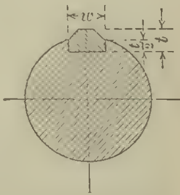


FIG. 6. Barth Key

Reducing sockets for drill presses are fitted with a Barth key dovetailed inside and a similar keyway on the outside. No. 1 Morse taper shank has a keyway for No. 1 Barth key and fits into a No. 1 reducing socket. No. 2 shank has No. 2 Morse taper and a keyway for No. 2 Barth key, etc. Dimensions of the various sizes of the dovetailed Barth key as used for sockets are shown in Table 8. The Barth key has been adapted to a complete line of standard taper sockets, shanks, driving keys, holdback keys, drifts, adapters, and reducers at the Watertown

**Table 5.—Dimensions of Woodruff Keys and Keyways**

S.A.E. Standard. All dimensions in inches. See Fig. 5

| Key No. | Nominal Size | Dimensions A | | | B | C | $C_1$ | D | E | F | G | H |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Key | Key-slot | | Key | Key | Key | Key | Key-slot | Key | Key-way | Key-way |
| | | Max.[a] | Min. | Max. | Max.[a] | Max.[b] | Max.[c] | Nominal | Max.[d] | Basic[e] | Class[f] | Min.[g] |
| 204 | 1/16 × 1/2 | 0.0635 | 0.0615 | 0.0630 | 0.500 | 0.203 | 0.194 | 3/64 | 0.1718 | 0.0312 | 0.0635 | 0.0372 |
| 304 | 3/32 × 1/2 | .0948 | .0928 | .0943 | .500 | .203 | .194 | 3/64 | .1561 | .0469 | .0948 | .0529 |
| 404 | 1/8 × 1/2 | .1260 | .1240 | .1255 | .500 | .203 | .194 | 3/64 | .1405 | .0625 | .1260 | .0685 |
| 305 | 3/32 × 5/8 | .0948 | .0928 | .0943 | .625 | .250 | .240 | 1/16 | .2031 | .0469 | .0948 | .0529 |
| 405 | 1/8 × 5/8 | .1260 | .1240 | .1255 | .625 | .250 | .240 | 1/16 | .1875 | .0625 | .1260 | .0685 |
| 505 | 5/32 × 5/8 | .1573 | .1553 | .1568 | .625 | .250 | .240 | 1/16 | .1719 | .0781 | .1573 | .0841 |
| 406 | 1/8 × 3/4 | .1260 | .1240 | .1255 | .750 | .313 | .303 | 1/16 | .2505 | .0625 | .1260 | .0685 |
| 506 | 5/32 × 3/4 | .1573 | .1553 | .1568 | .750 | .313 | .303 | 1/16 | .2349 | .0781 | .1573 | .0841 |
| 606 | 3/16 × 3/4 | .1885 | .1863 | .1880 | .750 | .313 | .303 | 1/16 | .2193 | .0937 | .1885 | .0997 |
| 507 | 5/32 × 7/8 | .1573 | .1553 | .1568 | .875 | .375 | .365 | 1/16 | .2969 | .0781 | .1573 | .0841 |
| 607 | 3/16 × 7/8 | .1885 | .1863 | .1880 | .875 | .375 | .365 | 1/16 | .2813 | .0937 | .1885 | .0997 |
| 807 | 1/4 × 7/8 | .2510 | .2487 | .2505 | .875 | .375 | .365 | 1/16 | .2500 | .1250 | .2510 | .1310 |
| 608 | 3/16 × 1 | .1885 | .1863 | .1880 | 1.000 | .438 | .428 | 1/16 | .3443 | .0937 | .1885 | .0997 |
| 808 | 1/4 × 1 | .2510 | .2487 | .2505 | 1.000 | .438 | .428 | 1/16 | .3130 | .1250 | .2510 | .1310 |
| 1008 | 5/16 × 1 | .3135 | .3111 | .3130 | 1.000 | .438 | .428 | 1/16 | .2818 | .1562 | .3135 | .1622 |
| 609 | 3/16 × 1 1/8 | .1885 | .1863 | .1880 | 1.125 | .484 | .475 | 5/64 | .3903 | .0937 | .1885 | .0997 |
| 809 | 1/4 × 1 1/8 | .2510 | .2487 | .2505 | 1.125 | .484 | .475 | 5/64 | .3590 | .1250 | .2510 | .1310 |
| 1009 | 5/16 × 1 1/8 | .3135 | .3111 | .3130 | 1.125 | .484 | .475 | 5/64 | .3278 | .1562 | .3135 | .1622 |
| 810 | 1/4 × 1 1/4 | .2510 | .2487 | .2505 | 1.250 | .547 | .537 | 5/64 | .4220 | .1250 | .2510 | .1310 |
| 1010 | 5/16 × 1 1/4 | .3135 | .3111 | .3130 | 1.250 | .547 | .537 | 5/64 | .3908 | .1562 | .3135 | .1622 |
| 1210 | 3/8 × 1 1/4 | .3760 | .3735 | .3755 | 1.250 | .547 | .537 | 5/64 | .3595 | .1875 | .3760 | .1935 |
| 811 | 1/4 × 1 3/8 | .2510 | .2487 | .2505 | 1.375 | .594 | .584 | 3/32 | .4690 | .1250 | .2510 | .1310 |
| 1011 | 5/16 × 1 3/8 | .3135 | .3111 | .3130 | 1.375 | .594 | .584 | 3/32 | .4378 | .1562 | .3135 | .1622 |
| 1211 | 3/8 × 1 3/8 | .3760 | .3735 | .3755 | 1.375 | .594 | .584 | 3/32 | .4065 | .1875 | .3760 | .1935 |
| 812 | 1/4 × 1 1/2 | .2510 | .2487 | .2505 | 1.500 | .641 | .631 | 7/64 | .5160 | .1250 | .2510 | .1310 |
| 1012 | 5/16 × 1 1/2 | .3135 | .3111 | .3130 | 1.500 | .641 | .631 | 7/64 | .4248 | .1562 | .3135 | .1622 |
| 1212 | 3/8 × 1 1/2 | .3760 | .3735 | .3755 | 1.500 | .641 | .631 | 7/64 | .4535 | .1875 | .3760 | .1935 |

[a] Tolerance allowed of −0.0010 in. from maximum.
[b] Tolerance allowed of −0.0005 in. from maximum.
[c] Tolerance allowed of −0.0006 in. from maximum.
[d] Tolerance allowed of −0.0050 in. from maximum.
[e] Tolerance allowed of ±0.0050 in. from basic.
[f] Two classes are listed. Class 1, the minimum width of keyway is the maximum width of key, minus 0.000, plus 0.005, with a tolerance of +0.0050 above minimum width. Class 2, minimum width of keyway is the maximum width of key minus 0.000, plus 0.002, with a tolerance of +0.0020 above minimum width.
[g] Tolerance allowed of +0.0050 above minimum.

**Table 6.—Woodruff Key Sizes for Different Shaft Diameters**

| Shaft Diam., in. | Key No. See Table 5 | Shaft Diam., in. | Key No. See Table 5 | Shaft Diam., in. | Key No. See Table 5 |
|---|---|---|---|---|---|
| 5/16 –3/8 | 204 | 7/8 – 15/16 | 505, 506, 507 | 1 3/8 –1 7/16 | 608, 609, 810 |
| 7/16 –1/2 | 304, 305 | 1 | 606, 607, 608 | 1 1/2 –1 5/8 | 808, 809, 810, 812 |
| 9/16 –5/8 | 404, 405 | 1 1/16–1 1/8 | 606, 607, 608, 609 | 1 11/16–1 3/4 | 809, 810, 812 |
| 11/16–3/4 | 404, 405, 406 | 1 3/16 | 607, 608, 609 | 1 13/16–2 | 1011, 1012 |
| 13/16 | 505, 506 | 1 1/4 –1 5/16 | 607, 608, 609, 810 | 2 1/16 –2 1/2 | 1012 |

**Table 7.—Dimensions of Square Keys and Feather Keys by Barth's Formula.** (See Fig. 6)

| Diameter of Shaft, in. | | Barth Key, Square Key, $t$, in. | Barth Key, $w$, in. | Diameter of Shaft, in. | | Barth Key, Square Key, $t$, in. | Barth Key, $w$, in. | Diameter of Shaft, in. | | Barth Key, Square Key, $t$, in. | Barth Key, $w$, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Min. | Max. | | | Min. | Max. | | | Min. | Max. | | |
| 7/32 | 1/4 | 1/16 | 3/32 | 1 1/16 | 1 1/4 | 1/4 | 3/8 | 5 7/16 | 6 9/16 | 1 | 1 1/2 |
| 9/32 | 5/16 | 5/64 | 1/8 | 1 5/16 | 1 5/8 | 5/16 | 1/2 | 6 5/8 | 8 3/16 | 1 1/4 | 1 7/8 |
| 11/32 | 3/8 | 3/32 | 9/64 | 1 11/16 | 2 | 3/8 | 9/16 | 8 1/4 | 9 7/8 | 1 1/2 | 2 1/4 |
| 13/32 | 7/16 | 7/64 | 11/64 | 2 1/16 | 2 7/16 | 7/16 | 21/32 | 9 15/16 | 11 1/2 | 1 3/4 | 2 5/8 |
| 15/32 | 17/32 | 1/8 | 3/16 | 2 1/2 | 2 15/16 | 1/2 | 3/4 | 11 9/16 | 13 3/16 | 2 | 3 |
| 9/16 | 11/16 | 5/32 | 1/4 | 3 | 3 3/4 | 5/8 | 15/16 | 13 1/4 | 14 15/16 | 2 1/4 | 3 3/8 |
| 23/32 | 27/32 | 3/16 | 9/32 | 3 13/16 | 4 9/16 | 3/4 | 1 1/8 | 15 | 16 5/8 | 2 1/2 | 3 3/4 |
| 7/8 | 1 | 7/32 | 11/32 | 4 5/8 | 5 3/8 | 7/8 | 1 5/16 | ........ | ........ | ........ | ...... |

Arsenal. The standards, which cover both Brown & Sharpe and Morse tapers are given in *Am. Mach.*, Dec. 24, 1914.

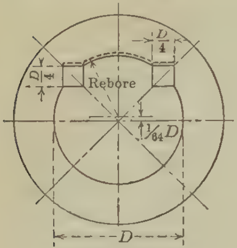


FIG. 7. Kennedy Key

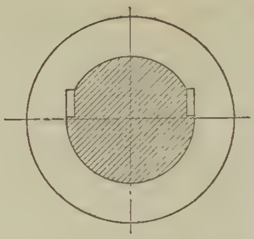


FIG. 8. Lewis Key

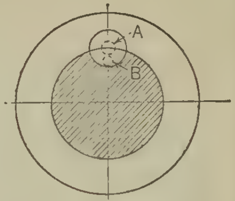


FIG. 9. Nordberg Key

**The Kennedy Key.**—The Kennedy key, largely used in rolling mill work, is shown in Fig. 7. In these keys, $w = t = {}^1/_4\,D$. They are tapered $^1/_8$ in. per foot on top, while the sides are a neat fit. The keys are so set in the shaft that diagonals through them intersect at the axis of the shaft. If in assembling, the hub is to be slid along the shaft for a considerable distance, the hub is bored for a press fit and then is rebored eccentrically about $^1/_{64}\,D$ off center. The keyways are cut in the eccentric side. General practice is to use single keys for diameters up to and including 6 in. where the torque is constant and the power transmitted always in one direction. For shafts above 6 in. diameter double keys should be used, and if the torque is intermittent and in alternate directions, double keys should be used down to shaft diameters of 4 in.

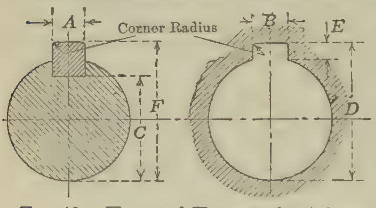


FIG. 10. Keys and Keyways for Arbors and Milling Cutters

Fig. 8 shows a key devised by Wilfred Lewis, which is similar in principle to the Kennedy key and which is used for lighter service, as machine tool work. It is subject to compression only but it has the disadvantage of being expensive and difficult to fit.

**The Nordberg Key.**—The Nordberg Mfg. Co. has adopted for the ends of shafts the round keys shown in Fig. 9. The advantages of this key are: No tendency toward deformation; they are a driven fit in the direction of the shear; they are always in true shear and are cheaper than the square key. In manufacturing, a hole *A* is drilled in the joint, and next a hole *B* as large as the size of the keyway will admit is drilled in the shaft in order to avoid the tendency of the drill used for drilling the keyway to size to crowd into

**Table 8.—Dimensions of Dovetailed Barth Keys**

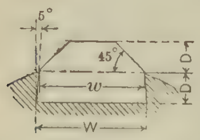
|  | No. of Barth Key. | No. of Morse Taper in Which Used | *w*, in. | *W*, in. | *D*, in. |
|---|---|---|---|---|---|
| | 1 | 1 | 1/8 | 0.132 | 5/128 |
| | 2 | 2 | 5/32 | 0.165 | 3/64 |
| | 3 | 3 | 3/16 | 0.199 | 1/16 |
| | 4 | 4 | 1/4 | 0.264 | 5/64 |
| | 5 | 5 | 5/16 | 0.329 | 3/32 |

**Table 9.—Dimensions of Nordberg Standard Round Keys.** (See Fig. 9)

| Diameter of Shaft, in. | Diameter of Reamer, in. | Cutting Length of Reamer, in. | Diameter of Shaft, in. | Diameter of Reamer, in. | Cutting Length of Reamer, in. | Diameter of Shaft, in. | Diameter of Reamer, in. | Cutting Length of Reamer, in. |
|---|---|---|---|---|---|---|---|---|
| 2 15/16–3 | 3/4 | 4 1/4 | 7<br>8<br>9 | 1 5/8 | 6 7/8 & 8 | 16<br>17<br>18 | 3 1/8 | 12 |
| 3 7/16 –3 1/2 | 7/8 | 4 1/2 | | | | | | |
| 3 7/8 –4 | 1 | 4 7/8 | | | | | | |
| 4 3/8 –4 1/2 | 1 1/8 | 5 | 10<br>11<br>12 | 2 | 10 1/4 | 19<br>20<br>21 | 3 11/16 | 13 |
| 5 | 1 1/4 | 4 5/8 | | | | | | |
| 5 1/2 | 1 3/8 | 4 7/8 | | | | | | |
| 6 | 1 1/2 | 6 1/8 | 13<br>14<br>15 | 2 9/16 | 12 | 22<br>23<br>24 | 4 1/4 | 14 1/4 |
| .............. | .......... | .......... | | | | | | |
| .............. | .......... | .......... | | | | | | |

**Table 10.—Keys and Keyways for Arbors and Milling Cutters**

A.S.A. B5C–1930. All dimensions in inches. See Fig. 10

| | | A | | B | | C | D | E | F | |
|---|---|---|---|---|---|---|---|---|---|---|
| Diameter of Arbor | Nominal Size, Square Key | Keyseat,* Max. | Key,† Max. | Keyway Max. | Keyway Min. | Max.† | Bottom of Keyway to Opposite Side of Bore, Min. | Nominal | Top of Key to Opposite Side of Shaft‡ | Corner Radius |
| 1/2 | 3/32 | 0.0947 | 0.0932 | 0.106 | 0.099 | 0.4531 | 0.5578 | 3/64 | 0.5468 | 0.020 |
| 5/8 | 1/8 | .126 | .1245 | .137 | .130 | .5625 | .6985 | 1/16 | .6875 | 1/32 |
| 3/4 | 1/8 | .126 | .1245 | .137 | .130 | .6875 | .8225 | 1/16 | .8125 | 1/32 |
| 7/8 | 1/8 | .126 | .1245 | .137 | .130 | .8125 | .9475 | 1/16 | .9375 | 1/32 |
| 1 | 1/4 | .251 | .2495 | .262 | .255 | .8438 | 1.104 | 3/32 | 1.094 | 3/64 |
| 1 1/4 | 5/16 | .3135 | .3120 | .325 | .318 | 1.063 | 1.385 | 1/8 | 1.375 | 1/16 |
| 1 1/2 | 3/8 | .376 | .3745 | .410 | .385 | 1.281 | 1.666 | 5/32 | 1.656 | 1/16 |
| 1 3/4 | 7/16 | .4385 | .4370 | .473 | .448 | 1.500 | 1.948 | 3/16 | 1.938 | 1/16 |
| 2 | 1/2 | .501 | .4995 | .535 | .510 | 1.687 | 2.198 | 3/16 | 2.188 | 1/16 |
| 2 1/2 | 5/8 | .626 | .6245 | .660 | .635 | 2.094 | 2.733 | 7/32 | 2.718 | 1/16 |
| 3 | 3/4 | .751 | .7495 | .785 | .760 | 2.500 | 3.265 | 1/4 | 3.250 | 3/32 |
| 3 1/2 | 7/8 | .876 | .8745 | .910 | .885 | 3.000 | 3.890 | 3/8 | 3.875 | 3/32 |
| 4 | 1 | 1.001 | .9995 | 1.035 | 1.010 | 3.375 | 4.390 | 3/8 | 4.375 | 3/32 |
| 4 1/2 | 1 1/8 | 1.126 | 1.1245 | 1.160 | 1.135 | 3.813 | 4.953 | 7/16 | 4.938 | 1/8 |
| 5 | 1 1/4 | 1.251 | 1.2495 | 1.285 | 1.260 | 4.250 | 5.515 | 1/2 | 5.500 | 1/8 |

* Tolerance allowed of −0.001 in. from maximum.
† Tolerance allowed of −0.005 in. from maximum.
‡ Tolerance allowed of −0.006 in. from maximum.

the soft cast iron. Dimensions are given in Table 9. In the table, the reamer diameters given are of the small end. The taper is $^1/_{16}$ in. per ft., measured on the diameter.

**Keys and Keyways for Arbors and Milling Cutters.**—Table 10 gives dimensions and tolerances of keys and keyways for milling cutters and arbors, as developed by the American Standards Association.

## 2. LIGHT KEYS

Where only small amounts of power are to be transmitted and it is impracticable or inadvisable to use a sunk key, a key driving on a flat, or a saddle key may be used. These two types of key drive mainly by friction.

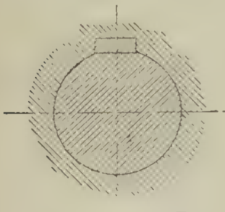


FIG. 11. Key on Flat

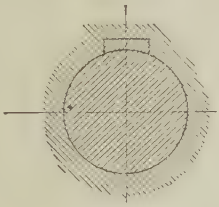


FIG. 12. Saddle Key

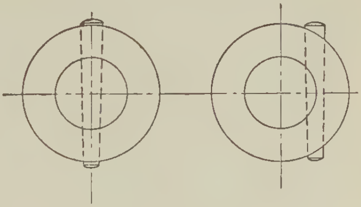


FIG. 13. Pin Keys

**KEY ON FLAT.**—If a small flat be milled or filed on the shaft, a flat key with a slight taper may be driven into a keyway in the hub opposite the flat on the shaft. See Fig. 11. This type of key is suitable for transmission of light powers, and has the advantage of ease of assembling. Such a key should not be subjected to sudden changes in torque, as it then may slip on the shaft and seriously damage it.

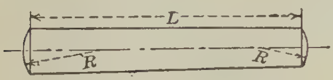


FIG. 14. Taper Pin

**SADDLE KEY.**—The saddle key, Fig. 12, depends entirely on friction to prevent motion of the hub relative to the shaft. The key is driven into a keyway in the hub, and the face of the key bearing on the shaft is made with a radius somewhat smaller than the radius of the shaft. The key should have a slight taper.

**PIN KEYS** are adapted to transmit power with light loads. Fig. 13 shows two methods of using taper pins as keys. Table 11 gives the dimensions of standard taper pins.

**FEATHER KEYS** are used when the hub must slide axially along the shaft, but at the same time rotate with it. The feather key is a sunk key. It either may be dove-

**Table 11.—Dimensions of Taper Pins**

Morse Twist Drill and Machine Co., New Bedford, Mass.

Taper 1/4 in. in 12 in. All dimensions in inches. See Fig. 14

| No. | Diameter, Large End * | Approximate Fractional Size | Radius, *R* | Length *L* | | Advancing by Increments of | No. | Diameter, Large End * | Approximate Fractional Size | Radius, *R* | Length *L* | | Advancing by Increments of |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Min. | Max. | | | | | | Min. | Max. | |
| 0 000 000 | 0.0625 | 1/16 | 1/16 | 1/4 | 3/4 | 1/8 | 5 | 0.289 | 19/64 | 7/32 | 3/4 | 4 | 1/4 |
| 000 000 | .078 | 5/64 | 1/16 | 1/4 | 1 | 1/8 | 6 | .341 | 11/32 | 7/32 | 3/4 | 5 | 1/4 |
| 00 000 | .094 | 3/32 | 1/16 | 1/2 | 1 | 1/8 | 7 | .409 | 13/32 | 1/4 | 1 | 5 | 1/4 |
| 0 000 | .109 | 7/64 | 1/16 | 1/2 | 1 | 1/8 | 8 | .492 | 1/2 | 3/8 | 1 1/4 | 5 | 1/4 |
| 000 | .125 | 1/8 | 1/16 | 1/2 | 1 | 1/8 | 9 | .591 | 19/32 | 1/2 | 1 1/2 | 6 | 1/4 |
| 00 | .141 | 9/64 | 1/16 | 1/2 | 2 1/2 | 1/4 | 10 | .706 | 23/32 | 5/8 | 1 1/2 | 6 | 1/4 |
| 0 | .156 | 5/32 | 5/64 | 1/2 | 3 | 1/4 | 11 | .860 | 55/64 | 7/8 | 2 | 9 | 1/2 |
| 1 | .172 | 11/64 | 3/32 | 1/2 | 3 | 1/4 | 12 | 1.032 | 1 1/32 | 1 1/16 | 2 1/2 | 9 | 1/2 |
| 2 | .193 | 3/16 | 7/64 | 3/4 | 3 1/2 | 1/4 | 13 | 1.241 | 1 1/4 | 1 1/4 | 3 | 11 | 1/2 |
| 3 | .219 | 7/32 | 5/32 | 3/4 | 3 1/2 | 1/4 | 14 | 1.523 | 1 33/64 | 1 1/2 | 4 | 13 | 1/2 |
| 4 | .250 | 1/4 | 3/16 | 3/4 | 4 | 1/4 | | | | | | | |

* Tolerance ±0.001 in.

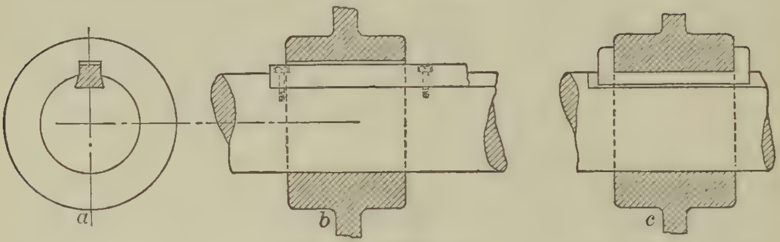


Fig. 15. Feather Keys

tailed into the shaft, or fastened in place by screws. Fig. 15 shows various forms. If the key is required to slide with the hub, the construction shown in Fig. 15 *c* may be used. Instead of feather keys, spline shafts now are largely used.

## 3. SPLINES AND SQUARE SHAFTS

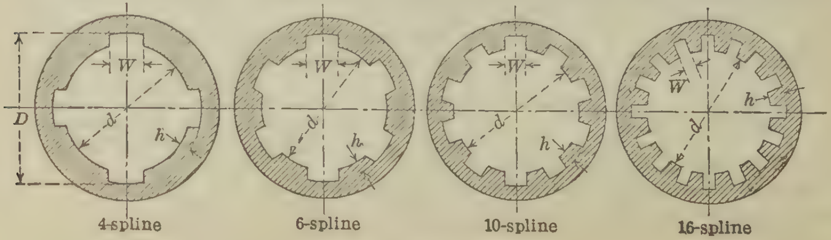
Instead of keys, spline shafts, or squared shafts may be used for the transmission of power from a shaft to a hub or *vice versa*. These applications are found in geared transmissions, machine tool drives, and in automotive work.

**MULTIPLE SPLINES** are, in effect, a series of parallel keys formed integral with the shaft, mating with corresponding grooves cut in the hub or fitting. They are particularly useful where the hub must slide axially on the shaft either under load or free from load. Spline fittings with four, six, ten and sixteen splines have been standardized by the S.A.E. Dimensions of these standardized fittings are given in Table 12 and Fig. 16. The relations between the internal diameter of the fitting and depth of spline are given in Table 13.

**Torque Capacity of Spline Fittings** may be determined by the formula

$$T = 1000\,N\,r\,h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [13]$$

where $N$ = number of splines; $r$ = mean radial distance from center of hole to center of spline; $h$ = depth of spline; all in inches. This formula is based on a pressure of 1000 lb. per sq. in. on the sides of the spline.



4-spline 6-spline 10-spline 16-spline

Fig. 16. Spline Fittings

### Table 12.—Dimensions and Torque Capacity of Spline Fittings

S.A.E. Standard. All dimensions in inches. See Fig. 16

| Nominal Diameter | For All Fits | | Permanent Fit | | To Slide When not Under Load | | To Slide When Under Load | |
|---|---|---|---|---|---|---|---|---|
| | D† max. | W‡ max. | d† | T* | d† | T* | d† | T* |
| | **Four-spline Fittings** | | | | | | | |
| 3/4 | 0.750 | 0.181 | 0.637 | 78 | 0.562 | 123 | ........ | ........ |
| 7/8 | 0.875 | .211 | .744 | 107 | .656 | 167 | ........ | ........ |
| 1 | 1.000 | .241 | .850 | 139 | .750 | 219 | ........ | ........ |
| 1 1/8 | 1.125 | .271 | .956 | 175 | .844 | 277 | ........ | ........ |
| 1 1/4 | 1.250 | .301 | 1.062 | 217 | .937 | 341 | ........ | ........ |
| 1 3/8 | 1.375 | .331 | 1.169 | 262 | 1.031 | 414 | ........ | ........ |
| 1 1/2 | 1.500 | .361 | 1.275 | 311 | 1.125 | 491 | ........ | ........ |
| 1 5/8 | 1.625 | .391 | 1.381 | 367 | 1.219 | 577 | ........ | ........ |
| 1 3/4 | 1.750 | .422 | 1.487 | 424 | 1.312 | 670 | ........ | ........ |
| 2 | 2.000 | .482 | 1.700 | 555 | 1.500 | 875 | ........ | ........ |
| 2 1/4 | 2.250 | .542 | 1.912 | 703 | 1.687 | 1106 | ........ | ........ |
| 2 1/2 | 2.500 | .602 | 2.125 | 865 | 1.875 | 1365 | ........ | ........ |
| 3 | 3.000 | .723 | 2.550 | 1249 | 2.250 | 1969 | ........ | ........ |
| | **Six-spline Fittings** | | | | | | | |
| 3/4 | 0.750 | 0.188 | 0.675 | 80 | 0.638 | 117 | 0.600 | 152 |
| 7/8 | 0.875 | .219 | 0.788 | 109 | .744 | 159 | .700 | 207 |
| 1 | 1.000 | .250 | 0.900 | 143 | .850 | 208 | .800 | 270 |
| 1 1/8 | 1.125 | .281 | 1.013 | 180 | .956 | 263 | .900 | 342 |
| 1 1/4 | 1.250 | .313 | 1.125 | 223 | 1.063 | 325 | 1.000 | 421 |
| 1 3/8 | 1.375 | .344 | 1.238 | 269 | 1.169 | 393 | 1.100 | 510 |
| 1 1/2 | 1.500 | .375 | 1.350 | 321 | 1.275 | 468 | 1.200 | 608 |
| 1 5/8 | 1.625 | .406 | 1.463 | 376 | 1.381 | 550 | 1.300 | 713 |
| 1 3/4 | 1.750 | .438 | 1.575 | 436 | 1.488 | 637 | 1.400 | 827 |
| 2 | 2.000 | .500 | 1.800 | 570 | 1.700 | 833 | 1.600 | 1080 |
| 2 1/4 | 2.250 | .563 | 2.025 | 721 | 1.913 | 1052 | 1.800 | 1367 |
| 2 1/2 | 2.500 | .625 | 2.250 | 891 | 2.125 | 1300 | 2.000 | 1688 |
| 3 | 3.000 | .750 | 2.700 | 1283 | 2.550 | 1873 | 2.400 | 2430 |
| | **Ten-spline Fittings** | | | | | | | |
| 3/4 | 0.750 | 0.117 | 0.683 | 120 | 0.645 | 183 | 0.608 | 241 |
| 7/8 | 0.875 | .137 | .796 | 165 | .753 | 248 | .709 | 329 |
| 1 | 1.000 | .156 | .910 | 215 | .860 | 326 | .810 | 430 |
| 1 1/8 | 1.125 | .176 | 1.024 | 271 | .968 | 412 | .911 | 545 |
| 1 1/4 | 1.250 | .195 | 1.138 | 336 | 1.075 | 508 | 1.013 | 672 |
| 1 3/8 | 1.375 | .215 | 1.251 | 406 | 1.183 | 614 | 1.114 | 813 |
| 1 1/2 | 1.500 | .234 | 1.365 | 483 | 1.290 | 732 | 1.215 | 967 |
| 1 5/8 | 1.625 | .254 | 1.479 | 566 | 1.398 | 860 | 1.316 | 1135 |
| 1 3/4 | 1.750 | .273 | 1.593 | 658 | 1.505 | 997 | 1.418 | 1316 |
| 2 | 2.000 | .312 | 1.820 | 860 | 1.720 | 1302 | 1.620 | 1720 |
| 2 1/4 | 2.250 | .351 | 2.048 | 1088 | 1.935 | 1647 | 1.823 | 2176 |
| 2 1/2 | 2.500 | .390 | 2.275 | 1343 | 2.150 | 2034 | 2.025 | 2688 |
| 3 | 3.000 | .468 | 2.730 | 1934 | 2.580 | 2929 | 2.430 | 3869 |
| 3 1/2 | 3.500 | .546 | 3.185 | 2632 | 3.010 | 3987 | 2.835 | 5266 |
| 4 | 4.000 | .624 | 3.640 | 3438 | 3.440 | 5208 | 3.240 | 6878 |
| 4 1/2 | 4.500 | .702 | 4.095 | 4351 | 3.870 | 6591 | 3.645 | 8705 |
| 5 | 5.000 | .780 | 4.550 | 5371 | 4.300 | 8137 | 4.050 | 10,746 |
| 5 1/2 | 5.500 | .858 | 5.005 | 6500 | 4.730 | 9846 | 4.455 | 13,003 |
| 6 | 6.000 | .936 | 5.460 | 7735 | 5.160 | 11,718 | 4.860 | 15,475 |
| | **Sixteen-spline Fittings** | | | | | | | |
| 2 | 2.000 | 0.196 | 1.820 | 1375 | 1.720 | 2083 | 1.620 | 2751 |
| 2 1/2 | 2.500 | .245 | 2.275 | 2149 | 2.150 | 3255 | 2.025 | 4299 |
| 3 | 3.000 | .294 | 2.730 | 3094 | 2.580 | 4687 | 2.430 | 6190 |
| 3 1/2 | 3.500 | .343 | 3.185 | 4212 | 3.010 | 6378 | 2.835 | 8426 |
| 4 | 4.000 | .392 | 3.640 | 5501 | 3.440 | 8333 | 3.240 | 11,005 |
| 4 1/2 | 4.500 | .441 | 4.095 | 6962 | 3.870 | 10,546 | 3.645 | 13,928 |
| 5 | 5.000 | .490 | 4.550 | 8595 | 4.300 | 13,020 | 4.050 | 17,195 |
| 5 1/2 | 5.500 | .539 | 5.005 | 10,395 | 4.730 | 15,754 | 4.455 | 20,806 |
| 6 | 6.000 | .588 | 5.460 | 12,377 | 5.160 | 18,749 | 4.860 | 24,760 |

* Torque capacity = 1000 × $n$ × mean radius × $h$ × $L$ where $n$ = number of splines, $L$ = length of spline bearing surface, in. Values of $h$ are determined by means of Table 12.

† Tolerance allowed of −0.001 in. for shafts 3/4 to 1 3/4 in. inclusive; of −0.002 for shafts 2 to 3 in., inclusive; −0.0003 in. for shafts 3 1/2 to 6-in., inclusive, for 4-, 6-, and 10-spline fittings; tolerance of −0.003 in. allowed for all sizes of 16-spline fittings.

‡ Tolerance allowed of −0.002 in. for shafts 3/4 to 1 3/4 in., inclusive; of −0.003 in. for shafts 2 to 6 in. inclusive, for 4-, 6-, and 10-spline fittings; tolerance of −0.003 allowed for all sizes of 16-spline fittings.

### Table 13.—Formulas for Spline Proportions. (See Fig. 16)

| No. of Splines | *W* For All Fits | Permanent Fit | | To Slide When Not Under Load | | To Slide Under Load | |
|---|---|---|---|---|---|---|---|
| | | *h* | *d* | *h* | *d* | *h* | *d* |
| Four.... | 0.241*D* | 0.0075*D* | 0.850*D* | 0.125*D* | 0.750*D* | ...... | ...... |
| Six...... | 0.250*D* | 0.050*D* | 0.900*D* | 0.075*D* | 0.850*D* | 0.100*D* | 0.800*D* |
| Ten..... | 0.156*D* | 0.045*D* | 0.910*D* | 0.070*D* | 0.860*D* | 0.095*D* | 0.810*D* |
| Sixteen.. | 0.098*D* | 0.045*D* | 0.910*D* | 0.070*D* | 0.860*D* | 0.095*D* | 0.810*D* |

**SQUARE SHAFTS.**—Table 14 and Fig. 17 give the proportions of square shafts and fittings as standardized by the S.A.E. For transmissions in which the hub or fittings must slide on the shaft, it is recommended that the ratio of the width across the flat to the width across the corners of the shaft, that is $A/B$, Fig. 17, be made equal to 0.73. If the hub or fitting is not to slide on the shaft, it is recommended that $A/B = 0.80$. It is recommended that holes be made to a standard size both for sliding and permanent fits, and that the size of the shaft be varied in accordance with the service required. The proportions given in Table 14 assume that the hub or fitting is not hardened and that the shafts may be either hardened or soft.

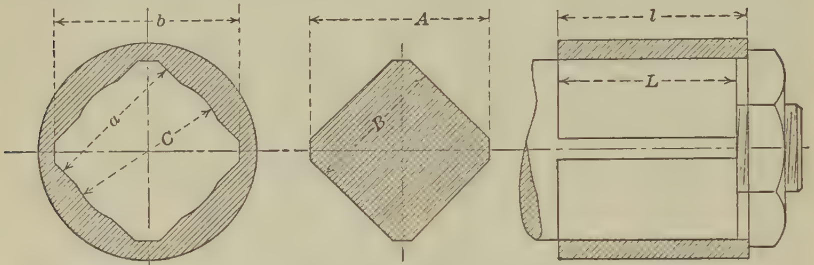


Fig. 17. Square Shaft and Fittings

**Table 14.—Dimensions of Square Shafts and Fittings**

S.A.E. Standard. All dimensions in inches. See Fig. 17

| Nominal Diameter | Sliding Fit | | | | | | Permanent Fit | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Shaft | | Fitting | | | | Shaft | | | Fitting | | | | |
| | Width Across Flats, *A*, Max.[1] | Width Across Corners, *B*, Max.[2] | Width Across Flats, *a*, Max.[3] | Width Across Corners, *b*, Max.[4] | Diameter of Recess *C* | Torque Capacity per 1-in. Length, in-lb. | Width Across Flats, *A*, Max.[1] | Width Across Corners, *B*, Max.[5] | Length of Square Section, *L* | Width Across Flats, *A*, Max.[3] | Width Across Corners, *b*, Max.[4] | Diameter of Recess *C* | Length of Fitting *l* | Torque Capacity per 1-in. Length, in.-lb. |
| 1/4 | 0.2480 | 0.3437 | 0.2500 | 0.3537 | 0.257 | 2.6 | 0.1895 | 0.250 | 5/16 | 0.1875 | 0.260 | 0.193 | 3/8 | ....... |
| 3/8 | .3730 | .5156 | .3750 | .5256 | 0.386 | 5.9 | .2832 | .375 | 7/16 | .2812 | .385 | 0.290 | 1/2 | ....... |
| 1/2 | .4980 | .6875 | .5000 | .6975 | 33/64 | 10.0 | .3770 | .500 | 11/16 | .3750 | .510 | 0.386 | 3/4 | 49 |
| 5/8 | .6230 | .8437 | .6250 | .8537 | 41/64 | 16.0 | .5020 | .625 | 11/16 | .5000 | .635 | 33/64 | 3/4 | 67 |
| 3/4 | .7480 | 1.0310 | .7500 | 1.0510 | 49/64 | 23.0 | .5645 | .750 | 15/16 | .5625 | .760 | 37/64 | 1 | 105 |
| 7/8 | .8730 | 1.1870 | .8750 | 1.2070 | 29/32 | 32.0 | .6895 | .875 | 1 1/8 | .6875 | .885 | 45/64 | 1 1/4 | 130 |
| 1 | .9980 | 1.3750 | 1.0000 | 1.3950 | 1 1/32 | 42.0 | .8155 | 1.000 | 1 3/8 | .8125 | 1.020 | 27/32 | 1 1/2 | 175 |
| 1 1/8 | 1.1230 | 1.5620 | 1.1250 | 1.5820 | 1 5/32 | 53.0 | .8780 | 1.125 | 1 3/8 | .8750 | 1.145 | 29/32 | 1 1/2 | 250 |
| 1 1/4 | 1.2480 | 1.6870 | 1.2500 | 1.7070 | 1 9/32 | 66.0 | 1.0030 | 1.250 | 1 3/8 | 1.0000 | 1.270 | 1 1/32 | 1 1/2 | 260 |
| 1 3/8 | 1.3730 | 1.8750 | 1.3750 | 1.8950 | 1 27/64 | 79 0 | 1.1280 | 1.375 | 1 7/8 | 1.1250 | 1.395 | 1 5/32 | 2 | 300 |
| 1 1/2 | 1.4980 | 2.0620 | 1.5000 | 2.0820 | 1 35/64 | 94.0 | 1.1280 | 1.500 | 1 7/8 | 1.1250 | 1.520 | 1 5/32 | 2 | 450 |
| 1 3/4 | 1.7480 | 2.3750 | 1.7500 | 2.3950 | 1 13/16 | 128.0 | 1.3780 | 1.750 | 2 1/8 | 1.3750 | 1.770 | 1 27/64 | 2 1/4 | 540 |
| 2 | 1.9975 | 2.7500 | 2.0000 | 2.7700 | 2 1/16 | 168.0 | 1.5040 | 2.000 | 2 7/8 | 1.5000 | 2.020 | 1 35/64 | 3 | 800 |
| 2 1/4 | 2.2475 | 3.0620 | 2.2500 | 3.0820 | 2 5/16 | 210.0 | 1.7540 | 2.250 | 2 7/8 | 1.7500 | 2.270 | 1 13/16 | 3 | 940 |
| 2 1/2 | 2.4975 | 3 4370 | 2.5000 | 3.4570 | 2 37/64 | 263.0 | 2.0040 | 2.500 | 3 3/8 | 2.0000 | 2.520 | 2 1/16 | 3 1/2 | 1050 |
| 2 3/4 | 2.7475 | 3.7500 | 2.7500 | 3.7700 | 2 55/64 | 315 0 | 2.2540 | 2.750 | 3 3/8 | 2.2500 | 2.770 | 2 5/16 | 3 1/2 | 1250 |
| 3 | 2.9970 | 4.1250 | 3.0000 | 4.1450 | 3 3/32 | 379 0 | 2.5040 | 3.000 | 3 7/8 | 2.5000 | 3.020 | 2 37/64 | 4 | 1300 |
| 3 1/2 | 3.4970 | 4.7500 | 3.5000 | 4.7700 | 3 39/64 | 514.0 | 2.7540 | 3.500 | 4 3/8 | 2.7500 | 3.520 | 2 55/64 | 4 1/2 | 2200 |
| 4 | 3 9970 | 5 5000 | 4.0000 | 5.5200 | 4 1/8 | 672 0 | 3.2540 | 4.000 | 5 3/8 | 3.2500 | 4 020 | 3 23/64 | 5 1/2 | 2800 |

[1] Tolerance allowed of −0.0010 in.

[2] Tolerance allowed of −0.0050 in.

[3] Tolerance allowed of −0.0010 in., for shafts 3/4 to 1 3/4 in. inclusive; −0.0015 in. for shafts 2 to 2 3/4 in., inclusive; −0.0020 in. for shafts 3 to 4 in., inclusive.

[4] Tolerance allowed of −0.0080 in. for shafts 1/4 to 5/8 in. inclusive; −0.0150 in. for shafts 3/4 to 4 in., inclusive.

[5] Tolerance allowed of −0.005 in.

[6] Tolerance allowed of −0.008 in. for shafts 1/4 to 7/8 in. inclusive; of −0.015 in. for shafts 1 in. to 4 in. inclusive.

## COTTERED JOINTS

A cotter is a form of key used to join parts that are subjected to a tension or compression, tending to separate them. The taper of cotters should not exceed $1/2$ in. per ft. of length unless a locking device is used. Various forms of cotters are given in Figs. 1 to 7,

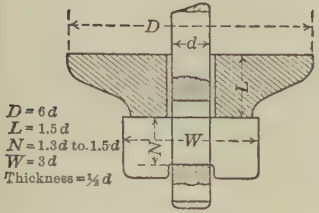


FIG. 1

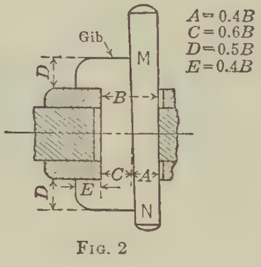


FIG. 2

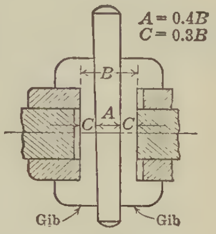


FIG. 3

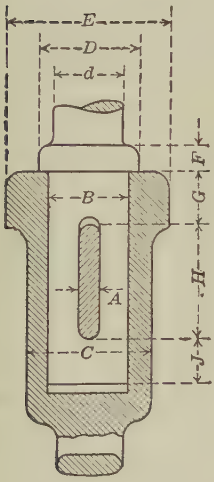


FIG. 4

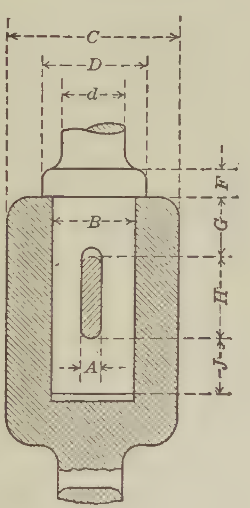


FIG. 5

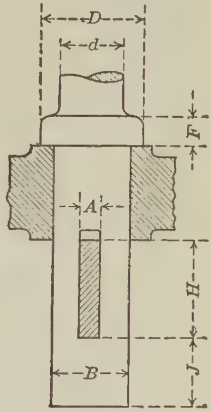


FIG. 6

and in Table 1 which shows the proportions of the joint. The function of the gib, in the gib and cotter joint, Figs. 2 and 3, is to prevent spreading of the strap. When a gib is used, the sides of the hole are made parallel, the taper being given to the gib. Where two cotters are used, as in Fig. 3, the taper is divided equally between them. The tapered joint, Fig. 7, usually is of permanent character. The cotter is removed only when the members which it holds are to be repaired or replaced. It is largely used for joining pistons and piston-rods.

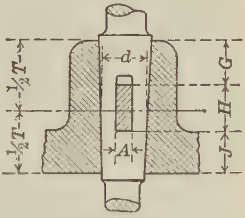


FIG. 7

**Table 1.—Dimensions of Cottered Joints.** (See Figs. 4–7)

| | Rod | Cotter | Dimensions | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *J* |
| Fig. 4 | W. I. | Steel | 0.3*d* | 1.21*d* | 1.75*d* | 1.5*d* | 2.4*d* | 0.42*d* | 0.8*d* | 1.6*d* | 0.8*d* |
| Fig. 5 | W. I. | Steel | 0.3*d* | 1.21*d* | 2.42*d* | 1.5*d* | ...... | 0.42*d* | 0.8*d* | 1.25*d* | 0.8*d* |
| Fig. 6 | Steel<br>W. I. | Steel<br>W. I. | 0.3*d* | 1.21*d* | ...... | 1.5*d* | ...... | 0.5*d* | ...... | 1.6*d* | 1.0*d* |
| Fig. 7 | Steel | Steel | 0.25*d* | ...... | ...... | ...... | ...... | ...... | 0.9*d* | 0.9*d* | 0.9*d* |

One form of grip taken from Machine Design Construction by Spooner is shown in Fig. 8. The joint may fail in any one of ten ways as follows:

**Notation.**—$f_t$ = ultimate strength of rod ends in tension; $f_s$ = ultimate strength of cotter in shear; $f_b$ = ultimate strength of bearing surfaces = say, $2f_t$; $f'_s$ = ultimate strength of wrought iron to resist shear in direction parallel with fibers; $F$ = axial force causing rupture of the joint; all in lb. per sq. in. The significance of the other quantities in the equations is shown in Fig. 8, all dimensions being taken in inches.

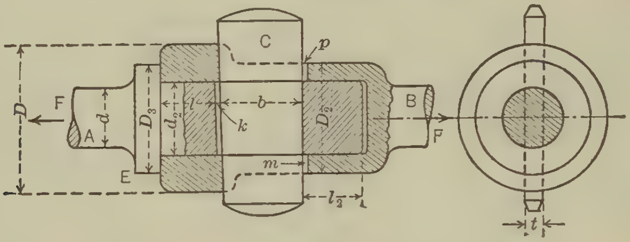


FIG. 8

1. The rods may fail in tension at A or B, in which case the external load

$$F = 1/4\pi d^2 f_t.$$

2. By the rod end tearing across the cotter hole; $F = (1/4\pi d_2^2 - d_2 t)f_t$. Equate cases [1] and [2] and let $t = 1/4\, d_2$; then

$$d_2 = 1.21\, d,\ d = 0.82\, d_2 \text{ and } t = 0.3\, d.$$

For joints with all parts either wrought iron or steel, the breadth of the cotter usually is about 1.6 $d$.

3. By the socket tearing across the cotter hole; then

$$F = \{1/4\pi(D_2^2 - d_2^2) - D_2 - d_2)t\}f_t,$$

whence by equating case [1], $D_2 = 1.63\, d$, or allowing for clearance, say, $D_2 = 1.75\, d$.

4. By double shearing of the cotter;

$$F = 2btf_s, \text{ and } b = 1.31\, df_t/f_s,$$

or 1.06 $d_2 f_t/f_s$, where $b$ = mean breadth of the cotter.

5. By crushing the cotter bearing surface in the rod end; $F = d_2 t f_b$.

6. By crushing the cotter bearing surface in socket; $F = t(D - d_2)f_b$ and $D = 2.42\, d$.

7. By shearing socket end; $F = 2\{l(D - d_2)f'_s\}$.

8. By shearing rod end; $F = 2\, l_2 d_2 f'_s$. $l = l_2 = 0.648\, d$; make $l$ from 0.75 $d$ to 1.0 $d$.

9. By crushing the collar E; $F = 1/4\pi(D_3^2 - d_2^2)f_b$ and $D_3 = 1.3915\, d$, say 1.4 $d$.

10. By shearing the collar E off the rod end; $F = d_2 \pi t_2 f'_s$ and $t_2 = 0.413\, d$, say $t_2 = 0.42\, d$.

## PINS

**STRENGTH OF KNUCKLE JOINT PIN.** (Spooner, Machine Design.)—The proportions of a forked pin joint, or knuckle joint are shown in Fig. 1. If there is no bending action, the pin will fail in double shear. If $d$ = diameter of rod, $d_1$ = diameter of the pin, $f_t$ = tensile strength of rod, lb. per sq. in., $f_s$ = shearing strength of pin, lb. per sq. in., = say, 0.8 $f_t$, then

$$1/4\pi d^2 f_t = 2(1/4\pi d_1^2 f_s), \text{ or } d_1 = 0.79\, d.$$

Unwin shows that where bending occurs and the joint is subjected to stresses reversing in sign, a skin stress of 5400 lb. per sq. in., due to bending will not be exceeded if the diameter of the pin = $0.0224\sqrt{P}$, where $P$ = axial force in the rod.

**BRIDGE PINS.**—Pins of various forms are used to connect rods or bars in groups, as in the lower chords of trusses. The pin must resist shearing, bending and bearing stresses, the latter two usually governing. Bar sizes should be so selected that the bearing area on the pin will be sufficient to avoid setting up excessive loads tending to crush either the pin or the bar. Also, bars must be so spaced that excessive bending stress is not set up in the pin. Let $M$ = moment of forces at any section of the pin; $f$ = unit stress in bending, lb. per sq. in.; $A$ = area of pin cross-section, sq. in.; $d$ = diameter of pin, in. Then

$$M = \pi f d^3/32 = f A d/8.$$

Fig. 2 shows plain pins and pins with heads, the pin being fastened with cotters. Dimensions are given in Table 1. Fig. 3 shows pins with recessed nuts. See Table 2 for dimensions. Bending moments and bearing stresses on pins are given in Table 3.

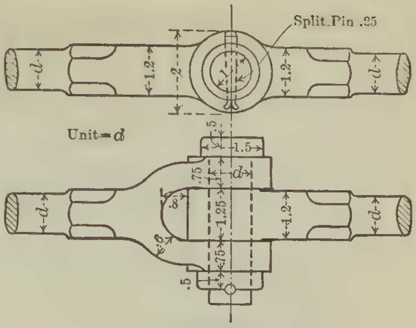


**Fig. 1.** Knuckle Joint

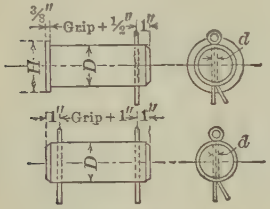


**Fig. 2**

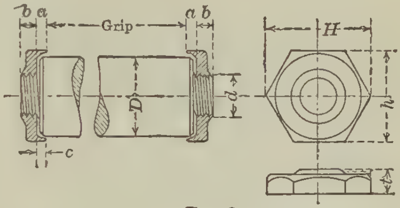


**Fig. 3**

**Table 1.—Dimensions of Cotter Pins**

American Bridge Co. Standard

All dimensions in inches. See Fig. 2

| Pin | | Cotter | | Pin | | Cotter | | Pin | | Cotter | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Diameter, *D* | Head Diameter, *H* | Diameter, *d* | Length | Diameter, *D* | Head Diameter, *H* | Diameter, *d* | Length | Diameter, *D* | Head Diameter, *H* | Diameter, *d* | Length |
| 1 1/4 | 1 1/2 | 1/4 | 2 | 2 1/4 | 2 5/8 | 3/8 | 3 1/4 | 3 1/4 | 3 3/4 | 1/2 | 5 |
| 1 1/2 | 1 3/4 | 1/4 | 2 1/2 | 2 1/2 | 2 7/8 | 3/8 | 3 3/4 | 3 1/2 | 4 | 1/2 | 6 |
| 1 3/4 | 2 | 1/4 | 2 3/4 | 2 3/4 | 3 1/8 | 3/8 | 4 | 3 3/4 | 4 1/4 | 1/2 | 6 |
| 2 | 2 3/8 | 3/8 | 3 | 3 | 3 1/2 | 1/2 | 5 | | | | |

**Table 2.—Pins and Recessed Pin Nuts**

American Bridge Co. Standard

All dimensions in inches. See Fig. 3

| Diameter of Pin, *D* | Pin | | | Nut | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Recess | | | |
| | *d* | *b* | *a* | Thickness, *t* | Short Diameter, *h* | Long Diameter, *H* | Rough Diameter | *c* | Diameter of Rough Hole | Weight, lb. |
| 2, 2 1/4 | 1 1/2 | 1 | 1/8 | 7/8 | 3 | 3 3/8 | 2 5/8 | 1/4 | 1 1/4 | 1 |
| 2 1/2, 2 3/4 | 2 | 1 1/8 | 1/8 | 1 | 3 5/8 | 4 1/8 | 3 1/8 | 1/4 | 1 3/4 | 2 |
| 3, 3 1/2 | 2 1/2 | 1 1/4 | 1/8 | 1 1/8 | 4 3/8 | 5 | 3 7/8 | 3/8 | 2 1/4 | 3 |
| 4 | 3 | 1 3/8 | 1/4 | 1 1/4 | 4 7/8 | 5 5/8 | 4 3/8 | 3/8 | 2 3/4 | 4 |
| 4 1/2 | 3 1/2 | 1 1/2 | 1/4 | 1 3/8 | 5 3/4 | 6 5/8 | 5 1/4 | 1/2 | 3 1/4 | 5 |
| 5 | 4 | 1 5/8 | 1/4 | 1 1/2 | 6 1/4 | 7 1/4 | 5 3/4 | 1/2 | 3 3/4 | 6 |
| 5 1/2, 6 | 4 1/2 | 1 3/4 | 1/4 | 1 5/8 | 7 | 8 1/8 | 6 1/2 | 5/8 | 4 1/4 | 8 |
| 7 | 5 1/2 | 2 | 3/8 | 1 7/8 | 8 1/8 | 9 3/8 | 7 1/2 | 3/4 | 5 1/4 | 12 |
| 8 | 6 | 2 1/4 | 3/8 | 2 1/8 | 9 3/8 | 10 7/8 | 8 3/4 | 3/4 | 5 3/4 | 19 |
| 9 | 6 | 2 1/4 | 3/8 | 2 1/8 | 10 1/4 | 11 7/8 | 9 5/8 | 3/4 | 5 3/4 | 24 |
| 10 | 6 | 2 3/8 | 3/8 | 2 1/4 | 11 1/4 | 13 | 10 5/8 | 3/4 | 5 3/4 | 32 |
| 11 | 6 | 2 1/2 | 1/2 | 2 7/8 | 12 1/2 | 14 3/8 | 11 3/4 | 3/4 | 5 3/4 | 60 |
| 12 | 6 | 2 5/8 | 1/2 | 2 7/8 | 13 1/2 | 15 5/8 | 12 3/4 | 3/4 | 5 3/4 | 73 |

### Table 3.—Bending Moments and Bearing Stresses on Pins

| Pin Diameter, in. | Pin Area, sq. in. | Unit Stress, lb. per sq. in. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 15,000 | 16,000 | 18,000 | 20,000 | 22,000 | 24,000 | 25,000 | 27,000 | 15,000 | 16,000 | 18,000 | 20,000 | 22,000 | 24,000 | 25,000 | 27,000 |
| | | Bending Moment, 1000 in.-lb. = Diam.$^3$ × 0.098175 × Unit Stress | | | | | | | | Bearing Value on Metal 1 in. Thick = Diam. × Unit Stress | | | | | | | |
| 1 | 0.785 | 1.5 | 1.6 | 1.8 | 2.0 | 2.2 | 2.4 | 2.5 | 2.7 | 15,000 | 16,000 | 18,000 | 20,000 | 22,000 | 24,000 | 25,000 | 27,000 |
| 1 1/4 | 1.227 | 2.9 | 3.1 | 3.5 | 3.8 | 4.2 | 4.6 | 4.8 | 5.2 | 18,750 | 20,000 | 22,500 | 25,000 | 27,500 | 30,000 | 31,250 | 33,750 |
| 1 1/2 | 1.767 | 5.0 | 5.3 | 6.0 | 6.6 | 7.3 | 8.0 | 8.3 | 8.9 | 22,500 | 24,000 | 27,000 | 30,000 | 33,000 | 36,000 | 37,500 | 40,500 |
| 1 3/4 | 2.405 | 7.9 | 8.4 | 9.5 | 10.5 | 11.6 | 12.6 | 13.2 | 14.2 | 26,250 | 28,000 | 31,500 | 35,000 | 38,500 | 42,000 | 43,750 | 47,250 |
| 2 | 3.142 | 11.8 | 12.6 | 14.1 | 15.7 | 17.3 | 18.8 | 19.6 | 21.2 | 30,000 | 32,000 | 36,000 | 40,000 | 44,000 | 48,000 | 50,000 | 54,000 |
| 2 1/4 | 3.976 | 16.8 | 17.9 | 20.1 | 22.4 | 24.6 | 26.8 | 28.0 | 30.2 | 33,750 | 36,000 | 40,500 | 45,000 | 49,500 | 54,000 | 56,250 | 60,750 |
| 2 1/2 | 4.909 | 23.0 | 24.5 | 27.6 | 30.7 | 33.7 | 36.8 | 38.3 | 41.4 | 37,500 | 40,000 | 45,000 | 50,000 | 55,000 | 60,000 | 62,500 | 67,500 |
| 2 3/4 | 5.940 | 30.6 | 32.7 | 36.8 | 40.8 | 44.9 | 49.0 | 51.0 | 55.1 | 41,250 | 44,000 | 49,500 | 55,000 | 60,500 | 66,000 | 68,750 | 74,250 |
| 3 | 7.069 | 39.8 | 42.4 | 47.7 | 53.0 | 58.3 | 63.6 | 66.3 | 71.6 | 45,000 | 48,000 | 54,000 | 60,000 | 66,000 | 72,000 | 75,000 | 81,000 |
| 3 1/4 | 8.296 | 50.6 | 53.9 | 60.7 | 67.4 | 74.1 | 80.9 | 84.3 | 91.0 | 48,750 | 52,000 | 58,500 | 65,000 | 71,500 | 78,000 | 81,250 | 87,750 |
| 3 1/2 | 9.621 | 63.1 | 67.3 | 75.8 | 84.2 | 92.6 | 101.0 | 105.2 | 113.7 | 52,500 | 56,000 | 63,000 | 70,000 | 77,000 | 84,000 | 87,500 | 94,500 |
| 3 3/4 | 11.045 | 77.7 | 82.8 | 93.2 | 103.5 | 113.9 | 124.3 | 129.4 | 139.8 | 56,250 | 60,000 | 67,500 | 75,000 | 82,500 | 90,000 | 93,750 | 101,250 |
| 4 | 12.566 | 94.2 | 100.5 | 113.1 | 125.7 | 138.2 | 150.8 | 157.1 | 169.6 | 60,000 | 64,000 | 72,000 | 80,000 | 88,000 | 96,000 | 100,000 | 108,000 |
| 4 1/4 | 14.186 | 113.0 | 120.6 | 135.7 | 150.7 | 165.8 | 180.9 | 188.4 | 203.5 | 63,750 | 68,000 | 76,500 | 85,000 | 93,500 | 102,000 | 106,250 | 114,750 |
| 4 1/2 | 15.904 | 134.2 | 143.1 | 161.0 | 178.9 | 196.8 | 214.7 | 223.7 | 241.6 | 67,500 | 72,000 | 81,000 | 90,000 | 99,000 | 108,000 | 112,500 | 121,500 |
| 4 3/4 | 17.721 | 157.8 | 168.3 | 189.4 | 210.4 | 231.5 | 252.5 | 263.0 | 284.1 | 71,250 | 76,000 | 85,500 | 95,000 | 104,500 | 114,000 | 118,750 | 128,250 |
| 5 | 19.635 | 184.1 | 196.4 | 220.9 | 245.4 | 270.0 | 294.5 | 306.8 | 331.3 | 75,000 | 80,000 | 90,000 | 100,000 | 110,000 | 120,000 | 125,000 | 135,000 |
| 5 1/4 | 21.648 | 213.1 | 227.3 | 255.7 | 284.1 | 312.5 | 340.9 | 355.2 | 383.6 | 78,750 | 84,000 | 94,500 | 105,000 | 115,500 | 126,000 | 131,250 | 141,750 |
| 5 1/2 | 23.758 | 245.0 | 261.3 | 294.0 | 326.7 | 359.3 | 392.0 | 408.3 | 441.0 | 82,500 | 88,000 | 99,000 | 110,000 | 121,000 | 132,000 | 137,500 | 148,500 |
| 5 3/4 | 25.967 | 280.0 | 298.6 | 336.0 | 373.3 | 410.6 | 447.9 | 466.6 | 503.9 | 86,250 | 92,000 | 103,500 | 115,000 | 126,500 | 138,000 | 143,750 | 155,250 |
| 6 | 28.274 | 318.1 | 339.3 | 381.7 | 424.1 | 466.5 | 508.9 | 530.1 | 572.6 | 90,000 | 96,000 | 108,000 | 120,000 | 132,000 | 144,000 | 150,000 | 162,000 |
| 6 1/4 | 30.680 | 359.5 | 383.5 | 431.4 | 479.4 | 527.3 | 575.2 | 599.2 | 647.1 | 93,750 | 100,000 | 112,500 | 125,000 | 137,500 | 150,000 | 156,250 | 168,750 |
| 6 1/2 | 33.183 | 404.4 | 431.4 | 485.3 | 539.2 | 593.1 | 647.1 | 674.0 | 728.0 | 97,500 | 104,000 | 117,000 | 130,000 | 143,000 | 156,000 | 162,500 | 175,500 |
| 6 3/4 | 35.785 | 452.9 | 483.1 | 543.5 | 603.9 | 664.3 | 724.6 | 754.8 | 815.2 | 101,250 | 108,000 | 121,500 | 135,000 | 148,500 | 162,000 | 168,750 | 182,250 |
| 7 | 38.485 | 505.1 | 538.8 | 606.1 | 673.5 | 740.8 | 808.2 | 841.8 | 909.2 | 105,000 | 112,000 | 126,000 | 140,000 | 154,000 | 168,000 | 175,000 | 189,000 |
| 7 1/2 | 44.179 | 621.3 | 662.7 | 745.5 | 828.4 | 911.2 | 994.0 | 1035.4 | 1118.3 | 112,500 | 120,000 | 135,000 | 150,000 | 165,000 | 180,000 | 187,500 | 202,500 |
| 8 | 50.265 | 754.0 | 804.3 | 904.8 | 1005.3 | 1105.8 | 1206.4 | 1256.6 | 1357.2 | 120,000 | 128,000 | 144,000 | 160,000 | 176,000 | 192,000 | 200,000 | 216,000 |
| 8 1/2 | 56.745 | 904.4 | 964.7 | 1085.3 | 1205.8 | 1326.4 | 1447.0 | 1507.3 | 1627.9 | 127,500 | 136,000 | 153,000 | 170,000 | 187,000 | 204,000 | 212,500 | 229,500 |
| 9 | 63.617 | 1073.5 | 1145.1 | 1288.3 | 1431.4 | 1574.5 | 1717.7 | 1789.2 | 1932.4 | 135,000 | 144,000 | 162,000 | 180,000 | 198,000 | 216,000 | 225,000 | 243,000 |
| 9 1/2 | 70.882 | 1262.6 | 1346.8 | 1515.1 | 1683.5 | 1851.8 | 2020.1 | 2104.3 | 2272.7 | 142,500 | 152,000 | 171,000 | 190,000 | 209,000 | 228,000 | 237,500 | 256,500 |
| 10 | 78.540 | 1472.6 | 1570.8 | 1767.1 | 1963.5 | 2159.8 | 2356.2 | 2454.4 | 2650.7 | 150,000 | 160,000 | 180,000 | 200,000 | 220,000 | 240,000 | 250,000 | 270,000 |
| 10 1/2 | 86.590 | 1704.7 | 1818.4 | 2045.7 | 2273.0 | 2500.3 | 2727.6 | 2841.2 | 3068.5 | 157,500 | 168,000 | 189,000 | 210,000 | 231,000 | 252,000 | 262,500 | 283,500 |
| 11 | 95.033 | 1960.1 | 2090.7 | 2352.1 | 2613.4 | 2874.8 | 3136.1 | 3266.8 | 3528.1 | 165,000 | 176,000 | 198,000 | 220,000 | 242,000 | 264,000 | 275,000 | 297,000 |
| 11 1/2 | 103.869 | 2239.7 | 2389.0 | 2687.6 | 2986.2 | 3284.9 | 3583.5 | 3732.8 | 4031.4 | 172,500 | 184,000 | 207,000 | 230,000 | 253,000 | 276,000 | 287,500 | 310,500 |
| 12 | 113.097 | 2544.7 | 2714.3 | 3053.6 | 3392.9 | 3732.2 | 4071.5 | 4241.2 | 4580.5 | 180,000 | 192,000 | 216,000 | 240,000 | 264,000 | 288,000 | 300,000 | 324,000 |

**Table 4.—Dimensions of Split Cotter Pins**

S.A.E. Standard

| Nominal Diameter | Actual Diameter | Limits | Length Min. | Length Max. | Length Advancing by Increments of | Nominal Diameter | Actual Diameter | Limits | Length Min. | Length Max. | Length Advancing by Increments of |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/16 | 0.061 | 0.058 | 1/2 | 7/8 | 1/8 in.* | 3/16 | 0.176 | 0.172 | 1 5/8 | 2 1/2 | 1/4 in.‡ |
| 3/32 | .090 | .086 | 1/2 | 1 1/8 | 1/8 in. | 7/32 | .207 | .202 | 2 | 3 | 1/4 in. |
| 1/8 | .122 | .118 | 7/8 | 1 3/4 | 1/8 in. | 1/4 | .225 | .220 | 2 1/4 | 3 | 1/4 in. |
| 5/32 | .150 | .146 | 1 1/4 | 2 1/4 | 1/8 in.† | 5/16 | .280 | .275 | 2 1/2 | 3 | 1/4 in. |

* Additional lengths, 5/16 and 7/16 in. † Increments above 1 3/4 in. are 1/4 in.
‡ First increment is 1/8 in. from 1 5/8 to 1 3/4 in.

# TAPERS

Up to 1937 there was no single standard taper in the United States for spindles and drill shanks. The more important tapers in common use were: the Sellers taper used for lathe, drilling and boring machine spindles; the Morse taper, used for drilling machine spindles and drill shanks; the Brown & Sharpe taper, used for milling machine spindles; the Reed taper, used for lathe spindles; the Jarno taper used for lathe and drilling machine spindles; the Cleveland Twist Drill Co. taper and the Standard Tool Co. taper, used for drill shanks. Dimensions of these various tapers are given in Tables 4 to 10.

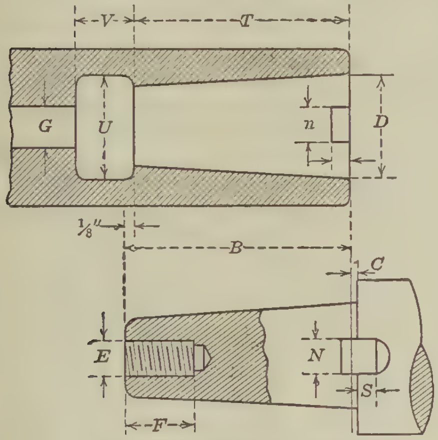


FIG. 1. Taper with Draw-bolt. See Table 1

**AMERICAN STANDARD TAPER.**—In 1937, the American Standards Association published standards for self-holding tapers which comprise 19 sizes. The three smaller sizes were taken from the Brown & Sharpe series, having a nominal taper of 1/2 in. per foot; the next six sizes were taken from the Morse series having a nominal taper of 5/8 in. per

**Table 1.—American Standard Key Drive Self-holding Taper. Shank Retained by Draw Bolt***

A.S.A. B5. 10–1937. All dimensions in inches. See Fig. 1

| | | | Shank | | | | | | Socket | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Drive Keyway | | | | | | Drive Keyway | |
| No. of Taper | Taper per Foot | Diameter at Gage Line, *D* | Overall Length, *B* | Exposed Length, *C* | Tap Size, † *E* | Length of Thread, *F* | Width, *N* | Depth, *S* | Length of Taper, *T* | Relief Diameter, *U* | Relief Depth, *V* | Draw-bolt Hole Diameter, *G* | Width, *n* | Depth, *S* |
| 200 | 0.750 | 2.000 | 5 | 1/8 | 7/8 | 1 3/4 | 1.010 | 1/2 | 4 3/4 | 1 13/16 | 1 | 1 | 1.000 | 1/2 |
| 250 | .750 | 2.500 | 5 3/4 | 1/8 | 7/8 | 1 3/4 | 1.010 | 1/2 | 5 1/2 | 2 1/4 | 1 | 1 | 1 000 | 1/2 |
| 300 | .750 | 3.000 | 6 1/2 | 1/8 | 1 | 2 | 1.010 | 1/2 | 6 1/4 | 2 3/4 | 1 | 1 1/8 | 1 000 | 1/2 |
| 350 | .750 | 3.500 | 7 1/4 | 1/8 | 1 | 2 | 1.010 | 1/2 | 7 | 3 3/16 | 1 | 1 1/8 | 1.000 | 1/2 |
| 400 | .750 | 4.000 | 8 | 1/8 | 1 1/4 | 2 1/2 | 2.010 | 3/4 | 7 3/4 | 3 5/8 | 1 | 1 3/8 | 2 000 | 3/4 |
| 500 | .750 | 5.000 | 9 9/16 | 3/16 | 1 1/4 | 2 1/2 | 2.010 | 3/4 | 9 1/4 | 4 5/8 | 1 | 1 3/8 | 2.000 | 3/4 |
| 600 | .750 | 6.000 | 11 1/16 | 3/16 | 1 1/2 | 3 | 2 010 | 3/4 | 10 3/4 | 5 1/2 | 1 | 1 5/8 | 2.000 | 3/4 |
| 800 | .750 | 8.000 | 14 1/16 | 3/16 | 1 1/2 | 3 | 4.010 | 1 | 13 3/4 | 7 3/8 | 1 | 1 5/8 | 4.000 | 1 |
| 1000 | .750 | 10.000 | 17 1/16 | 3/16 | 2 | 4 | 4.010 | 1 | 16 3/4 | 9 3/16 | 1 | 2 1/4 | 4.000 | 1 |
| 1200 | .750 | 12.000 | 20 1/16 | 3/16 | 2 | 4 | 4.010 | 1 | 19 3/4 | 10 | 1 | 2 1/4 | 4.000 | 1 |

* See p. 12–19 for tolerances. † Am. National Standard Coarse Thread.

## Table 2.—American Standard Tongue Drive, Self-holding Tapers *

A.S.A. B5. 10–1937. All dimensions in inches. See Figs. 2 and 3

NOTE. Standard friction-retained shanks Nos. .239 to 350 incl.; key-retained shanks Nos. 3 to 350 inclusive.

| | | | Friction-retained Shanks | | | | | | | | | | | | | Holdback Keyway. See Fig. 3 | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Shank | | Tongue | | | | | Socket | Keyway | | | | | Shank | | | Socket † | | |
| No. of Taper | Taper per Foot | Diameter at Gage Line, *D* | Overall Length, *B* | Exposed Length, *C* | Thickness, *E* | Length, *F* | Radius of Mill, *G* | Diameter, *H* | Corner Radius, *J* | Depth, *K* | Length of Taper, *M* | Width, *W* | Length, *P* | Tongue to Bottom, *R* | Small End to Top, *T* | Length, *L* | Width, *W* | End to Keyway, *S* | Length, *U* | Width, *W* |
| .239 | 0.500 | .239 | 1 9/32 | 3/32 | 1/8 | 3/16 | 3/16 | 11/64 | 1/32 | 1 1/16 | 15/16 | 0.141 | 3/8 | 1/8 | ...... | ...... | ...... | ...... | ...... | ...... |
| .299 | .500 | .299 | 1 19/32 | 3/32 | 5/32 | 1/4 | 3/16 | 7/32 | 1/32 | 1 5/16 | 1 11/64 | .172 | 1/2 | 11/64 | ...... | ...... | ...... | ...... | ...... | ...... |
| .375 | .500 | .375 | 1 31/32 | 3/32 | 3/16 | 5/16 | 3/16 | 9/32 | 3/64 | 1 5/8 | 1 15/32 | .203 | 5/8 | 7/32 | ...... | ...... | ...... | ...... | ...... | ...... |
| 1 | .600 | .475 | 2 5/8 | 3/16 | 13/64 | 3/8 | 3/16 | 11/32 | 3/64 | 2 3/16 | 2 1/16 | .219 | 3/4 | 3/8 | ...... | ...... | ...... | ...... | ...... | ...... |
| 2 | .600 | .700 | 3 1/8 | 3/16 | 1/4 | 7/16 | 1/4 | 17/32 | 1/16 | 2 5/8 | 2 1/2 | .266 | 7/8 | 7/16 | ...... | ...... | ...... | ...... | ...... | ...... |
| 3 | .602 | .938 | 3 7/8 | 3/16 | 5/16 | 9/16 | 9/32 | 23/32 | 5/64 | 3 1/4 | 3 1/16 | .328 | 1 1/16 | 9/16 | 2 21/32 | 1 1/8 | 0.328 | 1 1/8 | 1 3/16 | 0.328 |
| 4 | .623 | 1.231 | 4 7/8 | 1/4 | 15/32 | 5/8 | 5/16 | 31/32 | 3/32 | 4 1/8 | 3 7/8 | .484 | 1 1/4 | 1/2 | 3 7/32 | 1 3/16 | .484 | 1 1/2 | 1 1/4 | .484 |
| 4 1/2 | .623 | 1.500 | 5 3/8 | 1/4 | 9/16 | 11/16 | 3/8 | 1 13/64 | 7/64 | 4 9/16 | 4 5/16 | .594 | 1 3/8 | 9/16 | 3 13/32 | 1 1/4 | .594 | 1 13/16 | 1 3/8 | .594 |
| 5 | .630 | 1.748 | 6 1/8 | 1/4 | 5/8 | 3/4 | 3/8 | 1 13/32 | 1/8 | 5 1/4 | 4 15/16 | .656 | 1 1/2 | 9/16 | 3 7/8 | 1 3/8 | .656 | 2 1/8 | 1 1/2 | .656 |
| 200 | .750 | 2.000 | 5 3/4 | 1/4 | 5/8 | 13/16 | 9/16 | 1 37/64 | 1/8 | 4 7/8 | 4 1/2 | .656 | 1 9/16 | 9/16 | 3 1/2 | 1 9/16 | .656 | 2 1/8 | 1 11/16 | .656 |
| 250 | .750 | 2.500 | 6 3/4 | 1/4 | 3/4 | 1 1/16 | 11/16 | 2 1/32 | 5/32 | 5 5/8 | 5 3/16 | .781 | 1 15/16 | 5/8 | 3 3/4 | 1 9/16 | .781 | 2 7/8 | 1 11/16 | .781 |
| 300 | .750 | 3.000 | 7 5/8 | 1/4 | 1 | 1 1/4 | 3/4 | 2 31/64 | 5/32 | 6 3/8 | 5 15/16 | 1.031 | 2 3/16 | 3/4 | 3 15/16 | 1 9/16 | 1.031 | 3 9/16 | 1 11/16 | 1.031 |
| 350 | .750 | 3.500 | 8 3/8 | 1/4 | 1 | 1 1/4 | 7/8 | 2 15/16 | 3/16 | 7 1/8 | 6 5/8 | 1.031 | 2 1/4 | 3/4 | 4 1/2 | 2 | 1.031 | 3 3/4 | 2 1/8 | 1.031 |

* See p. 12–19 for tolerances.

† Corners at entrance side of keyways to be chamfered at 45 deg. as follows: No. 3, 3/64 in. deep; No. 4 to No. 350, inclusive, 1/16 in. deep.

## Table 3.—American Standard Key-drive Self-holding Taper. Shank Retained by Key *

A.S.A. Standard B5. 10–1937. All dimensions in inches. See Fig. 4

| | | | Shank | | | Holdback Keyway in Shank | | | Drive Keyway | | Socket | | | Knockout Keyway | | | | Holdback Keyway in Socket † | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| No. of Taper | Taper per Foot | Diameter at Gage Line, *D* | Overall Length, *B* | Exposed Length, *C* | Bevel of Shank End, *E* | Small End to Top, *R* | Length, *F* | Width, *W* | Width, *N* | Depth, *S* | End to Relief, *T* | Relief Diameter, *G* | Relief Depth, *H* | End of Socket to Keyway, *M* | Width, *W* | Length, *J* | Shank End to Bottom, *P* | End of Socket to Keyway, *K* | Length, *L* | Width, *W* |
| 200 | 0.750 | 2.000 | 5 1/4 | 1/8 | 1/16 | 3 1/8 | 1 9/16 | 0.656 | 1.000 | 1/2 | 4 3/4 | 1 13/16 | 1 | 4 1/2 | 0.656 | 1 9/16 | 15/16 | 2 1/8 | 1 11/16 | 0.656 |
| 250 | .750 | 2.500 | 6 | 1/8 | 1/16 | 3 1/8 | 1 9/16 | 0.781 | 1.000 | 1/2 | 5 1/2 | 2 1/4 | 1 | 5 3/16 | 0.781 | 1 15/16 | 1 1/4 | 2 7/8 | 1 11/16 | 0.781 |
| 300 | .750 | 3.000 | 6 3/4 | 1/8 | 1/16 | 3 3/16 | 1 9/16 | 1.031 | 1.000 | 1/2 | 6 1/4 | 2 3/4 | 1 | 5 15/16 | 1.031 | 2 3/16 | 1 1/2 | 3 9/16 | 1 11/16 | 1.031 |
| 350 | .750 | 3.500 | 7 9/16 | 1/8 | 1/8 | 3 13/16 | 2 | 1.031 | 1.000 | 1/2 | 7 | 3 3/16 | 1 | 6 5/8 | 1.031 | 2 1/4 | 1 7/16 | 3 3/4 | 2 1/8 | 1.031 |
| 400 | .750 | 4.000 | 8 5/16 | 1/8 | 1/8 | 4 1/2 | 2 1/4 | 1.031 | 2.000 | 3/4 | 7 3/4 | 3 5/8 | 1 | 7 1/4 | 1.031 | 2 3/8 | 1 7/16 | 3 7/8 | 2 3/8 | 1.031 |
| 500 | .750 | 5.000 | 9 15/16 | 3/16 | 3/16 | 4 15/16 | 2 5/8 | 1.031 | 2.000 | 3/4 | 9 1/4 | 4 5/8 | 1 | 8 3/4 | 1.031 | 2 3/4 | 1 3/4 | 5 | 2 3/4 | 1.031 |
| 600 | .750 | 6.000 | 11 1/2 | 3/16 | 3/16 | 5 7/8 | 3 | 1.281 | 2.000 | 3/4 | 10 3/4 | 5 1/2 | 1 | 10 1/8 | 1.281 | 3 1/4 | 2 1/16 | 5 5/8 | 3 1/4 | 1.281 |
| 800 | .750 | 8.000 | 14 9/16 | 3/16 | 1/4 | 7 11/16 | 4 | 1.781 | 4.000 | 1 | 13 3/4 | 7 3/8 | 1 | 12 7/8 | 1.781 | 4 1/4 | 2 3/4 | 6 7/8 | 4 1/4 | 1.781 |
| 1000 | .750 | 10.000 | 17 5/8 | 3/16 | 5/16 | 8 15/16 | 4 3/4 | 2.031 | 4.000 | 1 | 16 3/4 | 9 3/16 | 1 | 15 3/4 | 2.031 | 5 | 3 5/16 | 8 3/4 | 5 | 2.031 |
| 1200 | .750 | 12.000 | 20 11/16 | 3/16 | 3/8 | 10 3/4 | 5 3/4 | 2.531 | 4.000 | 1 | 19 3/4 | 11 | 1 | 18 1/2 | 2.531 | 6 | 4 | 10 | 6 | 2.531 |

* See p. 12–19 for tolerances.

† Corners at entrance side of keyways to be chamfered at 45 deg. as follows: No. 200 to No. 300, inclusive, 3/16 in. deep; No. 400 to No. 600, inclusive, 3/32 in. deep. No. 800 to No. 1200, inclusive, 1/8 in. deep.

foot; the ten largest sizes were based on a taper of $^3/_4$ in. per foot. The 13 smaller sizes are driven by a tongue or tang, except that the four largest sizes of this group may be driven by a key. The 13 smallest sizes are retained entirely by friction, or, in the case of the eight largest of this group, they may be retained by a holdback key. The ten largest sizes, except as noted above, are driven by a key, and may be retained in place either by a holdback key or a draw-bolt. Dimensions are given in Tables 1 to 3 and Figs. 1 to 4.

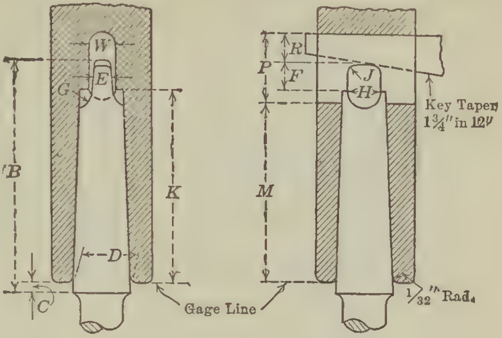


FIG. 2. Taper with Tongue Drive, Shank Retained by Friction. See Table 2

**TOLERANCES FOR AMERICAN STANDARD TAPERS.**—Diameter of Shank and Hole at Gage Line, dimension *D*: +0.002, −0.000.

**THICKNESS OF TONGUE,** dimension *E*, Table 2: From $^1/_8$ in. to $^1/_4$ in. inclusive, +0.000, −0.004 in.; from $^5/_{16}$ in. to $^{15}/_{32}$ in. inclusive, +0.000, −0.006 in.; from $^9/_{16}$ in. to 1 in. inclusive, +0.000, −0.008 in.

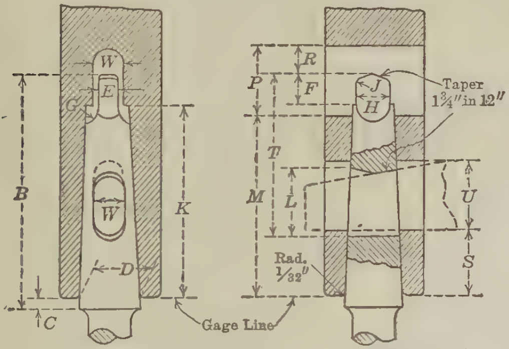


FIG. 3. Taper with Tongue Drive, Shank Retained by Key. See Table 2

**Concentricity of Tongue** with center line of taper, dimension *E*, Table 2: Tapers No. .239 to No. 4 inclusive, 0.0035 in.; tapers No. 4 $^1/_2$ to No. 350 inclusive, 0.007 in.

**Width of Keyway, Friction-retained and Key-retained Tapers,** dimension *W*, Table 2: Tapers No. .239 to No. 2 inclusive, +0.004, −0.000 in.; No. 3 and No. 4, +0.006, −0.000 in.; No. 4 $^1/_2$ to No. 350 inclusive, +0.008, −0.000; key drive tapers (Table 3) all sizes, +0.008, −0.000.

**Concentricity of Keyway,** dimension *W*, with center line of taper: Friction-retained and key-retained tapers (Table 2), tapers No. .239 to No. 4 inclusive, 0.0035 in.; tapers No. 4 $^1/_2$ to No. 350 inclusive, 0.007 in.; key-driven tapers (Table 3), all sizes, 0.007 in.

**Drive Keyway,** width, dimension *N*, Table 3: In spindle, +0.000, −0.001 in.; in shank, +0.010, −0.013 in.; width, dimension *N*, Table 1, +0.000, −0.001 in.; width, dimension *n*, Table 1, +0.003, −0.000 in.

**Tolerance on Fractional Dimensions,** for all types of tapers is +0.010, −0.010 in., unless otherwise specified.

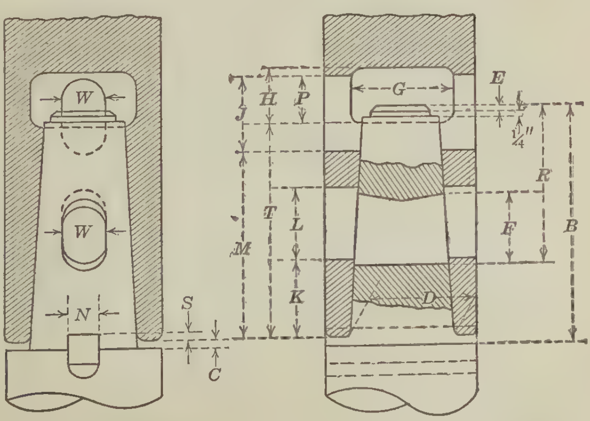


FIG. 4. Taper with Key Drive and Holdback Key. See Table 3

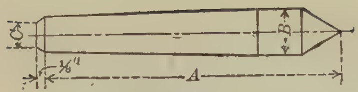


Fig. 5. Reed Lathe Center Taper. See Table 4

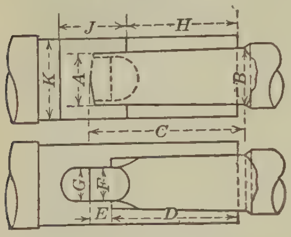


Fig. 6. Standard Tool Co. Taper See Table 5

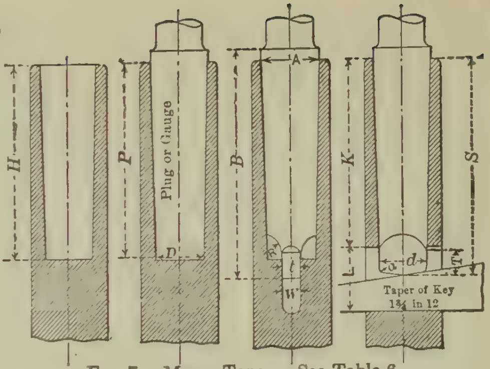


Fig. 7. Morse Taper. See Table 6

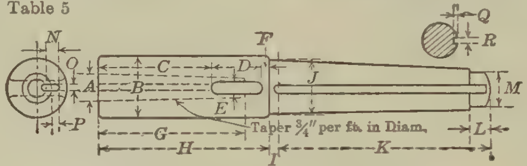


Fig. 8. Sellers Taper. See Table 8

### Table 4.—The Reed Lathe Center Taper

All dimensions in inches. See Fig. 5

| Size of lathe, in. | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 27 | 30 |
|---|---|---|---|---|---|---|---|---|---|---|
| Length, *A*, in. | 3 1/16 | 5 1/8 | 6 1/8 | 6 3/8 | 7 5/8 | 8 1/16 | 8 1/16 | 8 1/2 | 8 1/2 | 9 |
| Large diam., *B*, in. | 21/32 | 25/32 | 1 5/32 | 1 15/32 | 1 1/2 | 1 25/32 | 1 25/32 | 2 1/2 | 2 1/2 | 2 5/16 |
| Small diam., *C*, in. | 1/2 | 9/16 | 15/16 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 | 1 3/4 | 1 3/4 | 2 |

### Table 5.—Standard Tool Co. Taper for Twist Drill Shanks

All dimensions in inches. See Fig. 6

| No. of Taper | Diam. Shank, Small End | Diam. Shank Large End | Total Length of Shank | Depth of Socket Hole | Length of Tongue to End of Socket Hole | Thickness of Tongue | Width of Keyway | End of Socket to Keyway | Length of Keyway | Diam. of Socket | Taper per Foot | Taper per Inch |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *J* | *K* | | |
| 1 | 0.378 | 0.484 | 2 1/8 | 1 3/4 | 1/4 | 1/4 | 0.263 | 1 5/8 | 3/4 | 25/32 | 0.600 | 0.0500 |
| 2 | .587 | .706 | 2 3/8 | 1 15/16 | 5/16 | 3/8 | .388 | 1 3/4 | 1 | 1 1/16 | .602 | .05016 |
| 3 | .800 | .941 | 2 13/16 | 2 1/4 | 3/8 | 1/2 | .520 | 2 | 1 1/4 | 1 5/16 | .602 | .05016 |
| 4 | 1.050 | 1.244 | 3 3/4 | 3 | 9/16 | 5/8 | .645 | 2 11/16 | 1 1/2 | 1 11/16 | .623 | .05191 |
| 5 | 1.515 | 1.757 | 4 5/8 | 3 3/4 | 5/8 | 1 | 1.020 | 3 1/4 | 2 | 2 7/16 | .630 | .0525 |
| 6 | 2.169 | 2.501 | 6 3/8 | 5 | 3/4 | 1 1/4 | 1.270 | 4 3/8 | 2 1/2 | 2 7/8 | .626 | .05216 |
| 7 | 2.815 | 3.283 | 9 | 7 3/4 | 1 | 1 1/2 | 1.520 | 7 | 3 | ...... | .625 | .05208 |

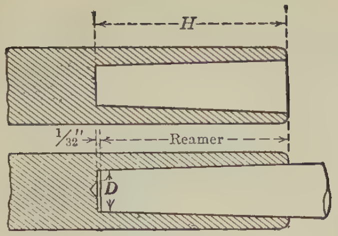
### Table 6.—Morse Tapers

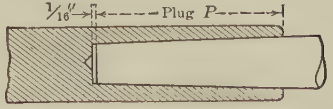
All dimensions in inches. See Fig. 7

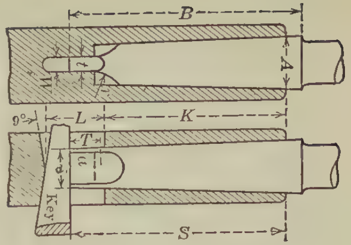
| No. of Taper | Diam. of Plug at Small End | Diam. at End of Socket | Standard Plug Depth | Whole Length of Shank | Depth of Hole | End of Socket to Keyway | Length of Keyway | Width of Keyway | Length of Tongue | Diam. of Tongue | Thickness of Tongue | Rad. of Mill for Tongue | Radius of Tongue | Shank Depth | Taper per Foot |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *D* | *A* | *P* | *B* | *H* | *K* | *L* | *W* | *T* | *d* | *t* | *R* | *a* | *S* | |
| 0 | 0.252 | 0.356 | 2 | 2 11/32 | 2 1/32 | 1 15/16 | 9/16 | 0.160 | 1/4 | 0.235 | 5/32 | 5/32 | 0.04 | 2 7/32 | .625 |
| 1 | .369 | .475 | 2 1/8 | 2 9/16 | 2 3/16 | 2 1/16 | 3/4 | .213 | 3/8 | .343 | 13/64 | 3/16 | .05 | 2 7/16 | .600 |
| 2 | .572 | .700 | 2 9/16 | 3 1/8 | 2 5/8 | 2 1/2 | 7/8 | .26 | 7/16 | 17/32 | 1/4 | 1/4 | .06 | 2 15/16 | .602 |
| 3 | .778 | .938 | 3 3/16 | 3 7/8 | 3 1/4 | 3 1/16 | 13/16 | .322 | 9/16 | 23/32 | 5/16 | 9/32 | .08 | 3 11/16 | .602 |
| 4 | 1.020 | 1.231 | 4 1/16 | 4 7/8 | 4 1/8 | 3 7/8 | 1 1/4 | .478 | 5/8 | 31/32 | 15/32 | 5/16 | .10 | 4 5/8 | .623 |
| 5 | 1.475 | 1.748 | 5 3/16 | 6 1/8 | 5 1/4 | 4 15/16 | 1 1/2 | .635 | 3/4 | 1 13/32 | 5/8 | 3/8 | .12 | 5 7/8 | .630 |
| 6 | 2.116 | 2.494 | 7 1/4 | 8 9/16 | 7 3/8 | 7 | 1 3/4 | .76 | 1 1/8 | 2 | 3/4 | 1/2 | .15 | 8 1/4 | .626 |
| 7 | 2.75 | 3.27 | 10 | 11 5/8 | 10 1/8 | 9 1/2 | 2 5/8 | 1.135 | 1 3/8 | 2 5/8 | 1 1/8 | 3/4 | .18 | 11 1/4 | .625 |

## Table 7.—Cleveland Twist Drill Tapers

All dimensions in inches.







| No. of Taper | Small Diam. of Plug | Standard Plug Depth | Depth of Hole | End of Socket to Keyway | Length of Keyway | Width of Key-way | Length of Tongue | Small Diam. of Shank | Thickness of Tongue | Radius of Mill for Tongue | Radius of Tongue | Shank Depth | Total Length of Shank | Taper per Foot | Diam. at End of Socket | Taper per Inch |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | D | P | H | K | L | W | T | d | t | r | a | S | B | | A | |
| 0 | 0.252 | 2 | 2 1/32 | 1 13/16 | 9/16 | 0.160 | 1/4 | 0.235 | 5/32 | 5/32 | 0.04 | 2 7/32 | 2 13/32 | 0.625 | 0.356 | 0.05208 |
| 1 | .369 | 2 1/8 | 2 3/16 | 2 1/16 | 3/4 | .213 | 3/8 | .353 | 13/64 | 3/8 | .05 | 2 7/16 | 2 11/16 | .600 | .475 | .05 |
| 2 | .572 | 2 9/16 | 2 5/8 | 2.488 | 7/8 | .265 | 7/16 | .553 | 1/4 | 3/8 | 1/16 | 2 15/16 | 3 3/16 | .602 | .700 | .05016 |
| 3 | .778 | 3 3/16 | 3 1/4 | 3 1/16 | 1 1/16 | .330 | 9/16 | .753 | 5/16 | 7/16 | 3/32 | 3 11/16 | 3 15/16 | .602 | .938 | .05016 |
| 4 | 1.02 | 4 1/16 | 4 1/8 | 3 7/8 | 1 1/4 | .490 | 5/8 | .991 | 15/32 | 7/16 | 3/32 | 4 5/8 | 5 1/8 | .623 | 1.231 | .05191 |
| 5 | 1.475 | 5 3/16 | 5 1/4 | 4 15/16 | 1 1/2 | .650 | 3/4 | 1.440 | 5/8 | 1/2 | 1/8 | 5 7/8 | 6 3/8 | .630 | 1.748 | .0525 |
| 6 | 2.116 | 7 1/4 | 7 3/8 | 7 | 1 3/4 | .780 | 1 | 2.064 | 3/4 | 11/16 | 1/8 | 8 1/4 | 8 3/4 | .626 | 2.494 | .05216 |

## Table 8.—The Sellers Taper

All dimensions in inches. (See Fig. 8)

| Diam. Large End of Socket Hole | Out-side Diam. | End of Socket to Keyhole | Length of Key-hole | Width of Key-hole | End of Keyhole to Bot-tom of Shank | Depth of Socket Hole | Length of Spindle | Bottom of Shank to End of Key-way | Large Diam. of Shank | Bottom of Shank to End of Tongue | Length of Tongue | Width of Tongue | Depth of Socket Key | Width of Socket Key | Depth of Socket Keyway | Depth of Shank Keyway | Width of Shank Keyway |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| A | B | C | D | E | F | G | H | I | J | K | L | M | N | O | P | Q | R |
| 1/2 | 1 | 1 15/16 | 3/4 | 1/4 | 3/16 | 2 7/16 | 2 7/8 | 1/8 | 11/16 | 3 | 1/4 | 7/16 | 9/64 | 5/64 | 5/64 | 3/32 | 1/8 |
| 1/2 | 1 | 1 15/16 | 3/4 | 1/4 | 3/16 | 2 7/16 | 2 7/8 | 1/8 | 7/8 | 3 1/2 | 5/16 | 9/16 | 9/64 | 5/64 | 5/64 | 3/32 | 1/8 |
| 11/16 | 1 5/16 | 2 3/8 | 7/8 | 5/16 | 1/4 | 2 15/16 | 3 1/2 | 1/8 | 1 1/8 | 4 1/2 | 3/8 | 3/4 | 11/64 | 7/64 | 3/32 | 1/8 | 3/16 |
| 7/8 | 1 5/8 | 3 1/16 | 1 | 3/8 | 5/16 | 3 11/16 | 4 3/8 | 1/8 | 1 3/8 | 5 1/2 | 3/8 | 15/16 | 11/64 | 7/64 | 3/32 | 5/32 | 1/4 |
| 1 1/8 | 2 | 4 1/8 | 1 | 3/8 | 3/8 | 4 3/4 | 5 1/2 | 1/4 | 1 5/8 | 6 1/2 | 7/16 | 1 1/8 | 15/64 | 11/64 | 1/8 | 5/32 | 1/4 |
| 1 5/8 | 2 7/8 | 5 7/8 | 1 1/2 | 5/8 | 1/2 | 6 3/4 | 7 7/8 | 1/4 | 2 | 8 | 1/2 | 1 3/8 | 21/64 | 15/64 | 3/16 | 3/16 | 7/16 |
| 2 3/8 | 3 1/2 | 7 1/4 | 1 3/4 | 5/8 | ........ | 9 3/8 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 27/64 | 1/4 | ........ | ........ |

### Table 9.—The Brown & Sharpe Taper

All dimensions in inches. See Fig. 9

| No. of Taper | Diam. of Plug at Small End *D* | Plug Depth *P* | Depth of Hole *H* | Keyway from End of Spindle *K* | Length of Key-way *L* | Width of Key-way *W* | Length of Arbor Tongue *T* | Thick-ness of Arbor Tongue *t* | Taper per Foot |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.20 | 15/16 | 1 1/16 | 15/16 | 3/8 | 0.135 | 3/16 | 1/8 | 0.500 |
| 2 | .25 | 1 3/16 | 1 5/16 | 1 11/64 | 1/2 | .166 | 1/4 | 5/32 | .500 |
| 3 | .312 | 2 | 2 1/8 | 1 31/32 | 5/8 | .197 | 5/16 | 3/16 | .500 |
| 4 | .35 | 1 1/4 | 1 3/8 | 1 13/64 | 11/16 | .228 | 11/32 | 7/32 | .500 |
| 5 | .45 | 1 3/4 | 1 7/8 | 1 11/16 | 3/4 | .260 | 3/8 | 1/4 | .500 |
| 6 | .50 | 2 3/8 | 2 1/2 | 2 19/64 | 7/8 | .291 | 7/16 | 9/32 | .500 |
| 7 | .60 | 3 | 3 1/8 | 2 29/32 | 15/16 | .322 | 15/32 | 5/16 | .500 |
| 8 | .75 | 3 9/16 | 3 11/16 | 3 29/64 | 1 | .353 | 1/2 | 11/32 | .500 |
| 9 | .90 | 4 | 4 1/8 | 3 7/8 | 1 1/8 | .385 | 9/16 | 3/8 | .500 |
| 9 | .90 | 4 1/4 | 4 3/8 | 4 1/8 | 1 1/8 | .385 | 9/16 | 3/8 | .500 |
| 10 | 1.0446 | 5 | 5 1/8 | 4 27/32 | 1 5/16 | .447 | 21/32 | 7/16 | .5161 |
| 10 | 1.0446 | 5 11/16 | 5 13/16 | 5 17/32 | 1 5/16 | .447 | 21/32 | 7/16 | .5161 |
| 11 | 1.25 | 6 3/4 | 6 7/8 | 6 19/32 | 1 5/16 | .447 | 21/32 | 7/16 | .500 |
| 12 | 1.50 | 7 1/8 | 7 1/4 | 6 15/16 | 1 1/2 | .510 | 3/4 | 1/2 | .500 |
| 13 | 1.75 | 7 3/4 | 7 7/8 | 7 9/16 | 1 1/2 | .510 | 3/4 | 1/2 | .500 |
| 14 | 2 | 8 1/4 | 8 3/8 | 8 1/32 | 1 11/16 | .572 | 27/32 | 9/16 | .500 |
| 15 | 2.25 | 8 3/4 | 8 7/8 | 8 17/32 | 1 11/16 | .572 | 27/32 | 9/16 | .500 |
| 16 | 2.50 | 9 1/4 | 9 3/8 | 9 | 1 7/8 | .635 | 15/16 | 5/8 | .500 |
| 17 | 2.75 | 9 3/4 | 9 7/8 | ......... | ......... | ......... | ......... | ......... | .500 |
| 18 | 3 | 10 1/4 | 10 3/8 | ......... | ......... | ......... | ......... | ......... | .500 |

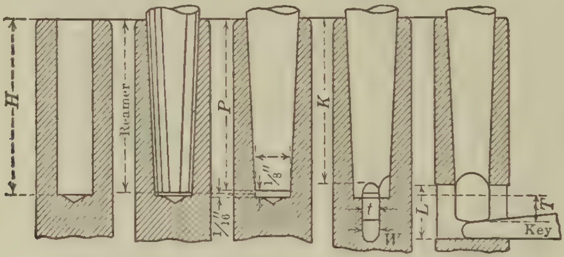


FIG. 9. Brown & Sharpe Taper. See Table 9

### Table 10.—Jarno Taper Shanks.

Taper 0.6 in. per ft. All dimensions in inches See Fig. 10.

| Taper No. | *A* | *B* | *C* | *D* | Taper No. | *A* | *B* | *C* | *D* |
|---|---|---|---|---|---|---|---|---|---|
| 2 | 1 1/8 | 1 | 0.20 | 0.250 | 12 | 6 1/4 | 6 | 1.20 | 1.500 |
| 3 | 1 5/8 | 1 1/2 | .30 | .375 | 13 | 6 3/4 | 6 1/2 | 1.30 | 1.625 |
| 4 | 2 3/16 | 2 | .40 | .500 | 14 | 7 1/4 | 7 | 1.40 | 1.750 |
| 5 | 2 11/16 | 2 1/2 | .50 | .625 | 15 | 7 3/4 | 7 1/2 | 1.50 | 1.875 |
| 6 | 3 3/16 | 3 | .60 | .750 | 16 | 8 5/16 | 8 | 1.60 | 2.000 |
| 7 | 3 11/16 | 3 1/2 | .70 | .875 | 17 | 8 13/16 | 8 1/2 | 1.70 | 2.125 |
| 8 | 4 3/16 | 4 | .80 | 1.000 | 18 | 9 5/16 | 9 | 1.80 | 2.250 |
| 9 | 4 11/16 | 4 1/2 | .90 | 1.125 | 19 | 9 13/16 | 9 1/2 | 1.90 | 2.375 |
| 10 | 5 1/4 | 5 | 1.00 | 1.250 | 20 | 10 5/16 | 10 | 2.00 | 2.500 |
| 11 | 5 3/4 | 5 1/2 | 1.10 | 1.375 | | | | | |

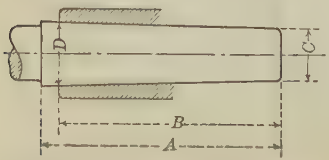


FIG. 10. Jarno Taper. See Table 10

**THE JARNO TAPER** is 0.05 in. per in. = 0.6 in. per ft. The number of the taper is its diameter in tenths of an inch at the small end, in eighths of an inch at the large end, and the length in halves of an inch. If $N$ = number of taper, $D$, $d$ = diameter at large and small ends, respectively, $L$=length, $D=N/8$, $d=N/10$, $L = N/2$. Thus, No. 3 Jarno taper is 1 1/2 in. long, 0.3 in. diameter at the small end and 3/8 in. diameter at the large end. See Table 10 and Fig. 10.

**Table 11.—Tapers per Foot and Corresponding Angles**

| Taper per Foot, in. | Included Angle | | Angle with Center Line | | Taper per Foot, in. | Included Angle | | Angle with Center Line | |
|---|---|---|---|---|---|---|---|---|---|
| | Deg. | Min. | Deg. | Min. | | Deg. | Min. | Deg. | Min. |
| 1/8 | 0 | 36 | 0 | 18 | 1 | 4 | 46 | 2 | 23 |
| 1/4 | 1 | 12 | 0 | 36 | 1 1/2 | 7 | 09 | 3 | 34 1/2 |
| 5/16 | 1 | 30 | 0 | 45 | 1 3/4 | 8 | 20 | 4 | 10 |
| 3/8 | 1 | 47 | 0 | 53 1/2 | 2 | 9 | 31 | 4 | 45 1/2 |
| 7/16 | 2 | 05 | 1 | 02 1/2 | 2 1/2 | 11 | 54 | 5 | 57 |
| 1/2 | 2 | 23 | 1 | 11 1/2 | 3 | 14 | 15 | 7 | 07 1/2 |
| 3/4 | 3 | 35 | 1 | 47 1/2 | 3 1/2 | 16 | 36 | 8 | 18 |
| 15/16 | 4 | 28 | 2 | 14 | 4 | 18 | 55 | 9 | 27 1/2 |

# FITS

The degree of tightness or looseness between two parts that are intended to act together is known as the fit of the parts. The character of the fit depends upon the use to which it is to be put. Thus the fit between members which move or rotate relative to each other, as a journal rotating in a bearing, is considerably different from the fit which is designed to prevent any relative motion of the fitted members as, for example, a tire fitted to a locomotive driving wheel center.

Manufacturers and engineering associations have adopted in the past many different standards for fits to meet their particular needs. For details of these practices, see earlier editions of this book and also *Machinery's Handbook*, 9th Edition. In 1925, the American Standards Association published standards for fits, tolerances and allowances for all classes of service. These standards are abstracted below. For fits in journal bearings, see p. 13–31.

**TERMS RELATING TO FITS.**—**Basic Size** is the exact theoretical size from which all limiting variations are measured.

**Allowance** is an intentional difference in the dimensions of mating parts, or it may represent the minimum clearance space that is intended between mating parts. It represents the tightest permissible fit, *i.e.*, the largest internal member mated with the smallest external member.

**Tolerance** is the amount of variation permitted in the size of a part. In manufacturing a large number of pieces, it is not possible to produce them to identical dimensions, and the tolerance allows variations from the basic size that will permit them to be produced, but which will not prevent their proper functioning. Variations must tend toward greater looseness than the ideal conditions of the smallest hole, or external member, and the largest shaft, or internal member. Tolerances should be stated on the drawing by giving the basic size and the maximum and minimum permissible variations therefrom. Thus,

$$\text{Shaft, 0.874 in.}\left\{\begin{matrix}+0.000\text{ in.}\\-0.001\text{ in.}\end{matrix}\right.;\quad \text{Hole, 0.875 in.}\left\{\begin{matrix}+0.001\text{ in.}\\-0.000\text{ in.}\end{matrix}\right.$$

This indicates a manufacturing tolerance of 0.001 in. on each part. The clearance with the greatest possible looseness is 0.003 in., and with the greatest possible tightness, 0.001 in.

**Limits** are the extreme permissible dimensions of a part.

**CLASSIFICATION OF FITS.**—Fits are classified as follows:

**Class 1, Loose Fit, Large Allowance.**—This fit provides for considerable freedom and embraces certain fits where accuracy is not essential. Examples of its use are the machined fits on agricultural and mining machinery, control apparatus for marine work, machinery in the textile, rubber and canning industries, and general machinery of similar grades. It is used for strictly interchangeable assembly. Tolerances and allowances are determined by the following formulas, in which $d$ = diameter of fit, in inches:

$$\text{Allowance} = 0.0025\sqrt[3]{d^2};\quad \text{hole tolerance and shaft tolerance} = 0.0025\sqrt[3]{d}.$$

Dimensions are given in Table 1.

**Class 2, Free Fit, Liberal Allowance.**—This fit is designed as a running fit for speeds of 600 r.p.m. or over, and journal pressures of 600 lb. per sq. in. or more. Examples of its use are electric generators, engines, machine tool parts, and automotive parts. It is used for strictly interchangeable assembly. Allowances and tolerances are determined by the following formulas:

$$\text{Allowance} = 0.0014\sqrt[3]{d^2};\quad \text{hole and shaft tolerance} = 0.0013\sqrt[3]{d}.$$

Dimensions are given in Table 1.

### Table 1.—American Standard Loose and Free Fits

A.S.A. B4a–1925. All dimensions in inches

| Size | | | Class 1. Loose Fit, Large Allowance, Interchangeable | | | | | Class 2. Free Fit, Liberal Allowance, Interchangeable | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Limits | | | | | Limits | | | | |
| From | Up to and Including | Mean | Hole or External Member * | Shaft or Internal Member | | Tightest Fit, Allowance | Loosest Fit, Allowance +Tolerances | Hole or External Member * | Shaft or Internal Member | | Tightest Fit, Allowance | Loosest Fit, Allowance +Tolerances |
| | | | Plus | Minus | Minus | Plus † | Plus † | Plus | Minus | Minus | Plus † | Plus † |
| 0 | 3/16 | 1/8 | 0.001 | 0.001 | 0.002 | 0.001 | 0.003 | 0.0007 | 0.0004 | 0.0011 | 0.0004 | 0.0018 |
| 3/16 | 5/16 | 1/4 | .002 | .001 | .003 | .001 | .005 | .0008 | .0006 | .0014 | .0006 | .0022 |
| 5/16 | 7/16 | 3/8 | .002 | .001 | .003 | .001 | .005 | .0009 | .0007 | .0016 | .0007 | .0025 |
| 7/16 | 9/16 | 1/2 | .002 | .002 | .004 | .002 | .006 | .0010 | .0009 | .0019 | .0009 | .0029 |
| 9/16 | 11/16 | 5/8 | .002 | .002 | .004 | .002 | .006 | .0011 | .0010 | .0021 | .0010 | .0032 |
| 11/16 | 13/16 | 3/4 | .002 | .002 | .004 | .002 | .006 | .0012 | .0012 | .0024 | .0012 | .0036 |
| 13/16 | 15/16 | 7/8 | .002 | .002 | .004 | .002 | .006 | .0012 | .0013 | .0025 | .0013 | .0037 |
| 15/16 | 1 1/16 | 1 | .003 | .003 | .006 | .003 | .009 | .0013 | .0014 | .0027 | .0014 | .0040 |
| 1 1/16 | 1 3/16 | 1 1/8 | .003 | .003 | .006 | .003 | .009 | .0014 | .0015 | .0029 | .0015 | .0043 |
| 1 3/16 | 1 3/8 | 1 1/4 | .003 | .003 | .006 | .003 | .009 | .0014 | .0016 | .0030 | .0016 | .0044 |
| 1 3/8 | 1 5/8 | 1 1/2 | .003 | .003 | .006 | .003 | .009 | .0015 | .0018 | .0033 | .0018 | .0048 |
| 1 5/8 | 1 7/8 | 1 3/4 | .003 | .004 | .007 | .004 | .010 | .0016 | .0020 | .0036 | .0020 | .0052 |
| 1 7/8 | 2 1/8 | 2 | .003 | .004 | .007 | .004 | .010 | .0016 | .0022 | .0038 | .0022 | .0054 |
| 2 1/8 | 2 3/8 | 2 1/4 | .003 | .004 | .007 | .004 | .010 | .0017 | .0024 | .0041 | .0024 | .0058 |
| 2 3/8 | 2 3/4 | 2 1/2 | .003 | .005 | .008 | .005 | .011 | .0018 | .0026 | .0044 | .0026 | .0062 |
| 2 3/4 | 3 1/4 | 3 | .004 | .005 | .009 | .005 | .013 | 0019 | .0029 | .0048 | .0029 | .0067 |
| 3 1/4 | 3 3/4 | 3 1/2 | .004 | .006 | .010 | .006 | .014 | .0020 | .0032 | .0052 | .0032 | .0072 |
| 3 3/4 | 4 1/4 | 4 | .004 | .006 | .010 | .006 | .014 | .0021 | .0035 | .0056 | .0035 | .0077 |
| 4 1/4 | 4 3/4 | 4 1/2 | .004 | .007 | .011 | .007 | .015 | .0021 | .0038 | .0059 | .0038 | .0080 |
| 4 3/4 | 5 1/2 | 5 | .004 | .007 | .011 | .007 | .015 | .0022 | .0041 | .0063 | .0041 | .0085 |
| 5 1/2 | 6 1/2 | 6 | .005 | .008 | .013 | .008 | .018 | 0024 | .0046 | .0070 | .0046 | .0094 |
| 6 1/2 | 7 1/2 | 7 | .005 | .009 | .014 | .009 | .019 | .0025 | .0051 | .0076 | .0051 | .0101 |
| 7 1/2 | 8 1/2 | 8 | .005 | .010 | .015 | .010 | .020 | .0026 | .0056 | .0082 | .0056 | .0108 |

### Table 2.—American Standard Medium and Snug Fits

A.S.A. B4a–1925. All dimensions in inches

| Size | | | Class 3. Medium Fit, Medium Allowance, Interchangeable | | | | | Class 4. Snug Fit, Zero Allowance, Interchangeable | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Limits | | | | | Limits | | |
| From | Up to and Including | Mean | Hole or External Member * | Shaft or Internal Member | | Tightest Fit, Allowance | Loosest Fit, Allowance + Tolerances | Hole or External Member * | Shaft or Internal Member ‡ | Loosest Fit, Allowance + Tolerances § |
| | | | Plus | Minus | Minus | Plus † | Plus † | Plus † | Minus | Plus † |
| 0 | 3/16 | 1/8 | 0.0004 | 0.0002 | 0.0006 | 0.0002 | 0.0010 | 0.0003 | 0.0002 | 0.0005 |
| 3/16 | 5/16 | 1/4 | .0005 | .0004 | .0009 | .0004 | .0014 | .0004 | .0003 | .0007 |
| 5/16 | 7/16 | 3/8 | .0006 | .0005 | .0011 | .0005 | .0017 | .0004 | .0003 | .0007 |
| 7/16 | 9/16 | 1/2 | .0006 | .0006 | .0012 | .0006 | .0018 | .0005 | .0003 | .0008 |
| 9/16 | 11/16 | 5/8 | .0007 | .0007 | .0014 | .0007 | .0021 | .0005 | .0003 | .0008 |
| 11/16 | 13/16 | 3/4 | .0007 | .0007 | .0014 | .0007 | .0021 | .0005 | .0004 | .0009 |
| 13/16 | 15/16 | 7/8 | .0008 | .0008 | .0016 | .0008 | .0024 | .0006 | .0004 | .0010 |
| 15/16 | 1 1/16 | 1 | .0008 | .0009 | .0017 | .0009 | .0025 | .0006 | .0004 | .0010 |
| 1 1/16 | 1 3/16 | 1 1/8 | .0008 | .0010 | .0018 | .0010 | .0026 | .0006 | .0004 | .0010 |
| 1 3/16 | 1 3/8 | 1 1/4 | .0009 | .0010 | .0019 | .0010 | .0028 | .0006 | .0004 | .0010 |
| 1 3/8 | 1 5/8 | 1 1/2 | .0009 | .0012 | .0021 | .0012 | .0030 | .0007 | .0005 | .0012 |
| 1 5/8 | 1 7/8 | 1 3/4 | .0010 | .0013 | .0023 | .0013 | .0033 | .0007 | .0005 | .0012 |
| 1 7/8 | 2 1/8 | 2 | .0010 | .0014 | .0024 | .0014 | .0034 | .0008 | .0005 | .0013 |
| 2 1/8 | 2 3/8 | 2 1/4 | .0010 | .0015 | .0025 | .0015 | .0035 | .0008 | .0005 | .0013 |
| 2 3/8 | 2 3/4 | 2 1/2 | .0011 | .0017 | .0028 | .0017 | .0039 | .0008 | .0005 | .0013 |
| 2 3/4 | 3 1/4 | 3 | .0012 | .0019 | .0031 | .0019 | .0043 | .0009 | .0006 | .0015 |
| 3 1/4 | 3 3/4 | 3 1/2 | .0012 | .0021 | .0033 | .0021 | .0045 | .0009 | .0006 | .0015 |
| 3 3/4 | 4 1/4 | 4 | .0013 | .0023 | .0036 | .0023 | .0049 | .0010 | .0006 | .0016 |
| 4 1/4 | 4 3/4 | 4 1/2 | .0013 | .0025 | .0038 | .0025 | .0051 | .0010 | .0007 | .0017 |
| 4 3/4 | 5 1/2 | 5 | .0014 | .0026 | .0040 | .0026 | .0054 | .0010 | .0007 | .0017 |
| 5 1/2 | 6 1/2 | 6 | .0015 | .0030 | .0045 | .0030 | .0060 | .0011 | .0007 | .0018 |
| 6 1/2 | 7 1/2 | 7 | .0015 | .0033 | .0048 | .0033 | .0063 | .0011 | .0008 | .0019 |
| 7 1/2 | 8 1/2 | 8 | .0016 | .0036 | .0052 | .0036 | .0068 | .0012 | .0008 | .0020 |

* Minimum limit = 0.0000 in. for all sizes.

‡ Maximum limit = 0.0000 in. for all sizes.

† Plus here denotes clearance or amount of looseness.

§ Tightest fit allowance = 0.0000 for all sizes.

**Class 3, Medium Fit, Medium Allowance.**—This fit is designed for running fits under 600 r.p.m. and with journal pressures of less than 600 lb. per sq. in. It also is used for sliding fits, and the more accurate machine tool and automotive parts. It is designed for strictly interchangeable assembly. Allowances and hole tolerances are determined by the following formulas:

$$\text{Allowance} = 0.0009\sqrt[3]{d^2};\ \text{hole tolerance and shaft tolerance} = 0.0008\sqrt[3]{d}.$$

Dimensions are given in Table 2.

**Class 4, Snug Fit, Zero Allowance.**—A snug fit is the closest fit that can be assembled by hand. It necessitates work of considerable precision. It is used where no perceptible shake is permitted, and where moving parts are not intended to move freely under load. An example of its use is the assembly of the body and jaws of a drill chuck. It is designed for strictly interchangeable assembly. Allowance = 0.0000. Hole and shaft tolerances are determined by the following formulas:

$$\text{Hole tolerance} = 0.0006\sqrt[3]{d};\ \text{shaft tolerance} = 0.0004\sqrt[3]{d}.$$

Dimensions are given in Table 2.

**Class 5, Wringing Fit, Zero to Negative Allowance.**—This fit also is known as a *tunking fit*. It is practically a metal-to-metal fit. Assembly usually is selective and not interchangeable. Examples of its use are locating keys, tongues or dowels. The selected average interference of metal is 0.0000 in. Hole and shaft tolerances are determined by the following formulas:

$$\text{Hole tolerance} = 0.0006\sqrt[3]{d};\ \text{shaft tolerance} = 0.0004\sqrt[3]{d}.$$

Dimensions are given in Table 3.

**Class 6, Tight Fit, Slight Negative Allowance.**—These fits are used where parts are to be more or less permanently assembled, and light pressure is required in the assembly operation. They are used for drive fits in thin sections, or very long fits in other sections, and also for shrink fits on very light sections. Examples of the use of this fit are jig bushings, and fixed ends of studs for gears, pulleys, rocker arms, in automotive, ordnance

**Table 3.—American Standard Wringing Fits and Tight Fits**

A.S.A. B4a–1925. All dimensions in inches

| Size | | | Class 5. Wringing Fit, Zero to Negative Allowance. Selective Assembly | | | | Class 6. Tight Fit, Slight Negative Allowance Selective Assembly | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Limits | | Tightest Fit, Allowance | Loosest Fit, Allowance + Tolerances | Limits | | | Tightest Fit, Allowance | Loosest Fit, Allowance + Tolerances | Selected Fit, Average Interference of Metal |
| From | Up to and Including | Mean | Hole or External Member * | Shaft or Internal Member * | | | Hole or External Member * | Shaft or Internal Member | | | | |
| | | | Plus | Plus | Minus † | Plus ‡ | Plus | Plus | Plus | Minus † | † ‡ | Minus † |
| 0 | 3/16 | 1/8 | 0.0003 | 0.0002 | 0.0002 | 0.0003 | 0 0003 | 0.0003 | 0.0000 | 0.0003 | +0.0003 | 0.0000 |
| 3/16 | 5/16 | 1/4 | .0004 | .0003 | .0003 | .0004 | .0004 | .0005 | .0001 | .0005 | + .0003 | .0001 |
| 5/16 | 7/16 | 3/8 | .0004 | .0003 | .0003 | .0004 | .0004 | .0005 | .0001 | .0005 | + .0003 | .0001 |
| 7/16 | 9/16 | 1/2 | .0005 | .0003 | .0003 | .0005 | .0005 | .0006 | .0001 | .0006 | + .0004 | .0001 |
| 9/16 | 11/16 | 5/8 | .0005 | .0003 | .0003 | .0005 | .0005 | .0007 | .0002 | .0007 | + .0003 | .0002 |
| 11/16 | 13/16 | 3/4 | .0005 | .0004 | .0004 | .0005 | .0005 | .0007 | .0002 | .0007 | + .0003 | .0002 |
| 13/16 | 15/16 | 7/8 | .0006 | .0004 | .0004 | .0006 | .0006 | .0008 | .0002 | .0008 | + .0004 | .0002 |
| 15/16 | 1 1/16 | 1 | .0006 | .0004 | .0004 | .0006 | .0006 | .0009 | .0003 | .0009 | + .0003 | .0003 |
| 1 1/16 | 1 3/16 | 1 1/8 | .0006 | .0004 | .0004 | .0006 | .0006 | .0009 | .0003 | .0009 | + .0003 | .0003 |
| 1 3/16 | 1 3/8 | 1 1/4 | .0006 | .0004 | .0004 | .0006 | .0006 | .0009 | .0003 | .0009 | + .0003 | .0003 |
| 1 3/8 | 1 5/8 | 1 1/2 | .0007 | .0005 | .0005 | .0007 | .0007 | .0011 | .0004 | .0011 | + .0003 | .0004 |
| 1 5/8 | 1 7/8 | 1 3/4 | .0007 | .0005 | .0005 | .0007 | .0007 | .0011 | .0004 | .0011 | + .0003 | .0004 |
| 1 7/8 | 2 1/8 | 2 | .0008 | .0005 | .0005 | .0008 | .0008 | .0013 | .0005 | .0013 | + .0003 | .0005 |
| 2 1/8 | 2 3/8 | 2 1/4 | .0008 | .0005 | .0005 | .0008 | .0008 | .0014 | .0006 | .0014 | + .0002 | .0006 |
| 2 3/8 | 2 3/4 | 2 1/2 | .0008 | .0005 | .0005 | .0008 | .0008 | .0014 | .0006 | .0014 | + .0002 | .0006 |
| 2 3/4 | 3 1/4 | 3 | .0009 | .0006 | .0006 | .0009 | .0009 | .0017 | .0008 | .0017 | + .0001 | .0008 |
| 3 1/4 | 3 3/4 | 3 1/2 | .0009 | .0006 | .0006 | .0009 | .0009 | .0018 | .0009 | .0018 | − .0000 | .0009 |
| 3 3/4 | 4 1/4 | 4 | .0010 | .0006 | .0006 | .0010 | .0010 | .0020 | .0010 | .0020 | − .0000 | .0010 |
| 4 1/4 | 4 3/4 | 4 1/2 | .0010 | .0007 | .0007 | .0010 | .0010 | .0021 | .0011 | .0021 | − .0001 | .0011 |
| 4 3/4 | 5 1/2 | 5 | .0010 | .0007 | .0007 | .0010 | .0010 | .0023 | .0013 | .0023 | − .0003 | .0013 |
| 5 1/2 | 6 1/2 | 6 | .0011 | .0007 | .0007 | .0011 | .0011 | .0026 | .0015 | .0026 | − .0004 | .0015 |
| 6 1/2 | 7 1/2 | 7 | .0011 | .0008 | .0008 | .0011 | .0011 | .0029 | .0018 | .0029 | − .0007 | .0018 |
| 7 1/2 | 8 1/2 | 8 | .0012 | .0008 | .0008 | .0012 | .0012 | .0032 | .0020 | .0032 | − .0008 | .0020 |

* Minimum limit = 0.0000 in. for all sizes.

† Minus here denotes interference of metal or negative allowance.

‡ Plus here denotes clearance or amount of looseness.

and general machine manufacturing. In this fit, the average interference of metal is the desired condition. It must be obtained by selective assembly, *i.e.*, by mating large shafts in large holes, and small shafts in small holes. In this fit, selected average interference of metal = 0.00025 $d$. Hole tolerances and shaft tolerances are given by the formula

$$\text{Tolerance} = 0.0006\sqrt[3]{d}.$$

Dimensions are given in Table 3. The stresses set up in the members, and the forces required for making these fits, are given in Table 4. The values of the greatest hole stress in Table 4 are for mean diameters only. For other sizes in the same group, the interference per inch of diameter, and hence the greatest hole stress will vary from the values given by less than 10%. If greater accuracy for intermediate steps is required, the interference per inch of diameter should be computed by the formula given above. The values given in Table 4 are true only when hub diameter equals twice the hole diameter.

**Class 7, Medium Force Fit, Negative Allowance.**—These fits are used where the parts are to be permanently assembled. Considerable pressure is required for assembly. They are the tightest fits recommended for cast iron holes or external members, since they will stress cast iron to its elastic limit. Examples of the use of this class of fit are the attachment to their axles or shafts of locomotive and car wheels, generator and motor armatures, and engine crank discs. This class of fit also is used as a shrink fit on medium sections or long fits. These fits are used for selective assembly only. Selected average interference of metal = 0.0005 $d$. The shaft and hole tolerances are determined by the formula

$$\text{Tolerance} = 0.0006\sqrt[3]{d}.$$

Dimensions are given in Table 5. Stresses set up in the parts, and the pressure required to make the fits, are given in Table 6. The values given in Table 6 for stress and force are true only when the hub diameter equals twice the hole diameter. The values of

### Table 4.—Stresses and Forces for Tight Fits, Class 6

A.S.A. B4a–1925

Force values are for fits 1 in. long. For longer fits multiply tabular values by length of fit in inches

| Mean size, in. | Interference of Metal per inch of Mean Size* | | Greatest Hole Stress, Steel Shaft in | | | | Force Required to Press Steel Shaft into | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Steel Hole† | | Cast-iron Hole‡ | | Steel Hole | | | Cast-iron Hole | | |
| | Tightest Fit, in. | Loosest Fit, in. | Tightest Fit, lb. per sq. in. | Loosest Fit, lb. per sq. in. | Tightest Fit, lb. per sq. in. | Loosest Fit, lb. per sq. in. | Tightest Fit, tons | Loosest Fit, tons | Selected Fit, tons | Tightest Fit, tons | Loosest Fit, tons | Selected Fit, tons |
| 1/8 | 0.00240 | No values, because of smallest shaft member in largest hole member. | 69,600 | No values, because of smallest shaft member in largest hole member. | 25,037 | No values, because of smallest shaft member in largest hole member. | 0.389 | No values, because of smallest shaft member in largest hole member. | 0.000 | 0.224 | No values, because of smallest shaft member in largest hole member. | 0.000 |
| 1/4 | .00200 | | 58,000 | | 20,864 | | .649 | | .130 | .374 | | .075 |
| 3/8 | .00133 | | 38,700 | | 13,909 | | .649 | | .130 | .374 | | .075 |
| 1/2 | .00120 | | 34,800 | | 12,518 | | .779 | | .130 | .448 | | .075 |
| 5/8 | .00112 | | 32,480 | | 11,684 | | .909 | | .260 | .523 | | .149 |
| 3/4 | .00093 | | 27,100 | | 9,737 | | .909 | | .260 | .523 | | .149 |
| 7/8 | .00091 | | 26,500 | | 9,538 | | 1.038 | | .260 | .598 | | .149 |
| 1 | .00090 | | 26,100 | | 9,389 | | 1.168 | | .389 | .672 | | .224 |
| 1 1/8 | .00080 | | 23,200 | | 8,346 | | 1.168 | | .389 | .672 | | .224 |
| 1 1/4 | .00072 | | 20,880 | | 7,511 | | 1.168 | | .389 | .672 | | .224 |
| 1 1/2 | .00073 | | 21,300 | | 7,650 | | 1.428 | | .519 | .822 | | .299 |
| 1 3/4 | .00063 | | 18,240 | | 6,557 | | 1.428 | | .519 | .822 | | .299 |
| 2 | .00065 | | 18,850 | | 6,781 | | 1.687 | | .649 | .971 | | .374 |
| 2 1/4 | .00062 | | 18,040 | | 6,491 | | 1.817 | | .779 | 1.046 | | .448 |
| 2 1/2 | .00056 | | 16,200 | | 5,842 | | 1.817 | | .779 | 1.046 | | .448 |
| 3 | .00057 | | 16,433 | | 5,911 | | 2.207 | | 1.038 | 1.270 | | .598 |
| 3 1/2 | .00051 | | 14,910 | | 5,365 | | 2.336 | | 1.168 | 1.345 | | .672 |
| 4 | .00050 | | 14,500 | | 5,216 | | 2.596 | | 1.298 | 1.494 | | .747 |
| 4 1/2 | .00047 | 0.00002 | 13,530 | 644 | 4,868 | 232 | 2.726 | 0.130 | 1.428 | 1.569 | 0.075 | .822 |
| 5 | .00046 | .00006 | 13,340 | 1740 | 4,799 | 625 | 2.985 | .389 | 1.687 | 1.719 | .224 | .971 |
| 6 | .00043 | .00007 | 12,570 | 1933 | 4,521 | 695 | 3.375 | .519 | 1.947 | 1.942 | .299 | 1.121 |
| 7 | .00041 | .00010 | 12,010 | 2900 | 4,322 | 1043 | 3.764 | .909 | 2.336 | 2.167 | .523 | 1.345 |
| 8 | .00040 | .00010 | 11,600 | 2900 | 4,173 | 1043 | 4.154 | 1.038 | 2.596 | 2.391 | .598 | 1.494 |
| | | | Stress = 29,000,000 $A/d$ § | | Stress = 10,432,000 $A/d$ § | | Force = 1298 $A$ § | | | Force = 747.2 $A$ § | | |

* Interference of metal on selected fits, all sizes, = 0.00025 in.
† Greatest stress in hole on selected fits, all sizes, = 7250 lb. per sq. in.
‡ Greatest stress in hole on selected fits, all sizes, = 2608 lb. per sq. in.
§ $A$ = allowance; $d$ = mean size.

greatest hole stress are for mean diameter sizes only. For other sizes in the group, the interference per inch of diameter, and hence the greatest hole stress will vary from the values given by less than 10%. If greater accuracy for intermediate steps is required, the interference per inch of diameter should be computed by the formula given above.

**Class 8, Heavy Force and Shrink Fit, Considerable Negative Allowance.**—These fits are used for steel holes where the metal can be highly stressed without exceeding its elastic limit. They cause excessive stress in cast-iron holes. Shrink fits are used where heavy force fits are impracticable, as on locomotive wheel tires, heavy crank discs, etc. They are for selective assembly only. Selected average interference of metal = 0.001 $d$. Hole and shaft tolerances are given by the formula

$$\text{Tolerance} = 0.0006\sqrt[3]{d}.$$

Dimensions are given in Table 7. Stresses set up by, and the force required to make these fits, are given in Table 8. The values given in Table 8 for stress and force are true only

**Table 5.—American Standard Medium Force Fit, Negative Allowance, Selective Assembly**

A.S.A. B4a–1925. All dimensions in inches

| Size | | | Limits | | | | Tightest Fit Allowance | Loosest Fit, Allowance + Tolerances | Selected Fit, Average Interference of Metal |
|---|---|---|---|---|---|---|---|---|---|
| From | Up to and Including | Mean | Hole or External Member | | Shaft or Internal Member | | | | |
| | | | Plus | | Plus | Plus | Minus* | *† | Minus* |
| 0 | 3/16 | 1/8 | 0.0003 | 0.0000 | 0.0004 | 0.0001 | 0.0004 | +0.0002 | 0.0001 |
| 3/16 | 5/16 | 1/4 | .0004 | .0000 | .0005 | .0001 | .0005 | + .0003 | .0001 |
| 5/16 | 7/16 | 3/8 | .0004 | .0000 | .0006 | .0002 | .0006 | + .0002 | .0002 |
| 7/16 | 9/16 | 1/2 | .0005 | .0000 | .0008 | .0003 | .0008 | + .0002 | .0003 |
| 9/16 | 11/16 | 5/8 | .0005 | .0000 | .0008 | .0003 | .0008 | + .0002 | .0003 |
| 11/16 | 13/16 | 3/4 | .0005 | .0000 | .0009 | .0004 | .0009 | + .0001 | .0004 |
| 13/16 | 15/16 | 7/8 | .0006 | .0000 | .0010 | .0004 | .0010 | + .0002 | .0004 |
| 15/16 | 1 1/16 | 1 | .0006 | .0000 | .0011 | .0005 | .0011 | + .0001 | .0005 |
| 1 1/16 | 1 3/16 | 1 1/8 | .0006 | .0000 | .0012 | .0006 | .0012 | .0000 | .0006 |
| 1 3/16 | 1 3/8 | 1 1/4 | .0006 | .0000 | .0012 | .0006 | .0012 | .0000 | .0006 |
| 1 3/8 | 1 5/8 | 1 1/2 | .0007 | .0000 | .0015 | .0008 | .0015 | − .0001 | .0008 |
| 1 5/8 | 1 7/8 | 1 3/4 | .0007 | .0000 | .0016 | .0009 | .0016 | − .0002 | .0009 |
| 1 7/8 | 2 1/8 | 2 | .0008 | .0000 | .0018 | .0010 | .0018 | − .0002 | .0010 |
| 2 1/8 | 2 3/8 | 2 1/4 | .0008 | .0000 | .0019 | .0011 | .0019 | − .0003 | .0011 |
| 2 3/8 | 2 3/4 | 2 1/2 | .0008 | .0000 | .0021 | .0013 | .0021 | − .0005 | .0013 |
| 2 3/4 | 3 1/4 | 3 | .0009 | .0000 | .0024 | .0015 | .0024 | − .0006 | .0015 |
| 3 1/4 | 3 3/4 | 3 1/2 | .0009 | .0000 | .0027 | .0018 | .0027 | − .0009 | .0018 |
| 3 3/4 | 4 1/4 | 4 | .0010 | .0000 | .0030 | .0020 | .0030 | − .0010 | .0020 |
| 4 1/4 | 4 3/4 | 4 1/2 | .0010 | .0000 | .0033 | .0023 | .0033 | − .0013 | .0023 |
| 4 3/4 | 5 1/2 | 5 | .0010 | .0000 | .0035 | .0025 | .0035 | − .0015 | .0025 |
| 5 1/2 | 6 1/2 | 6 | .0011 | .0000 | .0041 | .0030 | .0041 | − .0019 | .0030 |
| 6 1/2 | 7 1/2 | 7 | .0011 | .0000 | .0046 | .0035 | .0046 | − .0024 | .0035 |
| 7 1/2 | 8 1/2 | 8 | .0012 | .0000 | .0052 | .0040 | .0052 | − .0028 | .0040 |
| 8 1/2 | 9 1/2 | 9 | .0012 | .0000 | .0057 | .0045 | .0057 | − .0033 | .0045 |
| 10 1/2 | 11 1/2 | 10 | .0013 | .0000 | .0063 | .0050 | .0063 | − .0037 | .0050 |
| 11 1/2 | 13 | 12 | .0014 | .0000 | .0074 | .0060 | .0074 | − .0046 | .0060 |
| 13 | 15 | 14 | .0014 | .0000 | .0084 | .0070 | .0084 | − .0056 | .0070 |
| 15 | 17 | 16 | .0015 | .0000 | .0095 | .0080 | .0095 | − .0065 | .0080 |
| 17 | 19 | 18 | .0016 | .0000 | .0106 | .0090 | .0106 | − .0074 | .0090 |
| 19 | 22 | 20 | .0016 | .0000 | .0116 | .0100 | .0116 | − .0084 | .0100 |
| 22 | 26 | 24 | .0017 | .0000 | .0137 | .0120 | .0137 | − .0103 | .0120 |
| 26 | 30 | 28 | .0018 | .0000 | .0158 | .0140 | .0158 | − .0122 | .0140 |
| 30 | 34 | 32 | .0019 | .0000 | .0179 | .0160 | .0179 | − .0141 | .0160 |
| 34 | 38 | 36 | .0020 | .0000 | .0200 | .0180 | .0200 | − .0160 | .0180 |
| 38 | 44 | 40 | .0021 | .0000 | .0221 | .0200 | .0221 | − .0179 | .0200 |
| 44 | 52 | 48 | .0022 | .0000 | .0262 | .0240 | .0262 | − .0218 | .0240 |
| 52 | 60 | 56 | .0023 | .0000 | .0303 | .0280 | .0303 | − .0257 | .0280 |
| 60 | 68 | 64 | .0024 | .0000 | .0344 | .0320 | .0344 | − .0296 | .0320 |
| 68 | 76 | 72 | .0025 | .0000 | .0385 | .0360 | .0385 | − .0335 | .0360 |
| 76 | 88 | 80 | .0026 | .0000 | .0426 | .0400 | .0426 | − .0374 | .0400 |
| 88 | 104 | 96 | .0027 | .0000 | .0507 | .0480 | .0507 | − .0453 | .0480 |
| 104 | 120 | 112 | .0029 | .0000 | .0589 | .0560 | .0589 | − .0531 | .0560 |
| 120 | 136 | 128 | .0030 | .0000 | .0670 | .0640 | .0670 | − .0610 | .0640 |

* Minus here denotes interference of metal or negative allowance.

† Plus here denotes clearance or amount of looseness.

## Table 6.—Stresses and Forces for Medium Force Fits, Class 7

A.S.A. B4a–1925

Force values are for fits 1 in. long. For longer fits multiply tabular values by length of fit in inches

| Mean Size, in. | Interference of Metal per Inch of Mean Size * | | Greatest Hole Stress, Steel Shaft, in | | | | Force Required to Press Steel Shaft into | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Steel Hole † | | Cast-iron Hole ‡ | | Steel Hole | | | Cast-iron Hole | | |
| | Tightest Fit, in. | Loosest Fit, in. | Tightest Fit, in. | Loosest Fit, in. | Tightest Fit, in. | Loosest Fit, in. | Tightest Fit, tons | Loosest Fit, tons | Selected Fit, tons | Tightest Fit, tons | Loosest Fit, tons | Selected Fit, tons |
| 1/8 | 0.00320 | See Note 1 | 92,800 | See Note 1 | 33,382 | See Note 1 | 0.519 | See Note 1 | 0.130 | 0.299 | See Note 1 | 0.075 |
| 1/4 | .00200 | | 58,000 | | 20,864 | | .649 | | .130 | .374 | | .075 |
| 3/8 | .00160 | | 46,400 | | 16,691 | | .779 | | .260 | .448 | | .149 |
| 1/2 | .00160 | | 46,400 | | 16,691 | | 1.038 | | .389 | .598 | | .224 |
| 5/8 | .00128 | | 37,120 | | 13,353 | | 1.038 | | .389 | .598 | | .224 |
| 3/4 | .00120 | | 34,800 | | 12,513 | | 1.168 | | .519 | .672 | | .299 |
| 7/8 | .00114 | | 33,140 | | 11,922 | | 1.298 | | .519 | .747 | | .299 |
| 1 | .00110 | | 31,900 | | 11,475 | | 1.428 | | .649 | .822 | | .374 |
| 1 1/8 | .00107 | | 30,930 | | 11,127 | | 1.558 | | .779 | .897 | | .448 |
| 1 1/4 | .00096 | | 27,830 | | 10,015 | | 1.558 | | .779 | .897 | | .448 |
| 1 1/2 | .00100 | 0.00007 | 29,000 | 1,934 | 10,432 | 695 | 1.947 | 0.130 | 1.038 | 1.121 | 0.075 | .598 |
| 1 3/4 | .00091 | .00011 | 26,510 | 3,314 | 9,538 | 1192 | 2.077 | .260 | 1.168 | 1.196 | .149 | .672 |
| 2 | .00090 | .00010 | 26,100 | 2,900 | 9,389 | 1043 | 2.336 | .260 | 1.298 | 1.345 | .149 | .747 |
| 2 1/4 | .00084 | .00013 | 24,490 | 3,866 | 8,809 | 1391 | 2.466 | .389 | 1.428 | 1.420 | .224 | .822 |
| 2 1/2 | .00084 | .00020 | 24,340 | 5,800 | 8,763 | 2086 | 2.726 | .649 | 1.687 | 1.569 | .373 | .971 |
| 3 | .00080 | .00020 | 23,200 | 5,800 | 8,346 | 2086 | 3.115 | .779 | 1.947 | 1.793 | .448 | 1.121 |
| 3 1/2 | .00077 | .00026 | 22,370 | 7,457 | 8,048 | 2682 | 3.505 | 1.168 | 2.336 | 2.017 | .672 | 1.345 |
| 4 | .00075 | .00025 | 21,750 | 7,250 | 7,824 | 2608 | 3.894 | 1.298 | 2.596 | 2.242 | .747 | 1.494 |
| 4 1/2 | .00073 | .00029 | 21,270 | 8,377 | 7,652 | 3014 | 4.283 | 1.687 | 2.985 | 2.466 | .971 | 1.719 |
| 5 | .00070 | .00030 | 20,300 | 8,700 | 7,302 | 3130 | 4.543 | 1.947 | 3.245 | 2.615 | 1.121 | 1.868 |
| 6 | .00068 | .00032 | 19,810 | 9,183 | 7,129 | 3303 | 5.322 | 2.466 | 3.894 | 3.064 | 1.420 | 2.242 |
| 7 | .00066 | .00034 | 19,060 | 9,942 | 6,855 | 3577 | 5.971 | 3.115 | 4.543 | 3.437 | 1.793 | 2.615 |
| 8 | .00065 | .00035 | 18,850 | 10,150 | 6,781 | 3651 | 6.750 | 3.634 | 5.192 | 3.885 | 2.092 | 2.989 |
| 9 | .00063 | .00037 | 18,370 | 10,630 | 6,607 | 3875 | 7.399 | 4.283 | 5.841 | 4.259 | 2.466 | 3.362 |
| 10 | .00063 | .00037 | 18,270 | 10,730 | 6,572 | 3860 | 8.177 | 4.803 | 6.490 | 4.707 | 2.765 | 3.736 |
| 12 | .00062 | .00038 | 17,890 | 11,120 | 6,433 | 3999 | 9.605 | 5.971 | 7.788 | 5.529 | 3.437 | 4.483 |
| 14 | .00060 | .00040 | 17,400 | 11,600 | 6,259 | 4173 | 10.903 | 7.269 | 9.068 | 6.276 | 4.184 | 5.230 |
| 16 | .00059 | .00041 | 17,220 | 11,780 | 6,194 | 4238 | 12.331 | 8.437 | 10.384 | 7.098 | 4.857 | 5.978 |
| 18 | .00059 | .00041 | 17,080 | 11,920 | 6,143 | 4289 | 13.759 | 9.605 | 11.682 | 7.920 | 5.529 | 6.725 |
| 20 | .00058 | .00042 | 16,820 | 12,180 | 6,051 | 4381 | 15.057 | 10.903 | 12.980 | 8.668 | 6.276 | 7.472 |
| 24 | .00057 | .00043 | 16,550 | 12,450 | 5,955 | 4477 | 17.783 | 13.369 | 15.576 | 10.236 | 7.696 | 8.966 |
| 28 | .00056 | .00044 | 16,360 | 12,640 | 5,887 | 4173 | 20.508 | 14.538 | 18.172 | 11.806 | 8.369 | 10.461 |
| 32 | .00056 | .00044 | 16,220 | 12,780 | 5,835 | 4597 | 23.234 | 18.302 | 20.768 | 13.375 | 10.536 | 11.955 |
| 36 | .00056 | .00044 | 16,110 | 12,890 | 5,796 | 4636 | 25.960 | 20.768 | 23.364 | 14.944 | 11.955 | 13.450 |
| 40 | .00055 | .00045 | 16,020 | 12,980 | 5,764 | 4668 | 28.686 | 23.234 | 25.960 | 16.513 | 13.375 | 14.944 |
| 48 | .00055 | .00045 | 15,830 | 13,170 | 5,694 | 4738 | 34.008 | 28.296 | 31.152 | 19.577 | 16.289 | 17.933 |
| 56 | 00054 | .00046 | 15,690 | 13,310 | 5,644 | 4788 | 39.329 | 33.359 | 36.344 | 22.640 | 19.203 | 20.922 |
| 64 | 00054 | .00046 | 15,570 | 13,410 | 5,607 | 4825 | 44.651 | 38.421 | 41.536 | 25.704 | 22.117 | 23.910 |
| 72 | .00053 | .00047 | 15,500 | 13,490 | 5,578 | 4854 | 49.973 | 43.483 | 46.728 | 28.767 | 25.031 | 26.899 |
| 80 | .00053 | .00047 | 15,440 | 13,560 | 5,555 | 4877 | 55.295 | 48.545 | 51.920 | 31.831 | 27.945 | 29.888 |
| 96 | .00053 | .00047 | 15,320 | 13,680 | 5,509 | 4923 | 65.809 | 58.799 | 62.304 | 37.883 | 33.848 | 35.866 |
| 112 | .00053 | .00047 | 15,250 | 13,750 | 5,486 | 4946 | 76.452 | 68.924 | 72.688 | 44.010 | 39.676 | 41.843 |
| 128 | .00052 | .00048 | 15,180 | 13,820 | 5,461 | 4972 | 86.966 | 79.178 | 83.072 | 50.062 | 45.579 | 47.821 |
| | | | Stress = 29,000,000A/d§ | | Stress = 10,432,000A/d§ | | Force = 1298 A § | | | Force = 747.2A § | | |

Note 1.—No values are given for sizes 1/8 in. to 1 1/4 in. inclusive, because of the absence of interference of metal, due to the smallest shaft member being smaller than the largest hole member.

* Interference on selected fits, all sizes, = 0.0005 in.

† Greatest hole stress on selected fits, all sizes, = 14,500 lb. per sq. in.

‡ Greatest hole stress on selected fits, all sizes, = 5216 lb. per sq. in.

§ $A$ = allowance; $d$ = mean size.

when the hub diameter equals twice the hole diameter. The values of the greatest hole stress are for mean diameter sizes only. For other sizes in the group, the interference per inch of diameter, and hence the greatest hole stress, will vary from the values given by less than 10%. If greater accuracy for intermediate steps is required, the interference per inch of diameter should be computed by the formula given above.

STRESSES RESULTING FROM FORCE FITS.—In any force or shrink fit, the metal of the fitted member is deformed by an amount equal to the interference of metal between the external and internal members. Discussions of the stress set up by this deformation usually are based on the assumption that the materials are elastic and follow Hooke's law.

### Table 7.—American Standard Heavy Force and Shrink Fit, Considerable Negative Allowance, Selective Assembly, Class 8

A.S.A., B4a–1925. All dimensions in inches

| Size | | | Limits | | | | Tightest Fit, Allowance | Loosest Fit, Allowance + Tolerances | Selected Fit, Average Interference of Metal |
|---|---|---|---|---|---|---|---|---|---|
| From | Up to and Including | Mean | Hole or External Member | | Shaft or Internal Member | | | | |
| | | | Plus | | Plus | Plus | Minus* | *† | Minus* |
| 0 | 3/16 | 1/8 | 0.0003 | 0.0000 | 0.0004 | 0.0001 | 0.0004 | +0.0002 | 0.0001 |
| 3/16 | 5/16 | 1/4 | .0004 | .0000 | .0007 | .0003 | .0007 | + .0001 | .0003 |
| 5/16 | 7/16 | 3/8 | .0004 | .0000 | .0008 | .0004 | .0008 | .0000 | .0004 |
| 7/16 | 9/16 | 1/2 | .0005 | .0000 | .0010 | .0005 | .0010 | .0000 | .0005 |
| 9/16 | 11/16 | 5/8 | .0005 | .0000 | .0011 | .0006 | .0011 | − .0001 | .0006 |
| 11/16 | 13/16 | 3/4 | .0005 | .0000 | .0013 | .0008 | .0013 | − .0003 | .0008 |
| 13/16 | 15/16 | 7/8 | .0006 | .0000 | .0015 | .0009 | .0015 | − .0003 | .0009 |
| 15/16 | 1 1/16 | 1 | .0006 | .0000 | .0016 | .0010 | .0016 | − .0004 | .0010 |
| 1 1/16 | 1 3/16 | 1 1/8 | .0006 | .0000 | .0017 | .0011 | .0017 | − .0005 | .0011 |
| 1 3/16 | 1 3/8 | 1 1/4 | .0006 | .0000 | .0019 | .0013 | .0019 | − .0007 | .0013 |
| 1 3/8 | 1 5/8 | 1 1/2 | .0007 | .0000 | .0022 | .0015 | .0022 | − .0008 | .0015 |
| 1 5/8 | 1 7/8 | 1 3/4 | .0007 | .0000 | .0025 | .0018 | .0025 | − .0011 | .0018 |
| 1 7/8 | 2 1/8 | 2 | .0008 | .0000 | .0028 | .0020 | .0028 | − .0012 | .0020 |
| 2 1/8 | 2 3/8 | 2 1/4 | .0008 | .0000 | .0031 | .0023 | .0031 | − .0015 | .0023 |
| 2 3/8 | 2 3/4 | 2 1/2 | .0008 | .0000 | .0033 | .0025 | .0033 | − .0017 | .0025 |
| 2 3/4 | 3 1/4 | 3 | .0009 | .0000 | .0039 | .0030 | .0039 | − .0021 | .0030 |
| 3 1/4 | 3 3/4 | 3 1/2 | .0009 | .0000 | .0044 | .0035 | .0044 | − .0026 | .0035 |
| 3 3/4 | 4 1/4 | 4 | .0010 | .0000 | .0050 | .0040 | .0050 | − .0030 | .0040 |
| 4 1/4 | 4 3/4 | 4 1/2 | .0010 | .0000 | .0055 | .0045 | .0055 | − .0035 | .0045 |
| 4 3/4 | 5 1/2 | 5 | .0010 | .0000 | .0060 | .0050 | .0060 | − .0040 | .0050 |
| 5 1/2 | 6 1/2 | 6 | .0011 | .0000 | .0071 | .0060 | .0071 | − .0049 | .0060 |
| 6 1/2 | 7 1/2 | 7 | .0011 | .0000 | .0081 | .0070 | .0081 | − .0059 | .0070 |
| 7 1/2 | 8 1/2 | 8 | .0012 | .0000 | .0092 | .0080 | .0092 | − .0068 | .0080 |
| 8 1/2 | 9 1/2 | 9 | .0012 | .0000 | .0102 | .0090 | .0102 | − .0078 | .0090 |
| 10 1/2 | 11 1/2 | 10 | .0013 | .0000 | .0113 | .0100 | .0113 | − .0087 | .0100 |
| 11 1/2 | 13 | 12 | .0014 | .0000 | .0134 | .0120 | .0134 | − .0106 | .0120 |
| 13 | 15 | 14 | .0014 | .0000 | .0154 | .0140 | .0154 | − .0126 | .0140 |
| 15 | 17 | 16 | .0015 | .0000 | .0175 | .0160 | .0175 | − .0145 | .0160 |
| 17 | 19 | 18 | .0016 | .0000 | .0196 | .0180 | .0196 | − .0164 | .0180 |
| 19 | 22 | 20 | .0016 | .0000 | .0216 | .0200 | .0216 | − .0184 | .0200 |
| 22 | 26 | 24 | .0017 | .0000 | .0257 | .0240 | .0257 | − .0223 | .0240 |
| 26 | 30 | 28 | .0018 | .0000 | .0298 | .0280 | .0298 | − .0262 | .0280 |
| 30 | 34 | 32 | .0019 | .0000 | .0339 | .0320 | .0339 | − .0301 | .0320 |
| 34 | 38 | 36 | .0020 | .0000 | .0380 | .0360 | .0380 | − .0340 | .0360 |
| 38 | 44 | 40 | .0021 | .0000 | .0421 | .0400 | .0421 | − .0379 | .0400 |
| 44 | 52 | 48 | .0022 | .0000 | .0502 | .0480 | .0502 | − .0458 | .0480 |
| 52 | 60 | 56 | .0023 | .0000 | .0583 | .0560 | .0583 | − .0537 | .0560 |
| 60 | 68 | 64 | .0024 | .0000 | .0664 | .0640 | .0664 | − .0616 | .0640 |
| 68 | 76 | 72 | .0025 | .0000 | .0745 | .0720 | .0745 | − .0695 | .0720 |
| 76 | 88 | 80 | .0026 | .0000 | .0826 | .0800 | .0826 | − .0774 | .0800 |
| 88 | 104 | 96 | .0027 | .0000 | .0987 | .0960 | .0987 | − .0933 | .0960 |
| 104 | 120 | 112 | .0029 | .0000 | .1149 | .1120 | .1149 | − .1091 | .1120 |
| 120 | 136 | 128 | .0030 | .0000 | .1310 | .1280 | .1310 | − .1250 | .1280 |

* Minus here denotes interference of metal or negative allowance.
† Plus here denotes clearance or amount of looseness.

A. Lewis Jenkins (*Am. Mach.*, March 4, 1915) shows that for the usual combination of a cast-iron hub shrunk on a steel shaft, Hooke's law does not apply to the cast-iron hub. From a study of a wide variety of force and shrink fits, he deduced the following equations: Let $p$ = the unit stress between shaft and hub; $r_2$ = common radius of fitted shaft and hub; $r_3$ = external radius of hub; $e$ = total deformation of shaft and hub in pressing $= r_s - r_h$; $r_s$, $r_h$ = respectively, external radius of shaft and internal radius of hub before pressing. Then for cast-iron hubs on steel shafts

$$p = \{102{,}900{,}000\ (e/r_2)\} \div \{7.16 + (r_3/r_2)\}.$$

For steel hubs on steel shafts

$$p = [15{,}000{,}000\ (e/r_2)\{(r_3/r_2)^2 + 1\}] \div \{(r_3/r_2)^2\}.$$

In designing force and shrink fits, the interference $e$ usually is assumed. The resulting stress then should be calculated to make sure that the elastic limit of the material is not exceeded.

### Table 8.—Stresses and Forces for Heavy Force and Shrink Fits, Class 8

A.S.A. B4a–1925

Force values are for fits 1 in. long. For longer fits, multiply tabular values by length of fit in inches

| Mean Size, in. | Interference of Metal per Inch of Mean Size | | | Greatest Hole Stress, Steel Shaft in Steel Hole | | | Force Required to Press Steel Shaft into Steel Hole | | |
|---|---|---|---|---|---|---|---|---|---|
| | Tightest Fit, in. | Loosest Fit, in. | Selected Fit, in. | Tightest Fit, lb. per sq. in. | Loosest Fit, lb. per sq. in. | Selected Fit, lb. per sq. in. | Tightest Fit, tons | Loosest Fit, tons | Selected Fit, tons |
| 1/8 | 0.00320 | See Note 1 | 0.001 | 92,800 | See Note 1 | 29,000 | 0.519 | See Note 1 | 0.130 |
| 1/4 | .00280 | | .001 | 81,200 | | 29,000 | .909 | | .389 |
| 3/8 | .00213 | | .001 | 61,870 | | 29,000 | 1.038 | | .519 |
| 1/2 | .00200 | | .001 | 58,000 | | 29,000 | 1.298 | | .649 |
| 5/8 | .00176 | 0.00016 | .001 | 51,040 | 4,640 | 29,000 | 1.428 | 0.130 | .779 |
| 3/4 | .00173 | .00040 | .001 | 42,270 | 11,600 | 29,000 | 1.687 | .389 | 1.038 |
| 7/8 | .00171 | .00034 | .001 | 49,710 | 9,943 | 29,000 | 1.947 | .389 | 1.168 |
| 1 | .00160 | .00040 | .001 | 46,400 | 11,600 | 29,000 | 2.077 | .519 | 1.298 |
| 1 1/8 | .00151 | .00044 | .001 | 43,840 | 12,890 | 29,000 | 2.207 | .649 | 1.428 |
| 1 1/4 | .00152 | .00056 | .001 | 44,080 | 16,230 | 29,000 | 2.466 | .909 | 1.687 |
| 1 1/2 | .00147 | .00053 | .001 | 42,530 | 15,470 | 29,000 | 2.856 | 1.038 | 1.947 |
| 1 3/4 | .00143 | .00062 | .001 | 41,430 | 18,230 | 29,000 | 3.245 | 1.428 | 2.336 |
| 2 | .00140 | .00060 | .001 | 40,600 | 17,400 | 29,000 | 3.634 | 1.558 | 2.596 |
| 2 1/4 | .00138 | .00066 | .001 | 39,960 | 17,910 | 29,000 | 4.024 | 1.947 | 2.985 |
| 2 1/2 | .00132 | .00068 | .001 | 38,280 | 19,720 | 29,000 | 4.283 | 2.207 | 3.245 |
| 3 | .00130 | .00070 | .001 | 37,700 | 20,300 | 29,000 | 5.062 | 2.726 | 3.894 |
| 3 1/2 | .00126 | .00074 | .001 | 36,460 | 21,540 | 29,000 | 5.711 | 3.375 | 4.543 |
| 4 | .00125 | .00075 | .001 | 36,250 | 21,750 | 29,000 | 6.490 | 3.894 | 5.192 |
| 4 1/2 | .00122 | .00078 | .001 | 35,440 | 22,560 | 29,000 | 7.139 | 4.543 | 5.841 |
| 5 | .00120 | .00080 | .001 | 34,800 | 23,200 | 29,000 | 7.788 | 5.192 | 6.490 |
| 6 | .00118 | .00082 | .001 | 34,317 | 23,680 | 29,000 | 9.216 | 6.360 | 7.788 |
| 7 | .00116 | .00084 | .001 | 33,570 | 24,440 | 29,000 | 10.514 | 7.658 | 9.086 |
| 8 | .00115 | .00085 | .001 | 33,350 | 24,650 | 29,000 | 11.942 | 8.826 | 10.384 |
| 9 | .00113 | .00087 | .001 | 32,870 | 25,130 | 29,000 | 13.240 | 10.124 | 11.682 |
| 10 | .00113 | .00087 | .001 | 32,770 | 25,233 | 29,000 | 14.667 | 11.293 | 12.980 |
| 12 | .00112 | .00088 | .001 | 32,390 | 25,610 | 29,000 | 17.393 | 13.759 | 15.576 |
| 14 | .00110 | .00090 | .001 | 31,900 | 26,100 | 29,000 | 19.989 | 16.355 | 18.172 |
| 16 | .00109 | .00091 | .001 | 31,720 | 26,290 | 29,000 | 22.715 | 18.821 | 20.768 |
| 18 | .00108 | .00091 | .001 | 31,580 | 26,420 | 29,000 | 25.441 | 21.287 | 23.364 |
| 20 | .00108 | .00092 | .001 | 31,370 | 26,680 | 29,000 | 28.037 | 23.883 | 25.960 |
| 24 | .00107 | .00093 | .001 | 31,050 | 26,940 | 29,000 | 33.359 | 28.945 | 31.152 |
| 28 | .00106 | .00094 | .001 | 30,860 | 27,130 | 29,000 | 38.680 | 34.008 | 36.344 |
| 32 | .00106 | .00094 | .001 | 30,710 | 27,280 | 29,000 | 44.002 | 39.070 | 41.536 |
| 36 | .00106 | .00094 | .001 | 30,600 | 27,390 | 29,000 | 49.324 | 44.132 | 46.728 |
| 40 | .00105 | .00095 | .001 | 30,510 | 27,480 | 29,000 | 54.646 | 49.194 | 51.920 |
| 48 | .00105 | .00095 | .001 | 30,330 | 27,680 | 29,000 | 65.160 | 59.448 | 62.304 |
| 56 | .00104 | .00096 | .001 | 30,190 | 27,810 | 29,000 | 75.673 | 69.703 | 72.688 |
| 64 | .00104 | .00096 | .001 | 30,080 | 27,900 | 29,000 | 86.187 | 79.957 | 83.072 |
| 72 | .00103 | .00097 | .001 | 30,000 | 27,990 | 29,000 | 96.701 | 90.211 | 93.456 |
| 80 | .00103 | .00097 | .001 | 29,940 | 28,060 | 29,000 | 107.113 | 100.465 | 103.840 |
| 96 | .00103 | .00097 | .001 | 29,810 | 28,180 | 29,000 | 128.115 | 121.103 | 124.608 |
| 112 | .00103 | .00097 | .001 | 29,750 | 28,250 | 29,000 | 149.140 | 141.612 | 145.736 |
| 128 | .00102 | .00098 | .001 | 29,690 | 28,320 | 29,000 | 170.038 | 162.250 | 166.144 |
| | | | | Stress = 29,000,000 $A/d$ * | | | Force = 1298$A$ * | | |

* $A$ = allowance; $d$ = mean size.

NOTE 1. No values are given for sizes 1/8 in. to 1/2 in. inclusive because of the absence of intererence of metal, due to the smallest shaft member being smaller than the largest hole member.

**TEMPERATURES FOR SHRINKAGE FITS.**—The temperature to which the outer member in a shrinkage fit must be heated in order to assemble the parts depends upon the coefficient of linear expansion of the metal. For wrought iron and steel this is about 0.000 0065 for each degree F. and for cast-iron it is 0.000 0062. Low redness corresponds to about 900° F. and hence causes an expansion of the bore of about 0.005 in. per in. of diam.

# Section 13

# BEARINGS AND LUBRICANTS

## OIL FILM BEARINGS WITH PERFECT LUBRICATION

By H. A. S. Howarth

## LUBRICANTS

By J. C. Geniesse

## BALL BEARINGS

By J. S. Tawresey

## ROLLER BEARINGS

By Robert T. Kent

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# BEARINGS AND LUBRICATION

## CLASSIFICATION

Bearings may be classified according to Function, Form, Design, and Lubrication.

**CLASSIFICATION BY FUNCTION.**—Bearings may be defined according to function as follows:

**Radial Bearing.**—A bearing applied to a rotating shaft to hold its axis in line, and prevent movement in a radial direction.

**Thrust Bearing.**—A bearing applied to a shaft to prevent free endwise movement.

**Angular Bearing.**—A bearing to constrain a shaft against both radial and axial movement; the load is received in an angular direction and has radial and axial components.

**Guides or Ways.**—Bearings to permit and control the rectilinear movement of a sliding machine element, as a ram or crosshead, or slide. Guides usually are designed to prevent rotational movement of the sliding element.

**CLASSIFICATION BY FORM.**—Bearings may be divided into the following forms, depending on the way in which friction is produced in them:

**Sliding Surface Bearings** include all that depend on lubricating films, perfect or imperfect, of oil, grease, graphite or other material.

**Rolling Contact Bearings** include all forms of ball and roller bearings.

**Sliding Rolling Face Bearings** include those found in gear teeth, cams, and some forms of roller bearings.

**CLASSIFICATION BY DESIGN** may be made as follows:

**Self-aligning Bearings** are those in which the parts will automatically align themselves when assembled and loaded.

**Rigid Bearings.**—Bearings in which bearing surfaces are practically invariable in form and position after installation. Also known as non-aligning bearings.

**Pivoted Segment Bearings.**—Bearings in which one of the active surfaces is made up of segments so freely mounted as to permit the formation of natural wedge-shaped lubricating films.

**Elastic Bearings.**—Those in which one of the active surfaces is elastic or elastically supported so as to flex with respect to the other for the purpose of alignment and film formation.

**CLASSIFICATION BY LUBRICATION.—Bath Lubricated Bearings.**—Bearings in which active surfaces always are submerged in a well of lubricant.

**Automatically Lubricated or Self-oiling Bearings.**—Bearings in which the rotation of the shaft or movement of parts causes lubricant to be applied continuously to the bearing surfaces.

**Independent Pressure-lubricated Bearings.**—Bearings lubricated from a pressure source, independent of the bearing mountings, which may be operated before the machine is started, or may be geared into the machine, operating only when the machine runs. The system is then known as *Automatic Pressure Lubrication.*

**Periodically-lubricated Bearings** include all those using any form of intermittent application, by whatever means the lubricant is applied.

**FACTORS IN BEARING DESIGN AND OPERATION.**—The several factors listed below are important to the designer and user of bearings, from the standpoint of permissible loads, lubrication, etc.

**Surface Speed** rarely is a limitation with perfectly lubricated bearings. See Cooling of Bearings, p. 13–52.

**Straight-line Motion** of sliding bearing surfaces may be continuous in one direction, as wire in a drawing die. It may be intermittent in one direction as in the ram of a ratchet jack. It may be reciprocating motion, as in crossheads or machine guides.

**Circular Sliding Motion,** as in journal, thrust, and angular bearings may be continuous in one direction, or it may be reversible, *i.e.*, rotation in opposite directions for equal or unequal periods of time. It may be oscillating, as in a shaft carrying a rocking lever, or intermittent as in bearings for a ratchet wheel shaft. The angle of rotation is important in the last two classes.

**Approach and Recession** of bearing faces frequently is superimposed on sliding motion, as in rocking levers, wrist pins, and reversing shafts. It is caused by change in magnitude of bearing load, by reversal, shock or other cycle of load variation.

**Rotating Radial Load** is found more often in vertical than in horizontal shafts. It is an unusual condition, generally due to lack of balance, accidental or otherwise, but which may result as in rock crushers from the process of manufacture carried on in the machine.

# OIL FILM BEARINGS WITH PERFECT LUBRICATION

**By H. A. S. Howarth**

## 1. JOURNAL BEARINGS

**CLASSIFICATION.**—Bearings may be integral with the machine frame or separate and bolted thereto. The *bearing surface* may be *a*, integral with the machine frame; *b*, integral with an element of it called the bearing; or *c*, formed on a replaceable shell. The bearing surface usually is formed of softer metal than the material of the machine frame or of the bearing shell. See Bearing Metals, p. 4–62. If the bearing metal is cast into spaces in the machine frame, it would be classed as an integral bearing, as it is not readily replaceable. Hence, bearings may be classified broadly as integral or renewable bearings.

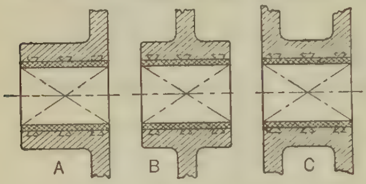


FIG. 1. Integral Bearings

**Integral Bearings.**—In the design of bearings, shaft deflection should be considered so that the deflection of the bearing, if appreciable, will tend to match that of the shaft. In Fig. 1, designs A and B may compensate for shaft bending. Design C conveys the idea of rigidity.

Integral bearings may be solid or split. The simplest joint is a plane kept in alignment by tapered or cylindrical dowels or fitted bolts. See Fig. 2. Fig. 3 shows a form of integral bearing commonly used in heavy machinery and in the bases of engines whose bearings invariably are capped. The vertical faces *aa* should fit closely and be of adequate area. Shims between the cap and frame take up clearance that otherwise might develop a bearing knock. Better forms of this type of bearing are those with a renewable shell, shown in Figs. 6 and 7.

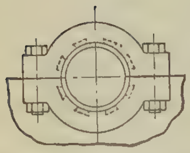


FIG. 2. Simple Split Bearing

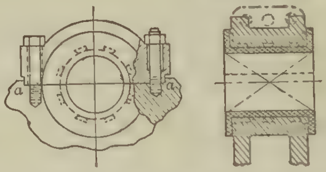


FIG. 3. Integral Bearing for Heavy Machinery

**Renewable Bearings.**—In general, renewable bearings have shells that either are *fixed* or *self-aligning*. The simplest form of renewable bearings is a plain cylindrical shell, lined or unlined, as in Fig. 4, which is a plain shell held by a dog-pointed set screw. It cannot stand much endwise bumping. Other forms are: Single-flanged shell, Figs. 5 and 11A, the former being used either split or solid; double flanged shells, Figs. 6A and 11B, which may be solid or split. Figs. 6B and 11C are forms of shell that have advantages in oil application not possessed by the others. The foregoing fixed shells in some instances may be used as renewable linings for self-aligning shells.

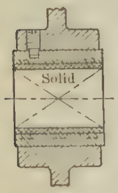


FIG. 4. Plain Renewable Shell

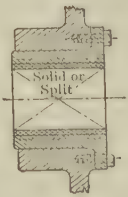


FIG. 5. Single-flanged Shell

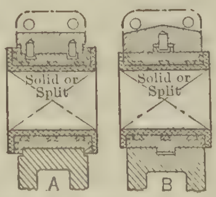


FIG. 6. Shells with Babbitt Faced Ends

The shells in Fig. 7B and C may be fixed or self-aligning, depending on the proportions and the amount of end play at the faced portions of the frame or mounting. They may be solid, or split and bolted. Because of their limited contact with the frame, the shells should be made strong and fairly hard. A soft metal lining is used in the larger sizes and in all sizes where the shell is of soft steel.

Self-aligning shells are used where ready alignment with the shaft journals is required at installation or under shaft deflection in service, or to compensate inaccuracies produced by assembly of parts, by warping, by machining, and by frame deflections. The original alignment may have to be maintained or to change with operating conditions, and shells should be designed for correction of initial and subsequent misalignment. The ball-seated shell is widely used in shaft hangers, pillow blocks and pedestals. The diameter of the spherical seats relative to the shaft is governed by the load to be carried. In general, the smaller the sphere the easier will it align the shell. Also the wider the shell, the easier it will align itself on a given spherical fit. Consequently, narrow, heavily-loaded bearing shells will not be sensitively self-aligning with the spherical proportions of Fig. 8A. Those of Fig. 8B are better. Fig. 8C shows a shell with adjustable sphere diameter used for heavy machinery. Careful setting of spherical clearances will allow for expansion in operation. The spherical angle should be as large as practicable.

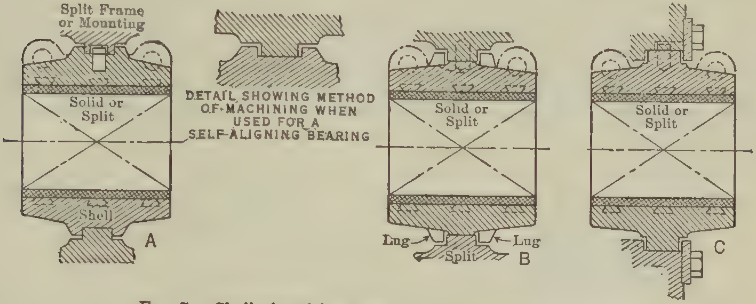


Fig. 7. Shells for either Fixed or Self-aligning Bearings.

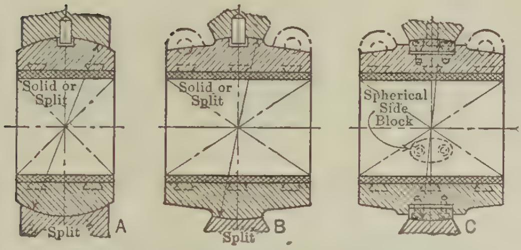


Fig. 8. Spherical-seated Self-aligning Shells

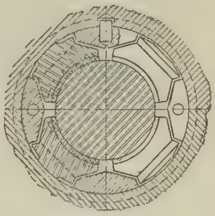


Fig. 9. Pivoted-shoe Shell, Self-aligning

The pivoted-shoe shell, Fig. 9, may be used in the space provided for an ordinary shell, like Fig. 7A, if the latter has ample cavity. Under ordinary conditions, the pivoted-shoe bearing would not be used, because its capacity generally is less than the plain journal bearing shell of similar dimensions, but under special conditions it is useful, as in very narrow self-aligning bearings. For heavy loads, hardened pivot contacts should be used in shoes and mountings.

A bearing shell with a relatively thin web, Fig. 10, can flex enough to satisfy minor shaft deflections or small errors in alignment. This form has possibilities when combined with the frame of the machine as in Fig. 1B.

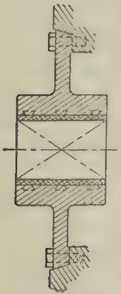


Fig. 10. Self-aligning Web-mounted Shell

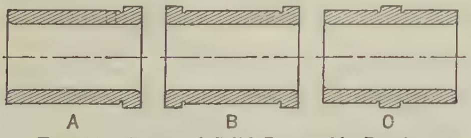


Fig. 11. Types of Solid Renewable Bearing Shells

**STANDARDIZATION OF BEARING ELEMENTS.**—Journal bearings of the film type usually are standardized as to mountings only. The fixed shell forms, Fig. 11A, B and C, lend themselves to standardization and sale of elements. Although solid bronze is widely used, there is a growing tendency to line such shells with babbitt. Many foundries carry large stocks of bronze bushings in cast form.

The self-aligning shell forms, Fig. 7, are suitable for standardization because they involve only cylindrical boring and facing of the mounting to receive them. Spherical seated shells alone are unsuited for separate sale when made as in Fig. 8, but if the two mating surfaces are sold together, as illustrated in Fig. 12, standardization is possible. The spherically bored ring around the shell of Fig. 13 has the same outside proportions as standard ball bearings offered for the same shaft. Ring oiling as shown here also is possible with bearings shown in Figs. 12A and 12C. Standardized mountings for journal bearings are available in many forms. Proportions of a few of these are given on p. 13–10. Catalogs of manufacturers of power transmission and of electrical machinery should be consulted for dimensions of other forms. See p. 24–05.

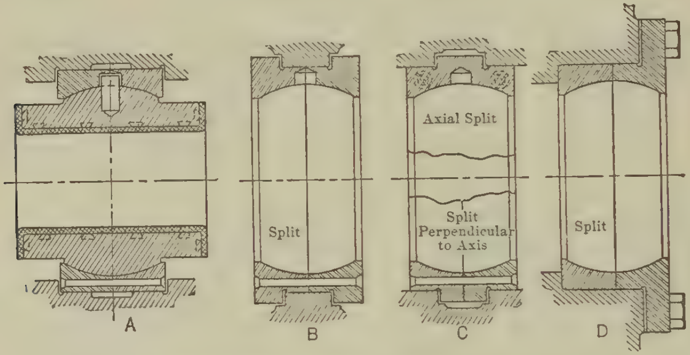


Fig. 12. Forms of Spherically-seated Bearing Saddles for Shells

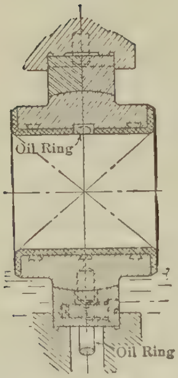


Fig. 13. Spherically-seated Shell and Saddle with Ring Oiling

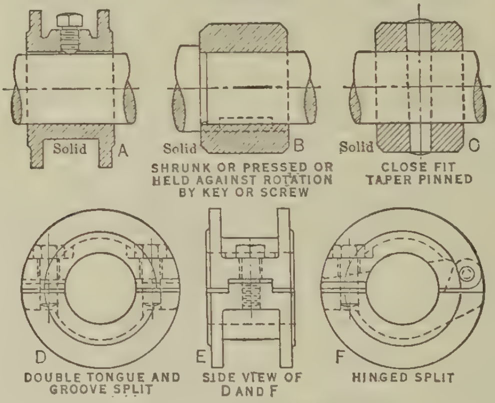

Solid
A
Solid
B
SHRUNK OR PRESSED OR HELD AGAINST ROTATION BY KEY OR SCREW
Solid
C
CLOSE FIT TAPER PINNED
D
DOUBLE TONGUE AND GROOVE SPLIT
E
SIDE VIEW OF D AND F
F
HINGED SPLIT

Fig. 14. Types of Shaft Collars

**CONTROL OF END PLAY.**—When no appreciable thrust load is present, end play is controlled by shoulders on the shaft. For such construction, the bearing-metal lining must bear against the shoulder as in Figs. 3, 6, 11, 12A and 13. For closer control of end play and for heavy axial loads see thrust bearing construction, p. 13–33. If the shoulder on a shaft is too small to control end play, separate collars of the forms shown in Fig. 14, are used. The face that forms the bearing should be square, flat and smooth.

**OIL GROOVES IN JOURNAL BEARINGS.**—Oil should be applied to a journal bearing at the point where there is no load, or where the load is lightest if it changes its radial direction. Grooves, therefore, may be used for the following purposes: 1. The application of oil. 2. The collection of side leakage. 3. The pumping of oil across bearing surfaces.

**Application Grooves** should be cut parallel with the shaft. The length may vary from an oil hole diameter at the middle of the bearing to the full length of the bearing surface. Such grooves usually do not extend to the sides of the bearing surface. See Fig. 15. While two such grooves frequently are cut diametrically opposite in a plane perpendicular to the direction of the load, one usually is enough. The grooves should be rounded on the edges parallel to the shaft and the raised edge resulting from cutting the groove scraped down. If the bearing revolves about a fixed shaft, the groove should be in the shaft on the unloaded side with oil supplied through the shaft to the groove.

If a load exerted by a shaft turning in a fixed bearing acts successively in all radial directions with a known portion of the shaft journal subjected to little or no load, the oil grooves should be located in the shaft at that place if practicable. Oil may be supplied to the groove through the shaft or from a ring groove at one side of the bearing, registering with one end of the groove in the shaft. Fig. 16 shows grooving for use where the direction of the radial load is unknown. This arrangement presents a fair bearing surface to the journal whenever the region of closest approach of the bearing surfaces passes over an oil groove. With pressure oiling, the circumferential groove may be omitted, thereby increasing the carrying capacity of the bearing.

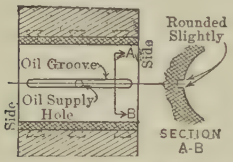


Fig. 15. Application Oil Groove

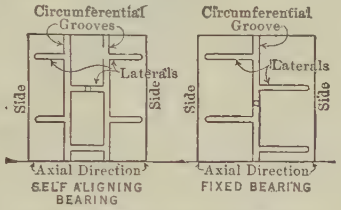


Fig. 16. Grooving where Direction of Radial Load is Unknown

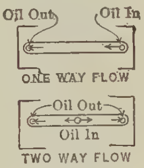


Fig. 17. Oil Grooves with Outlets

High-speed journals are cooled by continuously changing the oil. It then is necessary to provide oil outlets at the ends of the application grooves, as in Fig. 17. The grooves must be large enough to pass the desired quantity of oil with the pressure available. In Fig. 16 outlets would be provided at the ends of the laterals. The cross-section of the application groove may be, in general, similar to that shown in Fig. 15. If the grooves are used to limit the areas of the bearing surface, they sometimes are enlarged as shown in Fig. 18.

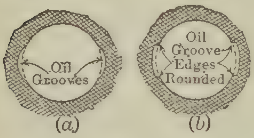


Fig. 18. Methods of Enlarging Oil Grooves

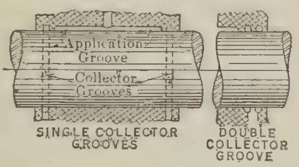


Fig. 19. Collector Grooves

**Collector Grooves** are circumferential grooves at the sides of automatically- or pressure-oiled bearings. They collect side leakage and return it to the oil well to prevent its being wasted by flow along the shaft. Such grooves usually are single, but frequently double. They should have an adequate discharge hole at the bottom. They should be continuously circumferential without a break. Oil application grooves should not enter the collector grooves. See Fig. 19.

**Pumping Grooves** are used both to pump and apply oil. The most common type is the helical groove used with a vertical shaft to lift oil from the well to the upper end of a bearing that may be higher than the surface of oil in the well. The most effective pumping groove is circumferential, if it stops short of extending all the way around the shaft, and is cut in the stationary bearing surface. The inlet end must open to the oil in the well in a manner that will readily receive oil. The other end must be dammed in the circumferential direction. From this point, suitable passages lead oil to any location within the limits of the available pressure. It simply may be led along the shaft through application grooves. Pumping grooves should be completely submerged in oil to avoid aeration of the oil bath. Aeration is, at best, hard to control in high-speed bearings.

The cross-section of a pumping groove should be shallow radially. The shallower the depth the greater will be the pressure developed, but the less the discharge. The wider the axial width the greater will be the rate of discharge. For calculations involved in design, see Viscous Flow, page 13–44. Ordinarily a simple groove of cross-section shown in Fig. 15, but shallower, will be satisfactory.

Many self-contained systems have been devised to lubricate the journal and thrust bearings of a machine continuously, which cause oil to flow to all surfaces needing it, and then return it to the well. Bearings so lubricated are durable and require a minimum of care. See Figs. 20 and 61.

**DIRT EXCLUSION AND OIL RETENTION.**—Dirt exclusion and oil retention involve the shaft, the bearings, and the mountings. Dirt must be excluded from the mountings, as it thickens the oil and closes passages and grooves. Grit in the oil scores shaft and bearing, and may cause overheating and seizing.

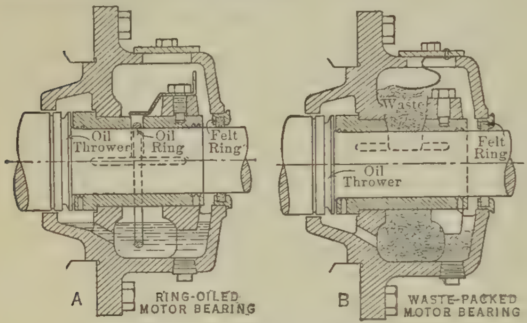


Fig. 20. Motor Bearings

In motor bearings, as Fig. 20, the problem is relatively simple because of high speed of the shaft. During operation, oil flows away in the collector grooves at the right and left and on the face of the bushing back into the well. When rotation ceases, oil drips off the shaft close to the bushing. The oil thrower at the left prevents oil leaking through the housing and the felt ring at the right excludes dirt. The oil ring, or the waste packing, lifts more oil than is needed, most of which runs back between the webs that hold the bushing. Side leakage from the bearing film is mostly taken by the collector groove and a little flows away at the faces of the bushing.

**CLOSURES.**—Conditions to be guarded against by closures include: 1. Creeping of oil along the shaft away from the bearing surfaces. 2. Rising of oil level within bearing mounting, with a tendency to overflow where the shaft passes out. 3. Collection of foam on the surface of oil within the mounting. 4. Continuing presence of contamination outside mounting at shaft opening, which may be dust or other impalpable material, abrasive particles, water or other liquids, or injurious vapors or gases. The type of closure will depend on the speed of the shaft the quantity of contamination, and the tendency of oil to flow from the mounting. High-speed shafts are easier to protect than those operating at low speeds.

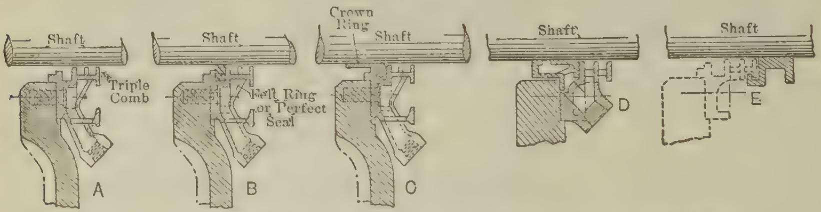


Fig. 21. Closures, Plain, with Felt, and with Crown Ring, for Large Shafts

Various closure designs are shown in Figs. 21 and 22. Those in Fig. 21 are the simplest, and require a minimum of pattern expense for a given shaft diameter. Fig. 21A is a simple comb, made in halves, bolted together, and held against the bearing mounting by cap screws. A small oil pocket is formed at the bottom between the second and third rings of the comb. Both of the outer spaces are drilled at the bottom. This comb closes the space between the mounting bore and the shaft, when the mounting is not bored with a close clearance. The comb is of brass or bronze, with a clearance around the shaft a little greater than the running clearance in the bearing. The shaft would not be harmed by contact with the soft comb. The simple comb will not prevent oil from creeping outward, but will prevent outer contaminations from working inward.

In Fig. 21B, a felt ring is added to the comb, and is fairly effective against dust entering the mounting. It tends to prevent oil leaking outward but is not reliable for this purpose. The felt ring is held in place by several pins, not shown, that pass through it axially, two being close to the joint in each half. The thickness of the felt, from which the ring strip is cut, is the axial width of the ring, so that the felt does not pucker when bent against the shaft.

If oil tends to creep outward along the shaft, the patented crown ring closure, Figs. 21C and 22, is very effective in retaining the oil. This ring is made to fit the recess in the simple comb, Fig. 21A. If a quantity of oil flows along the shaft, there may be some small leakage past the ring, which can be collected and carried to the oil reservoir of the bearing through the drain at the bottom. However, the crown ring runs usually without leakage.

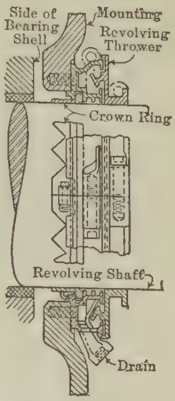


FIG. 22. Dredge Pump Bearing Arranged to Retain Oil and Exclude Water.

If a continuous head of oil is to be held or if the amount to be restrained is so great as to practically constitute a head of oil on the closure, the crown ring must be supplemented by an additional seal ring, as shown in Fig. 21D. The comb ring that retains this double seal crown ring requires longer cap screws to hold it, and it must be centered on the outside of the flange. Some oil always will leak past the larger outer flange of the floating double seal crown ring, and past its sealing band that encircles the shaft at the left of the crown. This oil must be led away from the comb to a sump, which may be in the bearing housing, and whose level is below the lower side of the shaft.

The water thrower, Fig. 21E, made in halves, can be used with any of the foregoing closures, as indicated by the dotted lines adjacent to the ring section. It protects the closure against the entrance of water or other external contaminations.

The closure, Fig. 22, is used for journal bearings having openings adjacent to sources of serious water contamination, as leakage from the packing of a pump shaft. The comb is double flanged, enabling the closure to carry away a large volume of water, with a minimum of leakage into the space between the outer flange and the comb. Any water that may pass is drained at the bottom of the space. The double-flange closure can be made in the four forms corresponding, respectively, in general function with the designs of Fig. 21. In the closures shown in Figs. 21A and B, the oil take-off shown may be omitted if the amount of leakage is small.

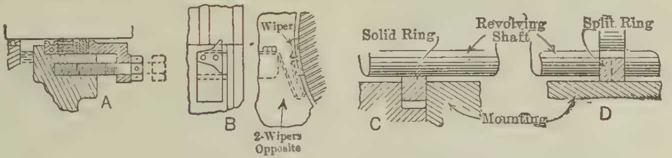


FIG. 23. Seal Rings for Retaining Oil

**PACKING AND SEAL RINGS.**—The common closure with packing, Fig. 23A, usually has two rings of packing. If a considerable head of oil is to be held in a bearing, three or four rings of packing may be used, to avoid setting up the packing very tight. Woven square packing constitutes a nearly rigid form of closure, with insufficient elasticity to permit the shaft to change position naturally within its bearing without permitting leakage past the packing. Such packing tends to dry out and to become inelastic. If not occasionally renewed or set up it will develop an opening that will permit leakage. When packing is set up against a revolving shaft, best practice is to tighten sufficiently to cause the packing to fill the space between housing and shaft, and to then relax the screws and reset them very lightly against the gland ring. This practice, if there is some oil on the shaft, or lubrication in the packing itself, will tend less to cause wear of the shaft and heat it less. Packing set up injudiciously tight, will generate more heat than the bearing itself. Special types of packing are available with the woven portion wrapped in babbitt foil, or with thin strips of babbitt on the inner surface that will press against the shaft. These run cooler than ordinary packing.

**Wipers** sometimes are used to prevent flow of oil along a shaft. Fig. 23B shows one form of wiper. Two of these, opposite, may be used. A single spring wiper may be applied to the lower side of the shaft and fastened to the closure comb. The crown ring, Fig. 22, amounts to a consecutive series of wipers mounted on the edge of a continuous band of metal and set in a groove so that it can float as the shaft moves in its clearance.

**Seal Rings.**—The seal ring has largely displaced packed stuffing boxes. See Fig. 23. In Fig. 23C, running clearance is needed between ring and shaft and sliding clearance between ring and groove. The ring may or may not be keyed against rotation. If made with fixed circumference, provision must be made for expansion due to running temperature. The split ring, Fig. 22D, may be in the form of a piston ring, or in halves butted or held by screws or a spring ring.

**ADJUSTABILITY OF BEARINGS** frequently is desirable in order to align the shaft. Jack screws, or wedges and shims are used. Compensating for wear in the bearing itself is difficult and unsatisfactory. Perfect lubrication to reduce wear to a minimum, and simple renewable bearing parts are a much better solution.

**STANDARDIZED MOUNTINGS.**—The standard tendency in bearing design is to reduce the axial width of the journal. Table 1 gives dimensions of three types of mountings, Figs. 25 to 27, in which elements of a journal bearing, Fig. 24, are incorporated. The effective width of the bearing shell is based on a standard of $1.2D$, where $D$ = journal diameter. Tabulated dimensions will deviate from this standard because each nominal size of mounting covers two standard shaft diameters. The mounting dimensions given in Table 1 are suitable for standard-duty bearing shells of the pressure type and ring oiling type. The capacity of these bearings, with a maximum load of $400A^2$ lb. is given in Table 2. $A$ here is the diameter of the journal in inches.

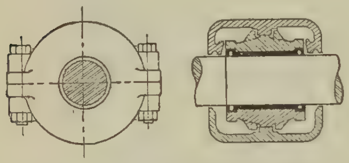


Fig. 24. End View and Axial Section of Barrel Portion of Mounting for Self-aligning Journal Bearings

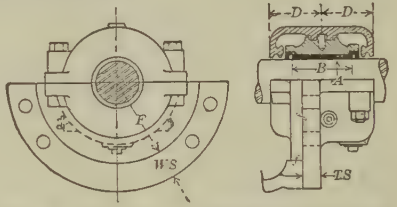


Fig. 25. End View and Part Axial Section of Semi-circular Flanged Mounting

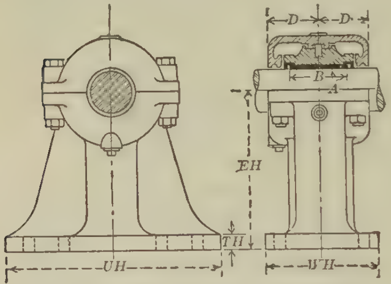


Fig. 26. End View and Part Axial Section of High Pedestal Mounting

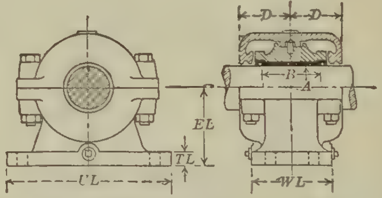
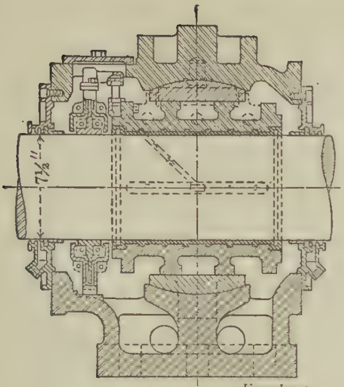


Fig. 27. End View and Part Axial Section of Low Pedestal Mounting

**Table 1.—Dimensions of Typical Bearing Mountings**

(Kingsbury Machine Works, Inc., Philadelphia)

| Dimension | Shaft Diameter, in. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $A_1$ | 2.500 | 3.000 | 3.500 | 4.000 | 4.500 | 5.250 | 6.000 | 7.000 | 8.000 | 9.000 |
| $A_2$ | 2.750 | 3.250 | 3.750 | 4.250 | 4.875 | 5.625 | 6.500 | 7.500 | 8.500 | 9.500 |
| | Journal Width, in. | | | | | | | | | |
| $B$ | 3 1/4 | 4 | 4 1/2 | 5 | 5 3/4 | 6 3/4 | 7 3/4 | 9 | 10 1/4 | 11 1/2 |
| | Dimensions, in. See Figs. 25 to 27. | | | | | | | | | |
| $D$ | 3 1/16 | 3 1/2 | 4 1/16 | 4 1/2 | 5 1/8 | 5 7/8 | 6 9/16 | 7 5/16 | 8 1/4 | 9 3/16 |
| $EH$ | 10 3/4 | 11 3/4 | 13 1/2 | 15 | 16 3/4 | 19 | 2 1/12 | 23 3/4 | 28 1/4 | 32 1/2 |
| $EL$ | 4 3/4 | 5 3/8 | 6 3/8 | 7 1/8 | 7 7/8 | 8 3/4 | 9 7/8 | 10 3/4 | 12 5/8 | 14 |
| $F$ | 9.000 | 10.750 | 12.500 | 14.250 | 16.000 | 19.000 | 21.250 | 24.000 | 28.250 | 31.625 |
| $TH$ | 1 | 1 | 1 1/8 | 1 1/8 | 1 1/4 | 1 1/2 | 1 3/4 | 1 3/4 | 2 | 2 1/4 |
| $TL$ | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/8 | 1 1/4 | 1 1/4 | 1 1/2 | 1 5/8 | 1 7/8 |
| $TS$ | 1 | 1 1/8 | 1 1/4 | 1 3/8 | 1 1/2 | 1 1/2 | 1 7/8 | 1 7/8 | 2 1/4 | 2 1/4 |
| $UH$ | 13 1/4 | 15 1/2 | 18 | 20 1/2 | 23 | 26 1/2 | 30 | 33 1/2 | 39 1/2 | 44 1/2 |
| $UL$ | 9 1/2 | 10 3/4 | 12 1/2 | 14 1/4 | 15 1/4 | 18 1/4 | 19 3/4 | 22 1/4 | 25 1/2 | 28 5/8 |
| $WH$ | 6 1/8 | 7 | 8 1/8 | 9 | 10 1/4 | 11 3/4 | 13 1/8 | 14 5/8 | 16 1/2 | 18 3/8 |
| $WL$ | 5 | 6 | 6 1/2 | 7 1/8 | 8 1/4 | 10 | 10 1/4 | 12 | 13 3/8 | 15 1/4 |
| $WS$ | 1 5/8 | 2 | 2 3/8 | 2 5/8 | 2 7/8 | 3 1/4 | 3 7/8 | 4 1/2 | 5 | 5 5/8 |



*Kingsbury*

FIG. 28. Heavy Duty Journal Bearing with Closures and Disc Oiling

**HEAVY DUTY JOURNAL BEARINGS** are bearings designed to carry loads up to about 1200 lb. per sq. in. on an area equal to the square of the journal diameter in inches. The design and minimum proportions of such a bearing of the low pedestal type are indicated in Fig. 28. The shell stiffness should match that of the shaft to compensate for deflection under load. The journal may present its load to the bearing shell either with convex or concave flexure. For extra heavy duty service, pressures range from 1000 to 3000 lb. per sq. in. See also Roll Neck Bearings, p. 13–48.

**JOURNAL BEARING CAPACITIES.**—The factors governing carrying capacity of journal bearings are more fully discussed on pp. 13–15 to 13–30.

Table 2 will be found useful for a wide range of journal bearing applications of the standard-duty, self-aligning type (see Figs. 25, 26, 27), whose loads usually range from 50 to 400 lb. per sq. in. on an area equal to the square of the journal diameter in inches. The

**Table 2.—Rated Load Capacities of Journal Bearings ***

Journal length = Journal diameter. For other lengths multiply rated load by factors below.

| Journal Diam., in. | Max. Load, lb. | Revolutions per Minute | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 50 | 100 | 200 | 400 | 800 | 1200 | 1800 | 2400 | 3600 |
| | | Rated Load, lb. | | | | | | | | |
| 2 | 1,600 | 100 | 145 | 205 | 285 | 405 | 495 | 610 | 700 | 860 |
| 2 1/4 | 2,000 | 135 | 190 | 270 | 380 | 535 | 655 | 800 | 930 | 1,140 |
| 2 1/2 | 2,500 | 170 | 240 | 340 | 485 | 680 | 840 | 1,030 | 1,190 | 1,450 |
| 2 3/4 | 3,000 | 205 | 305 | 425 | 600 | 850 | 1,050 | 1,280 | 1,480 | 1,810 |
| 3 | 3,600 | 260 | 370 | 520 | 740 | 1,040 | 1,280 | 1,570 | 1,810 | 2,220 |
| 3 1/4 | 4,200 | 315 | 445 | 630 | 890 | 1,260 | 1,540 | 1,890 | 2,180 | 2,670 |
| 3 1/2 | 4,900 | 375 | 530 | 750 | 1,060 | 1,500 | 1,840 | 2,250 | 2,600 | 3,180 |
| 3 3/4 | 5,600 | 440 | 625 | 840 | 1,250 | 1,760 | 2,060 | 2,640 | 3,050 | 3,740 |
| 4 | 6,400 | 510 | 725 | 1,025 | 1,450 | 2,050 | 2,500 | 3,070 | 3,550 | 4,340 |
| 4 1/4 | 7,200 | 590 | 830 | 1,180 | 1,670 | 2,360 | 2,780 | 3,540 | 4,080 | 5,000 |
| 4 1/2 | 8,100 | 675 | 950 | 1,350 | 1,910 | 2,690 | 3,300 | 4,050 | 4,670 | 5,700 |
| 4 7/8 | 9,500 | 810 | 1,150 | 1,625 | 2,300 | 3,250 | 3,980 | 4,880 | 5,600 | 6,900 |
| 5 1/4 | 11,000 | 970 | 1,370 | 1,930 | 2,730 | 3,870 | 4,730 | 5,800 | 6,700 | 8,200 |
| 5 5/8 | 12,700 | 1,140 | 1,610 | 2,270 | 3,220 | 4,550 | 5,600 | 6,800 | 7,900 | 9,600 |
| 6 | 14,400 | 1,320 | 1,860 | 2,630 | 3,730 | 5,300 | 6,500 | 7,900 | 9,100 | ........ |
| 6 1/2 | 16,900 | 1,590 | 2,250 | 3,180 | 4,500 | 6,400 | 7,800 | 9,500 | 11,000 | ........ |
| 7 | 19,600 | 1,890 | 2,670 | 3,770 | 5,300 | 7,600 | 9,200 | 11,300 | 13,100 | ........ |
| 7 1/2 | 22,500 | 2,220 | 3,140 | 4,430 | 6,300 | 8,900 | 10,900 | 13,300 | 15,400 | ........ |
| 8 | 25,500 | 2,580 | 3,650 | 5,200 | 7,300 | 10,300 | 12,600 | 15,500 | 17,900 | ........ |
| 8 1/2 | 28,900 | 2,970 | 4,200 | 5,900 | 8,400 | 11,900 | 14,500 | 17,800 | 20,600 | ........ |
| 9 | 32,400 | 3,400 | 4,800 | 6,800 | 9,600 | 13,600 | 16,600 | 20,400 | ........ | ........ |
| 9 5/8 | 37,100 | 3,980 | 5,600 | 7,900 | 11,200 | 15,900 | 19,500 | 23,900 | ........ | ........ |
| 10 1/4 | 42,200 | 4,600 | 6,500 | 9,200 | 13,000 | 18,400 | 22,500 | 27,600 | ........ | ........ |
| 10 7/8 | 47,500 | 5,300 | 7,400 | 10,500 | 14,900 | 21,100 | 25,800 | 31,600 | ........ | ........ |
| 11 1/2 | 53,000 | 6,000 | 8,500 | 12,000 | 17,000 | 24,000 | 29,400 | 36,100 | ........ | ........ |
| 12 1/8 | 58,900 | 6,800 | 9,600 | 13,600 | 19,300 | 27,200 | 33,300 | ........ | ........ | ........ |
| 12 7/8 | 66,500 | 7,900 | 11,100 | 15,700 | 22,200 | 31,300 | 38,300 | ........ | ........ | ........ |
| 13 5/8 | 74,100 | 9,000 | 12,600 | 17,900 | 25,300 | 35,800 | 43,800 | ........ | ........ | ........ |
| 14 3/8 | 82,800 | 10,100 | 14,300 | 20,200 | 28,600 | 40,500 | 49,500 | ........ | ........ | ........ |
| 15 1/8 | 91,600 | 11,600 | 16,300 | 23,100 | 32,700 | 46,200 | 57,000 | | | |
| 15 7/8 | 101,000 | 12,800 | 18,100 | 25,600 | 36,100 | 51,000 | 63,000 | | | |
| 16 3/4 | 112,300 | 14,500 | 20,500 | 28,900 | 40,900 | 58,000 | 71,000 | | | |
| 17 5/8 | 124,500 | 16,300 | 23,100 | 32,600 | 46,200 | 65,000 | ........ | | | |
| 18 1/2 | 137,000 | 18,200 | 25,800 | 36,500 | 51,700 | 73,000 | ........ | | | |
| 19 3/8 | 150,500 | 20,400 | 28,800 | 40,700 | 57,500 | 81,000 | ........ | | | |
| 20 1/4 | 164,000 | 22,600 | 31,900 | 45,100 | 64,000 | 90,000 | ........ | | | |
| 21 1/4 | 181,000 | 25,300 | 35,700 | 50,500 | 71,000 | 101,000 | ........ | | | |
| 22 1/4 | 198,000 | 28,100 | 39,800 | 56,000 | 79,000 | 112,000 | ........ | | | |
| 23 1/4 | 216,000 | 31,200 | 44,100 | 62,500 | 88,000 | 125,000 | ........ | | | |

Rating Factors for Various Ratios of Journal length ÷ Journal Diam:

| $L/D$ | Rating Factor |
|---|---|
| 0.5 | 0.21 |
| 0.8 | 0.63 |
| 1 | 1.00 |
| 1.2 | 1.42 |
| 1.5 | 2.07 |

* With suitable lubricant.

loads are based on a mean viscosity of oil in the loaded side of the film of $3.4 \times 10^{-6}$ Reyn (0.235 poise), or 125 Saybolt seconds for an oil of average base. The bearings will carry heavier loads than the standard ratings. See Column 2 for load limit for mechanical strength of the mounting, with load acting toward the base. For loads above rated load, up to the maximum load values, mean film viscosity should be increased above $3.4 \times 10^{-6}$ Reyn in proportion to the increase of load for the given speed. The relation between supply temperature of oil and mean film temperature depends on the circulation rate and unit loading of the bearing. If the bearing is lubricated by a stream of oil circulated at a rate of 0.6 gal. per Hp. of friction loss in the bearing, the oil supply temperature should be approximately $(10 + W/20)$ deg. F. above the mean temperature desired in the loaded area of the bearing film, $W$ being the unit load, lb. per sq. in., on the bearing.

**VERTICAL JOURNAL BEARINGS.**—These are known also as **Guide Bearings, and Steady Bearings.** Except for the difficulty of providing automatic self-alignment and automatic self-lubrication the problem is the same as for horizontal journal bearings. In Fig. 29, the weight of the shell causes it to lie against the top of its support. If alignment is incorrect, and the pressure of the shaft is applied to the top, the shell will tip freely into proper alignment. If, however, the shaft presses first at the bottom, the shell is apt to cramp. Solutions of this difficulty are (*a*) the supporting of the bearing shell a little above the shoulder by an elastic suspension, (*b*) the use of a spherical seat, (*c*) the use of pivoted shoes (Figs. 9 and 90). Self-aligning vertical bearings seldom are needed because vertical shafts are not deflected by the weights carried by them, as are shafts in horizontal bearings. In vertical shafts the weights produce axial loads, which are carried by thrust bearings. Because radial loads are light, and for simplicity, vertical bearings usually are of the fixed type. Axial width (height) should be as small as practicable.

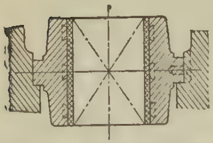


FIG. 29. Vertical Journal Bearing

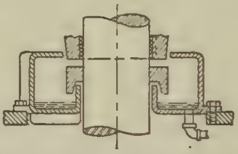


FIG. 30. Collar and Pan for Collecting Oil below a Vertical Journal Bearing.

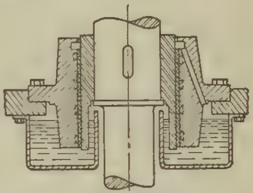


FIG. 31. Lubricating System for Vertical Journal Bearing, Spiral Groove not shown.

**Lubrication of Vertical Journal Bearings.**—With pressure lubrication, the simplest bearing form may be used. The journal surface may be on the shaft itself. To collect the oil flowing down the shaft below the bearing, a collar on the shaft throws the oil into a stationary pan, whence it flows to the sump of the oiling system. See Fig. 30.

Vertical journals may be on the surface of sleeves shrunk or otherwise fastened on the shaft. See Fig. 31. The lower end of the sleeve, which is bored out larger than the shaft projects into the stationary oil pan. Oil may be taken into the bearing film by capillarity, by viscosity pumping, centrifugally, or by combinations of these methods.

If the shaft terminates at the bottom in a vertical journal bearing, lubrication is a relatively simple problem. With a through shaft, simple constructions also may be used, if lubrication is by cup, and if some waste of oil or grease is unimportant. Leakage along the shaft may be minimized by a stuffing-box if the speed of rotation is not so high as to cause heating of the shaft by the packing.

## 2. SHAFT DEFLECTIONS

**DEFLECTION FORMULAS.**—Shaft deflections within journal bearings may be studied by the application of formulas for cantilever beams. The deflection may be considerable, especially when compared to the thickness of the oil film. The basic equation is the general one of the elastic curve

$$(d^2y/dx^2) = M/EI \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

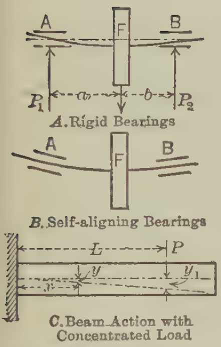
where $y$ = deflection at any point at a distance $x$ from the support; $M$ = bending moment at the cross-section under consideration; $E$ = modulus of elasticity of the material; $I$ = moment of inertia of the cross-section, referred to its neutral axis.

Figs. 32 and 33 are two typical arrangements of shaft and bearings, exaggerated to show the relation of the deflected shaft to both rigid and self-aligning bearings. Bearing load may be assumed as concentrated for the purpose of determining shaft deflections and the general direction of its center line in the bearing. To study oil film shape in a self-aligning bearing, or where shaft direction is so little out of line as not to require self-aligning bearings, bearing reaction may be assumed as uniformly distributed. From equation [1], equations [2] to [6] can be derived for the case shown in Fig. 32 which represents a concentrated load. In these equations, $P$ = concentrated load on bearing, and $L$ = distance of point of application of load from point of support, *i.e.*, from the center line of the bearings A and B, other notation as before. The diagrams show the significance of $y_1$.

$$y = (P/EI)\{(Lx^2/2) - (x^3/6)\} \qquad [2]$$
$$dy/dx = (P/EI)\{Lx - (x^2/2)\} \qquad [3]$$
$$y_1 = (L^3/3)(P/EI) \qquad [4]$$
$$dy_1/dx = (L^2/2)(P/EI) \qquad [5]$$
$$y_1 = (2L/3)(dy_1/dx) \qquad [6]$$

For the case shown in Fig. 33*c*, representing a uniformly distributed load, the following equations may be derived, in which $W$ = total uniform load, $\Delta$ = deflection within the bearing, other notation being as before:

$$y = (W/24LEI)\{(L - x)^4 + 4L^3x - L^4\} \qquad [7]$$
$$dy/dx = (W/6LEI)\{L^3 - (L - x)^3\} \qquad [8]$$
$$y_1 = (L^3/8)(W/EI) \qquad [9]$$
$$dy_1/dx = (L^2/6)(W/EI) \qquad [10]$$
$$y_1 = (3L/4)(dy_1/dx) \qquad [11]$$
$$\Delta = (y_1/2) - y \qquad [12]$$



A
F
B
$P_1$
a
b
$P_2$
A. Rigid Bearings
A
F
B
B. Self-aligning Bearings
L
P
y
$y_1$
x
C. Beam Action with Concentrated Load

FIG. 32. Shaft with Load between Bearings

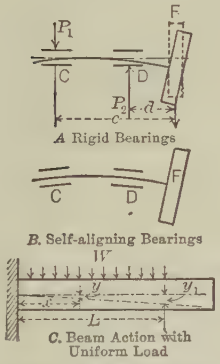


FIG. 33. Shaft with Overhung Load

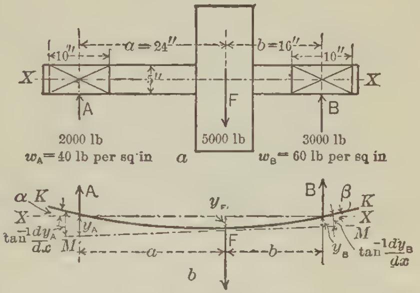


FIG. 34. Data for Solution of Example 1

EXAMPLE 1.—Fig. 34*a* shows a 5-in. shaft, supported in two bearings each 10 in. long, spaced 40 in. center to center. The point of application of load is 24 in. from bearing A and 16 in. from bearing B. It is required to determine the deflection of the shaft, and the angle of the deflected shaft at the bearings with the center line $XX$ through the bearings.

*Solution.*—Reaction of bearing A = 5000 × (16/40) = 2000 lb.; reaction of bearing B = (24/40) × 5000 = 3000 lb. Taking the modulus of elasticity of steel as 30,000,000, and the moment of inertia as $\pi d^4/64 = 30.68$ for a 5-in. shaft, and applying equation [4], the deflections with reference to bearings A and B are:

$$y_A = (24^3/3) \times \{2000/(30{,}000{,}000 \times 30.68)\} = 0.01002$$
$$y_B = (16^3/3) \times \{3000/(30{,}000{,}000 \times 30.68)\} = 0.00446$$

These relations as shown in Fig. 34*b*, in which $XX$ is the center-line of the undeflected shaft, $KK$ is the center-line of the deflected shaft, and $y_A$ and $y_B$, are respectively, the deflections of the shaft at bearings A and B. A line $MM$ drawn through the extremities of $y_A$ and $y_B$ will be tangent to the center-line of the deflected shaft at the point of application of the load F. Its angle with reference to $XX$ will be $\tan^{-1}\{(y_A - y_B)/AB\} = \tan^{-1}\{(0.01002 - 0.00446)/40\} = \tan^{-1} 0.00014$. From equation [5] the direction of the deflected shaft at the bearings is

At bearing A $\quad dy_1/dx_1 = (24^2/2)(2000/30{,}000{,}000 \times 30.68) = 0.000626$

At bearing B $\quad dy_1/dx_1 = (16^2/2)(3000/30{,}000{,}000 \times 30.68) = 0.000417$

These values are the tangents of the angles between the center-line of the deflected shaft at the bearings and the line $MM$, tangent to the shaft at the point of application of load F. In these problems, the angles are so small that the tangents of them can be added and subtracted instead of the angles themselves, without affecting the result. The angles $\alpha$ and $\beta$, respectively, the angles between the center-lines of deflected and undeflected shaft are

$$\tan \alpha = \{(dy_1/dx) - \tan (y_A - y_B/\overline{AB})\} = 0.000626 - 0.00014 = 0.000486$$

$$\tan \beta = \{(dy_1/dx) + \tan (y_A - y_B)/\overline{AB}\} = 0.000417 + 0.00014 = 0.000557$$

Misalignment of the bearings is the product of tan $\alpha$ or tan $\beta$ and the bearing length. Hence, misalignment of bearing A = 10 × 0.000486 = 0.00486 in., and of bearing B = 10 × 0.000557 = 0.00557 in. Since the oil film thickness in the direction of the load in a rigid bearing would be in the neighborhood of 0.002 in., a self-aligning bearing is indicated.

The deflection of the shaft within the bearing length, however, also must be investigated. This may be taken as a case of a uniformly-loaded cantilever beam (see Fig. 35), although the distribution of oil pressure across the film is approximately parabolic for a film of uniform thickness along the axis. Uniformly distributed pressure will show a higher shaft deflection than actual, and thus is on the safe side. Referring to Fig. 35, deflection $y_1$ is found by equation [9], and deflection $y_2$ by equation [7], in which $x$ is taken equal to $L/2$.

In bearing A,

$$y_1 = (10{,}000/8) \times (2000/30{,}000{,}000 \times 30.68) = 0.0002716$$

$$y_2 = (2000/24 \times 30{,}000{,}000 \times 30.68) \times \{(10 - 5)^4 + (4 \times 10^3 \times 5) - 10^4\} = 0.0000961$$

From equation [12] the deflection within the bearing is

$$\Delta = (y_1/2) - y_2 = (0.0002716/2) - 0.0000961 = 0.0000397.$$

By a similar calculation for bearing B, the deflection $\Delta$ = 0.00006.

It should be noted that the deflections above found, although negligible, are proportional to the unit pressures for a given journal. The unit pressures in Example 1, *i.e.*, 40 and 60 lb. per sq. in. are moderate, and much higher pressures may occur with some types of machinery. If heavier loads are applied without suitable compensations in bearing design, the deflections in the shafts and journals may become dangerous.

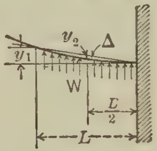


FIG. 35.

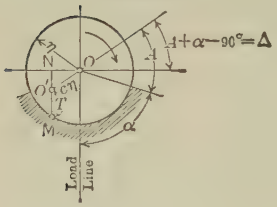


FIG. 36.

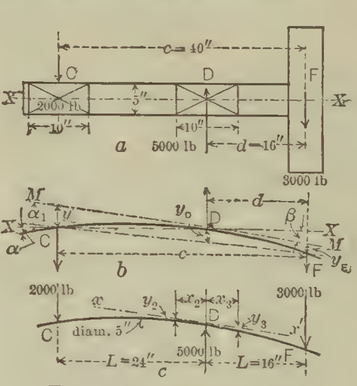


FIG. 37. Data for Example 2

EXAMPLE 2.—Determine the deflections of the shaft of the dimensions and arrangement shown in Fig. 37.

*Solution.*—Draw the axis $MM$ (Fig. 37*b*) tangent to the center line of the deflected shaft at $D$. Compute the deflections $y_C$ = 0.01002 and $y_F$ = 0.00446 by equation [4]. These will be found to be the same as $y_A$ and $y_B$ in Example 1. Determine the angle the deflected shaft, at the bearings, makes with the center-line of the bearings. From equation [5]

$$\tan \alpha_1 = (dy_1/dx) = (24^2/2)\{2000/(30{,}000{,}000 \times 30.68)\} = 0.000626,$$

and $$\tan \beta = y_C/24 = 0.000418.$$

But $$\alpha = \alpha_1 - \beta \therefore \tan \alpha = \tan \alpha_1 - \tan \beta = 0.000626 - 0.000418 = 0.000208.$$

These deflections are too large for the use of fixed bearings, unless the oil films are abnormally thick.

Bearing D, however, must be examined by means of formula [2], and deflections $y_2$ and $y_3$ (Fig. 37*c*) determined. $x_2 = x_3 \doteq 5$; $L_2 = 24$; $L_3 = 16$; $P_2 = 2000$; $P_3 = 3000$.

$$y_2 = (P/EI)\{(Lx_2^2/2) - (x_2^3/6)\} = \{2000/(30 \times 10^6 \times 30.68)\}[\{(24 \times 5^2)/2\} - (5^3/6)] = 0.000607$$

$$y_3 = \{3000/(30 \times 10^6 \times 30.68)\}[\{(16 \times 5^2)/2\} - (5^3/6)] = 0.000584.$$

The unit bearing pressure is $\{5000/(10 \times 5)\}$ = 100 lb. per sq. in., which is permissible, but the deflections within the bearing are sufficient, even with self-alignment, to require a heavy oil for producing a thick enough film.

For a more complete discussion of the methods of determining thickness of oil films, see pp. 13-17 to 13-30.

In calculating shaft deflection in a bearing, the position of the center of the shaft, when running, is important. It shifts to the point $O'$, Fig. 36 within the clearance circle of radius $\eta$. The plane of shaft deflection cuts this circle at $M$, which is the point at which the deflected shaft will make contact with the bearing, assuming that the latter does not deflect. Referring to Fig. 36, the following relations hold: Let $(A + \alpha - 90°) = \Delta$. Then $\overline{ON} = c\,\eta \cos \Delta$; $MN = \sqrt{\eta^2 - \overline{ON}^2}$; $TO' = MN - c\,\eta \sin \Delta$. The deflection should be compared with distance $O'M$, and should not be an unreasonable percentage, say 30%, of $O'M$. The shaft usually should be examined both for strength and stiffness.

## 3. LOADING AND FRICTION OF JOURNAL BEARNGS

**JOURNAL BEARINGS**, for the purpose of study of loading and friction, may be divided broadly into Clearance Bearings and Fitted Bearings. Clearance bearings may be either **full** or **partial bearings.** Fitted bearings invariably are partial bearings. The angular extent to which the useful bearing surface surrounds the journal is designated by $\beta$, and determines whether a bearing is full or partial. In a full bearing $\beta$ = 360 deg.; in a partial bearing $\beta$ usually is less than 180 deg., and averages about 120 deg., except that for fitted bearings it averages about 90 deg.

**Clearance Bearings** are those in which the radius of curvature of the bearing is larger than that of the journal.

**Fitted Bearings** are those in which radius of curvature is the same in both bearing and journal. Values of $\beta$ are essentially less than 180 deg., averaging about 90 deg. Characteristics are given below for such bearings for values of $\beta$ up to 150 deg.

**LOAD DIRECTION.**—Partial bearings may be centrally loaded or eccentrically loaded. There may be different degrees of the latter. The angular extent of bearing surface ahead of the point of load is known as the leading angle $\alpha$. See Figs. 39, 42 and 47.

**Centrally-loaded Bearings** have $\beta$ divided into two equal parts by the line of action of the applied load. That is $\alpha = \beta/2$.

**Eccentrically-loaded Bearings** have $\beta$ divided into two unequal parts by the line of action of the applied load. The ratio $\alpha/\beta$ may vary considerably when bearings are in service, by reason of lack of analysis in design, but some ratios of $\alpha/\beta$ are better than others.

**JOURNAL BEARING CHARACTERISTICS.**—Journal bearings may be divided into six classes as follows, for the purpose of co-ordinating the available analyses: A. Eccentrically loaded partial bearings with running clearance. See Fig. 39. B. Fitted partial bearings, Fig. 43. C. Centrally loaded partial bearings with running clearance, Fig. 47. D. Offset partial bearings with running clearance, Fig. 39. E. Fitted partial bearings, Fig. 43, with variable ratio of $\alpha/\beta$. F. Full bearing with running clearance, Fig. 55. The characteristics of these several types of bearings are discussed below. Classes B, D and E all are eccentrically loaded and different in some respects. For Classes A, B and C, Kingsbury's optimum conditions are given in the charts. For Classes C, D, E, and F, Howarth's charts for general conditions are given. Classes A and D are similar, but not quite alike. The same is true of Classes B and E. In chart Fig. 56, general conditions of loading are given for a wide variety of 120-deg. clearance bearings including those of Classes A, C, and D.

**Notation.**—In the discussion which follows, the significance of the symbols used is given below. Unless otherwise specified in the text, all linear dimensions are in inches, and forces (loads) are in pounds.

$a$ = radius of journal.
$A$ = least angle from line of centers to leading edge of partial bearing, measured in direction of rotation. See exception in study of fitted bearings, Fig. 43, p. 13–20.
$b$ = width of bearing surface measured along the shaft axis in a journal bearing, or, in general, at right angles to direction of motion.
$c$ = eccentricity factor. When multiplied by the radial clearance $\eta$, the product $c\eta$ equals distance from center of bearing to center of journal.
$F$ = total tangential friction force at surface of a journal, or at mean diameter of a thrust bearing, per unit of width of bearing surface.
$F_1$ = correction factor for friction, *with side leakage.*
$h$ = film thickness at any angular position $\theta$ in journal bearings, or any distance $x$ for flat tapered film. For fitted journal bearings $h = \epsilon \sin \theta$, and for clearance journal bearings $h = \eta(1 + c \cos \theta)$.
$h_o$ = minimum thickness of oil film in a bearing, usually. When the line of centers for a clearance bearing does not intersect the partial bearing surface, $h_o$ is measured where the bearing surface, if extended, would cross that line.
$h_1$ = thickness of oil film at region of maximum pressure.
$h_A$ = thickness of oil film at entering edge of bearing film, if less than $h_m$.
$h_E$ = film thickness at trailing edge of partial clearance bearing, when greater or less than $h_o$.

$h_m$ = maximum thickness of oil film.

$H$ = horsepower friction developed by a journal bearing, or thrust bearing, per unit of bearing width.

Hp. = horsepower.

$l$ = length of a bearing face in the direction of motion. In a partial journal bearing, whose angular extent $\beta$ is expressed in degrees, $l = \pi\beta a/180$. This is useful for determining length-width ratio, $l/b$, used in selecting side leakage correction factors.

$L_1$ = correction factor for load carrying capacity, *with side leakage.*

$N$ = revolutions per minute.

$n$ = number of pivoted segments in a thrust bearing.

$O$ = center of curvature of journal when running on oil film.

$O'$ = center of curvature of bearing surface supporting journal.

$p$ = pressure within the oil film where the thickness is $h$.

$p_o$ = nominal mean pressure acting on bearing. In journal bearings, $p_o$ is found by dividing total applied load by (journal diameter × axial bearing width). It also is expressed by $W/2a$. $W$ is taken per unit of width. In thrust bearings, $p_o = R/n\,l\,b$. See Fig. 82.

$P_m$ or $P_{\text{max.}}$ = maximum pressure within oil film of a bearing.

$R$ = total load applied by a journal to a bearing, or by a collar to a thrust bearing.

$R'$ = total reaction of bearing against the journal.

$s$ = a film form dimension in Fig. 91, which see.

$U$ = relative velocity of the bearing surfaces (usually in. per sec.). In thrust bearings it is measured on the mean diameter.

$V$ = relative velocity of bearing surfaces, ft. per min.

$w$ = nominal mean unit load a bearing would carry if side leakage could be neglected. See $p_o$. The relation is $p_o = L_1 w$.

$W$ = actual load applied by a journal to its bearing, per unit of axial width = $R/b$.

$Z$ = character expressing viscosity coefficient in centipoises. See $\mu$.

$\alpha$ = leading angle for partial bearing, measured in direction of motion of journal from leading edge of bearing to line of action of load.

$\beta$ = angular extent of bearing surface over which perfect film lubrication is maintained. Usually expressed in degrees.

$\epsilon$ = eccentricity of journal axis when running in a fitted bearing.

$\eta$ = radial clearance provided in a clearance bearing. When $a$ = journal radius, $(a + \eta)$ = radius of curvature of a bearing surface, and $c\eta$ = eccentricity of running position of journal axis in a clearance bearing.

$\theta$ = angle from reference diameter to point within oil film where thickness is $h$.

$\theta_1$ = angle $\theta$ corresponding with location of maximum unit pressure within the oil film.

$\lambda$ = coefficient of friction for journal surface. In thrust bearings, $\lambda$ is the coefficient for mean diameter of thrust collar face. $\lambda = (F_1/L_1)\lambda_H$.

$\lambda'$ = coefficient of friction for bearing surface.

$\lambda_H$ = uncorrected coefficient of journal friction in charts.

$\mu$ = coefficient of viscosity. When not otherwise specified, $\mu$ is expressed in lb.-sec. per sq. in. (Reyns). The term *Reyn* is applied to the coefficient when expressed in English units. The c.g.s. unit for $\mu$ is the *Poise* measured in dyne-seconds per sq. cm. One poise = 0.000 0145 Reyn; 1 Reyn = 69,000 poises.

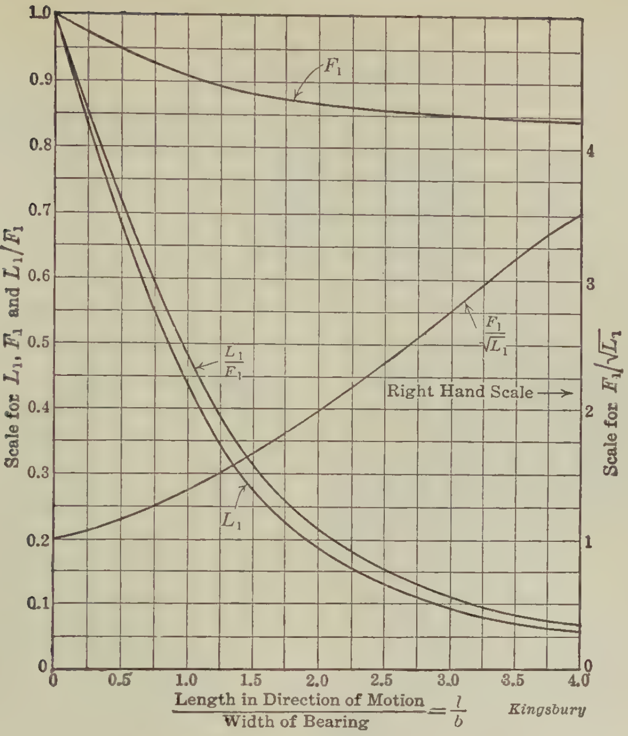
$\phi$ = angle relating load line to line of centers of journal and bearing in Kingsbury's optimum condition (see p. 13–29). See also bearing cross-sections, Figs. 39, 43 and 47.

$\phi_H$ = angle relating load line to line of centers of journal and bearing in bearing cross-sections in Figs. 51 to 56.

**CORRECTION FOR SIDE LEAKAGE.**—Side leakage has an important influence on load capacity and friction of journal bearings. If this influence be neglected in the solution of bearing problems, the data of the charts, Figs. 40 to 56, will not agree well with actual conditions, and side leakage factors, therefore, must be included in the bearing problem data. Fig. 38 gives factors applying to optimum film proportions, but which are fairly dependable when applied to conditions not too far different from optimum. These are the best generally applicable data available (1937) for side leakage corrections. When applied to abnormal conditions, they tend to bring results toward correct values, and usually are on the safe side.

Fig. 38 is used as follows: In the dynamical tables and curves following which apply to the various classes of bearings, as enumerated above, numerical coefficients are for infinitely wide conditions, under which side leakage is zero. For bearings of finite width and a given ratio of (length/width) the numerical coefficients are to be multiplied by a correction factor from Fig. 38. The proper correction curve in Fig. 38 to use for the various symbols or symbol groups of the dynamical tables and curves is as follows:

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Symbol Group................ | $W$ | $F$ | $\lambda$ or $F/W$ | $H$ | $H/W$ | $W/H$ | $\lambda/\sqrt{\mu N/p_o}$ |
| Correction Factor............ | $L_1$ | $F_1$ | $F_1/L_1$ | $F_1$ | $F_1/L_1$ | $L_1/F_1$ | $F_1/\sqrt{L_1}$ |



*Kingsbury*

FIG. 38. Correction Factors for Influence of Side Leakage upon Load-carrying Capacity and Friction

**CHARTS FOR DETERMINATION OF BEARING PROPORTIONS.**—The charts, Figs. 42, 46 and 50 give data useful in the design of optimum bearings of classes A, B, and C respectively. They show the relation of proportion, dimensions, power loss, film thickness, speed and load for a variety of conditions with each type. Total load $R$ on the bearing, and speed, usually are the known factors, and the minimum thickness of the oil film $h_o$ may be assumed. (See examples below.) The permissible loading of the bearing $p_o$ also may be assumed. The projected area of the bearing then is $R/p_o$; it also is $2a^2\ (b/a)$. The ratio of width to radius of journal *i.e.*, $b/a$, when decided, may be substituted in $2a^2\ (b/a)$. By equating the two expressions for projected area, the value of $a$ can be determined. Then from $R$ and $b$ the value of $W$ is obtained.

In Figs. 42, 46 and 50, the power loss for any ratio of $b/a$ at any angular contact $\beta$ can be found by reading on the appropriate curve. The minimum power loss is at the point of inflection of the curve, but does not include any losses in clearance spaces outside of the arc of angular contact $\beta$.

**A. Eccentrically-loaded Partial Bearings with Running Clearance** (Fig. 39).—The radius of curvature of the bearing is greater than that of the journal. The line of application of the load does not pass through the center of the bearing arc, and $\alpha > \beta/2$. The optimum condition for best ratio of $\alpha/\beta$ for values of $\beta$ from 0 to 360 deg. are given in Figs. 40–42 and Tables 3 and 4. Their use is best shown by an example.

EXAMPLE 1.—Determine the proportions of a bearing of minimum friction to support a load of 25,000 lb. at 1500 r.p.m. with $h_o = 0.003$ in., with a pressure of 200 lb. per sq. in. of projected area.

*Solution.*—Projected area = 25,000/200 = 125 sq. in. = $2a^2(b/a)$. Assume $b/a = 3$, whence $a = \sqrt{(125/6)} = 4.564$ in. This value should be checked for strength in bending and in torsion.

From the curve $b/a = 3$ in Fig. 42, the minimum power loss $H$ per unit of width $b$ occurs with a value of $\beta = 153.6$ deg. = 2.68 radians, corresponding to a value of $WNh_o/H = 16{,}900$, whence

$$\text{Total power loss } (H \times b) = 25{,}000 \times 1500 \times 0.003/16{,}900 = 6.66 \text{ Hp.}$$

This figure does not include losses in any clearance spaces outside of the arc of contact $\beta$.

(*Continued on p.* 13–24)

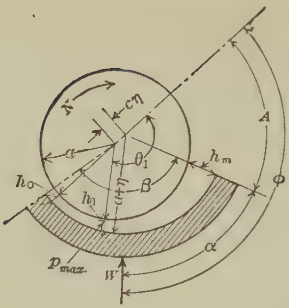


FIG. 39. Eccentrically-loaded Clearance Bearing, $\alpha > \beta/2$, Class A.

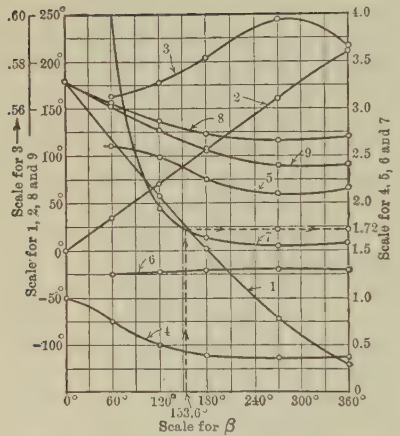


FIG. 40. Geometrical Optima for Minimum Friction Coefficient. See Table 4

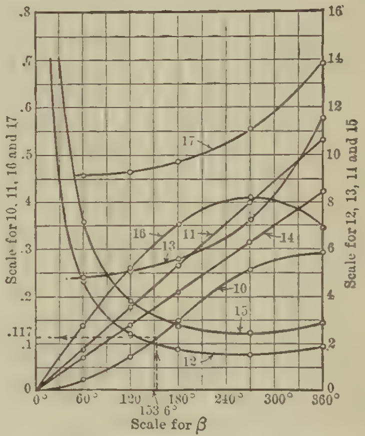
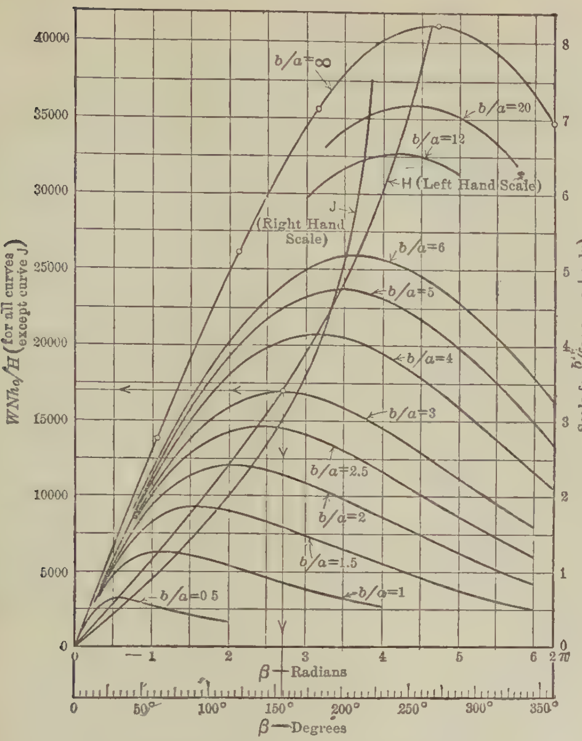


FIG. 41. Dynamical Optima for Minimum Friction Coefficient. See Table 4

**Table 3.—Optimum Values for Maximum Load $W$ of Class A Journal Bearings** (Fig. 39)

| Geometrical Optima | | | | | | | Dynamical Optima | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Symbol Group or Curve No. | | Values of $\beta$ | | | | | Symbol Group or Curve No. | | Values of $\beta$ | | | | |
| | | 60° | 120° | 180° | 270° | 360° | | | 60° | 120° | 180° | 270° | 360° |
| 1 | $A$ | 122.3° | 66.33° | 12.17° | −62.48° | −118.6° | 10 | $W\frac{h_o^2}{\mu N a^3}$ | 0.0201 | 0.0759 | 0.1543 | 0.2635 | 0.2939 |
| 2 | $\alpha$ | 32.94° | 66.49° | 101.3° | 156.6° | 208.6° | 11 | $F\frac{h_o}{\mu N a^2}$ | 0.0936 | 0.1863 | 0.2768 | 0.4080 | 0.5379 |
| 3 | $\alpha/\beta$ | 0.5490 | 0.5541 | 0.5627 | 0.5799 | 0.5796 | 12 | $\lambda\frac{a}{h_o}$ | 4.660 | 2.453 | 1.794 | 1.549 | 1.830 |
| 4 | $c$ | 0.7481 | 0.4821 | 0.3655 | 0.3310 | 0.3379 | 13 | $\lambda\frac{l}{h_o}$ | 4.880 | 5.138 | 5.637 | 7.297 | 11.50 |
| 5 | $h_m/h_o$ | 2.382 | 2.305 | 2.139 | 1.990 | 2.021 | 14 | $H\frac{h_o(10)^6}{\mu N^2 a^3}$ | 1.485 | 2.956 | 4.392 | 6.473 | 8.534 |
| 6 | $h_1/h_o$ | 1.200 | 1.209 | 1.226 | 1.257 | 1.266 | 15 | $H\frac{(10)^5}{W N h_o}$ | 7.394 | 3.893 | 2.847 | 2.457 | 2.904 |
| 7 | $\eta/h_o$ | 3.970 | 1.931 | 1.576 | 1.495 | 1.510 | 16 | $\frac{W N h_o}{H(10)^5}$ | 0.1352 | 0.2569 | 0.3513 | 0.4070 | 0.3444 |
| 8 | $\theta_1$ | 158.8° | 140.8° | 127.4° | 118.8° | 118.6° | 17 | $\lambda\Big/\sqrt{\frac{\mu N}{p_o}}$ | 0.4671 | 0.4780 | 0.4983 | 0.5620 | 0.7016 |
| 9 | $\phi$ | 155.3° | 132.8° | 113.5° | 94.09° | 90.00° | | | | | | | |



This chart shows values of non-dimensional group $(W\,N\,h_o/H)$ as affected by side leakage, with film forms for minimum coefficient of friction. Curve *J* gives values of $b/a$ at which group $(W\,N\,h_o/H)$ is greatest for each value of $\beta$. See Figs. 40 and 41 and Tables 3 and 4 for optimum conditions, assuming constant viscosity and no side leakage. When a curve for a given ratio of $b/a$ is chosen, each ordinate of it will be correct for the optimum relation of the symbols in this group. Hence the correction factors for side leakage are already allowed for in the heights of those ordinates.

FIG. 42. Side Leakage Chart for Class A Journal Bearings for Minimum Friction Coefficient. See Fig. 39.

**Table 4.—Optimum Values for Minimum Friction Coefficients of Class A Journal Bearings**
(See Fig. 39)

| Geometrical Optima. See Fig. 40 | | | | | | | Dynamical Optima. See Fig. 41 | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Symbol Group or Curve No. | | Values of $\beta$ | | | | | Symbol Group or Curve No. | | Values of $\beta$ | | | | |
| | | 60° | 120° | 180° | 270° | 360° | | | 60° | 120° | 180° | 270° | 360° |
| 1 | $A$ | 117.9° | 58.04° | 1.119° | −72.15° | −121.0° | 10 | $W\frac{h_o^2}{\mu N a^3}$ | 0.0197 | 0.0742 | 0.1507 | 0.2602 | 0.2925 |
| 2 | $\alpha$ | 33.89° | 68.54° | 104.7° | 161.4° | 211.0° | 11 | $F\frac{h_o}{\mu N a^2}$ | 0.0902 | 0.1790 | 0.2658 | 0.3985 | 0.5327 |
| 3 | $\alpha/\beta$ | 0.5649 | 0.5711 | 0.5815 | 0.5977 | 0.5860 | 12 | $\lambda\frac{a}{h_o}$ | 4.584 | 2.413 | 1.763 | 1.531 | 1.821 |
| 4 | $c$ | 0.7496 | 0.4911 | 0.3844 | 0.3520 | 0.3660 | 13 | $\lambda\frac{l}{h_o}$ | 4.801 | 5.054 | 5.540 | 7.217 | 11.44 |
| 5 | $h_m/h_o$ | 2.586 | 2.476 | 2.249 | 2.086 | 2.155 | 14 | $H\frac{h_o(10)^6}{\mu N^2 a^3}$ | 1.432 | 2.840 | 4.217 | 6.323 | 8.452 |
| 6 | $h_1/h_o$ | 1.248 | 1.263 | 1.283 | 1.292 | 1.280 | 15 | $H\frac{(10)^5}{WNh_o}$ | 7.274 | 3.829 | 2.798 | 2.430 | 2.890 |
| 7 | $\eta/h_o$ | 3.986 | 1.965 | 1.624 | 1.543 | 1.577 | 16 | $\frac{WNh_o}{H(10)^5}$ | 0.1375 | 0.2612 | 0.3574 | 0.4115 | 0.3460 |
| 8 | $\theta_1$ | 156.4° | 136.7° | 123.1° | 117.5° | 121.0° | 17 | $\lambda\Big/\sqrt{\frac{\mu N}{p_o}}$ | 0.4548 | 0.4647 | 0.4841 | 0.5524 | 0.6965 |
| 9 | $\phi$ | 151.8° | 126.6° | 105.8° | 89.24° | 90.0° | | | | | | | |

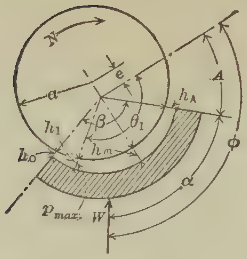


**Fig. 43.** Fitted Partial Bearing. $\alpha > \beta/2$, Class B

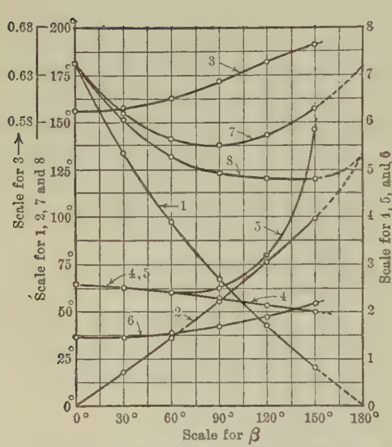


Fig. 44. Geometrical Optima for Minimum Friction Coefficients for Class B Journal Bearings. See Table 6.

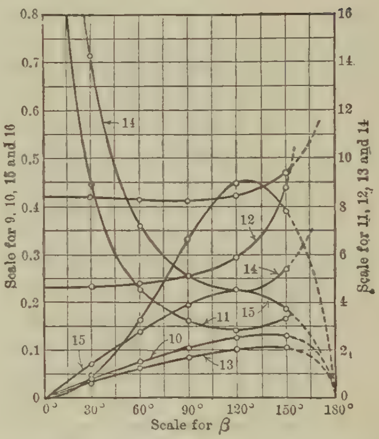


Fig. 45. Dynamical Optima for Minimum Friction Coefficients for Class B Journal Bearings. See Table 6.

**Table 5.—Optimum Values for Maximum Load $W$ of Class B Journal Bearings** (Fig. 43)

| Geometrical Optima | | | | | | | | Dynamical Optima | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Symbol Group or Curve No. | | Values of $\beta$ | | | | | | Symbol Group or Curve No. | | Values of $\beta$ | | | | | |
| | | 0° | 30° | 60° | 90° | 120° | 150° | | | 0° | 30° | 60° | 90° | 120° | 150° |
| 1 | $A$ | 180° | 120.1° | 92.21° | 64.30° | 40.67° | 19.39° | 9 | $W\frac{10h_o^2}{\mu Na^3}$ | 0 | 0.0451 | 0.1685 | 0.3309 | 0.4534 | 0.3924 |
| 2 | $\alpha$ | 0° | 17.47° | 35.79° | 55.72° | 77.76° | 101.1° | 10 | $F\frac{h_o}{\mu Na^2}$ | 0 | 0 0407 | 0.0774 | 0.1073 | 0.1284 | 0.1339 |
| 3 | $\alpha/\beta$ | 0.5774 | 0.5822 | 0.5964 | 0.6191 | 0.6480 | 0 6739 | 11 | $\lambda\frac{a}{h_o}$ | ∞ | 9.027 | 4.597 | 3.242 | 2.831 | 3.413 |
| 4 | $h_A/h_o$ | 2.189 | 2.176 | 2.143 | 2.078 | 1.969 | 1.804 | 12 | $\lambda\frac{l}{h_o}$ | 4.706 | 4.727 | 4.814 | 5.093 | 5.930 | 8.935 |
| 5 | $h_m/h_o$ | 2.189 | 2.176 | 2.143 | 2.306 | 3.021 | 5.433 | 13 | $H\frac{h_o(10)^6}{\mu N^2a^3}$ | 0 | 0.6455 | 1.229 | 1.702 | 2.037 | 2.125 |
| 6 | $h_1/h_o$ | 1.373 | 1.398 | 1.480 | 1.624 | 1.843 | 2.131 | 14 | $H\frac{(10)^5}{WNh_o}$ | ∞ | 14.32 | 7.293 | 5.144 | 4.492 | 5.415 |
| 7 | $\theta_1$ | 180° | 150.1° | 136.4° | 135.2° | 142.4° | 156.9° | 15 | $\frac{WNh_o}{H(10)^5}$ | 0 | 0.0698 | 0.1371 | 0.1944 | 0.2226 | 0.1847 |
| 8 | $\phi$ | 180° | 146.6° | 128.0° | 120.0° | 118.4° | 120.5° | 16 | $\lambda\Big/\sqrt{\frac{\mu N}{p_o}}$ | 0.4311 | 0.4285 | 0.4219 | 0.4171 | 0.4263 | 0.4781 |

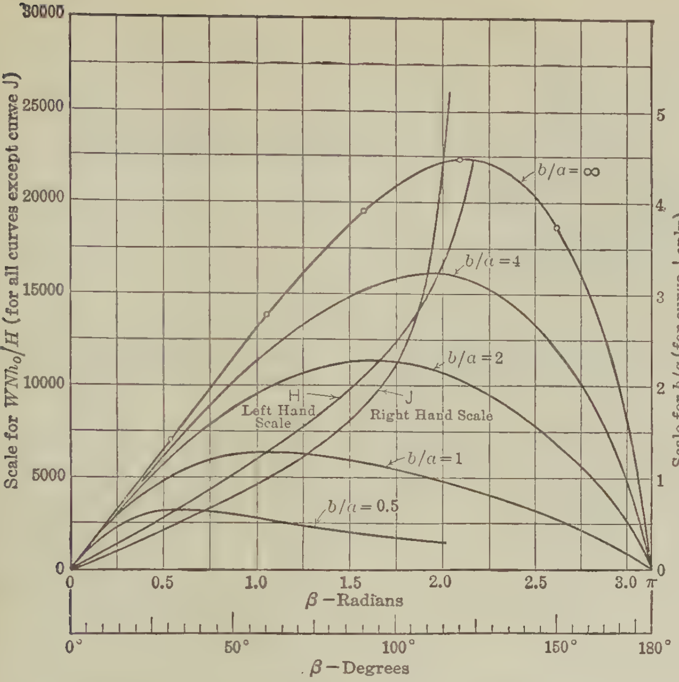


FIG. 46. Side Leakage Chart for Class B Journal Bearings for Minimum Friction Coefficients. See Fig. 43.

This chart shows values of the non-dimensional group ($W\ N\ h_o/H$) as affected by side leakage with film forms for the minimum coefficient of friction. Curve *J* gives the value of $b/a$ at which group ($W\ N\ h_o/H$) is greatest for each value of $\beta$. See Figs. 44 and 45 and Tables 5 and 6 for optimum conditions, assuming constant viscosity and no side leakage. When the curve for a given ratio of $b/a$ is chosen, each ordinate of it will be correct for the optimum relation of the symbols in this group. Hence the correction factors for side leakage are already allowed for in the heights of those ordinates.

**Table 6.—Optimum Values of Minimum Friction Coefficient for Class B Journal Bearings**
(See Fig. 43)

| Geometrical Optima. See Fig. 44 | | | | | | | | Dynamical Optima. See Fig. 45 | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Symbol Group or Curve No. | | Values of $\beta$ | | | | | | Symbol Group or Curve No. | | Values of $\beta$ | | | | | |
| | | 0° | 30° | 60° | 90° | 120° | 150° | | | 0° | 30° | 60° | 90° | 120° | 150° |
| 1 | $A$ | 180° | 133.0° | 95.59° | 66.29° | 41.73° | 20.08° | 9 | $W\frac{10h_o^2}{\mu Na^3}$ | 0 | 0.0443 | 0.1663 | 0.3281 | 0.4504 | 0.3876 |
| 2 | $\alpha$ | 0° | 17.84° | 36.26° | 55.95° | 77.45° | 99.41° | 10 | $F\frac{h_o}{\mu Na^2}$ | 0 | 0.0393 | 0.0755 | 0.1054 | 0.1267 | 0.1307 |
| 3 | $\alpha/\beta$ | 0.5914 | 0.5946 | 0.6043 | 0.6217 | 0.6455 | 0.6627 | 11 | $\lambda\frac{a}{h_o}$ | $\infty$ | 8.877 | 4.538 | 3.214 | 2.813 | 3.372 |
| 4 | $h_A/h_o$ | 2.533 | 2.501 | 2.408 | 2.276 | 2.122 | 1.993 | 12 | $\lambda\frac{l}{h_o}$ | 4.622 | 4.648 | 4.752 | 5.049 | 5.892 | 8.829 |
| 5 | $h_m/h_o$ | 2.533 | 2.501 | 2.408 | 2.486 | 3.189 | 5.804 | 13 | $H\frac{h_o(10)^6}{\mu N^2a^3}$ | 0 | 0.6238 | 1.197 | 1.673 | 2.010 | 2.074 |
| 6 | $h_1/h_o$ | 1.434 | 1.457 | 1.528 | 1.662 | 1.873 | 2.165 | 14 | $H\frac{(10)^5}{WNh_o}$ | $\infty$ | 14.08 | 7.201 | 5.100 | 4.463 | 5.351 |
| 7 | $\theta_1$ | 180° | 154.8° | 140.8° | 138.1° | 144.0° | 158.1° | 15 | $\frac{WNh_o}{H(10)^5}$ | 0 | 0.0710 | 0.1389 | 0.1961 | 0.2240 | 0.1869 |
| 8 | $\phi$ | 180° | 150.8° | 131.9° | 122.2° | 119.2° | 119.5° | 16 | $\lambda\Big/\sqrt{\frac{\mu N}{p_o}}$ | 0.4194 | 0.4177 | 0.4138 | 0.4117 | 0.4221 | 0.4695 |

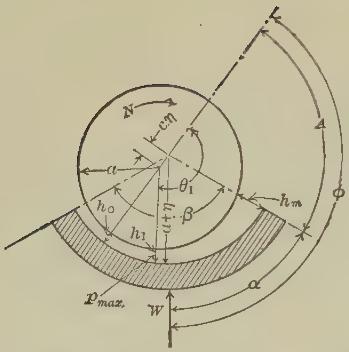


FIG. 47. Centrally-loaded Clearance Bearing, $\alpha = \beta/2$, Class C.

**Table 7.—Optimum Values for Maximum Load $W$ of Class C Journal Bearings** (Fig. 47)

| Geometrical Optima | | | | | Dynamical Optima | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Symbol Group or Curve No. | | Values of $\beta$ | | | Symbol Group or Curve No. | | Values of $\beta$ | | |
| | | 60° | 120° | 180° | | | 60° | 120° | 180° |
| 1 | $A$ | 131.2° | 83.66° | 37.83° | 10 | $W\dfrac{h_o^2}{\mu N a^3}$ | 0.0180 | 0.0673 | 0.1352 |
| 2 | $\alpha$ | 30° | 60° | 90° | 11 | $F\dfrac{h_o}{\mu N a^2}$ | 0.0982 | 0.1954 | 0.2902 |
| 3 | $\alpha/\beta$ | 0.50 | 0.50 | 0.50 | 12 | $\lambda\dfrac{a}{h_o}$ | 5.464 | 2.906 | 2.147 |
| 4 | $c$ | 0.7466 | 0.4688 | 0.3437 | 13 | $\lambda\dfrac{l}{h_o}$ | 5.722 | 6.087 | 6.744 |
| 5 | $h_m/h_o$ | 2.004 | 1.980 | 1.937 | 14 | $H\dfrac{h_o(10)^6}{\mu N^2 a^3}$ | 1.559 | 3.101 | 4.605 |
| 6 | $h_1/h_o$ | 1.137 | 1.144 | 1.158 | 15 | $H\dfrac{(10)^5}{WNh_o}$ | 8.670 | 4.611 | 3.406 |
| 7 | $\eta/h_o$ | 3.946 | 1.883 | 1.524 | 16 | $\dfrac{WNh_o}{H(10)^5}$ | 0.1153 | 0.2169 | 0.2936 |
| 8 | $\theta_1$ | 162.5° | 146.8° | 134.3° | 17 | $\lambda\Big/\sqrt{\dfrac{\mu N}{p_o}}$ | 0.5181 | 0.5329 | 0.5581 |
| 9 | $\phi$ | 161.2° | 143.7° | 127.8° | | | | | |

**Table 8.—Optimum Values for Minimum Friction Coefficient of Class C Journal Bearings**

| Geometrical Optima. See Fig. 48. | | | | | Dynamical Optima. See Fig. 49. | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Symbol Group or Curve No. | | Values of $\beta$ | | | Symbol Group or Curve No. | | Values of $\beta$ | | |
| | | 60° | 120° | 180° | | | 60° | 120° | 180° |
| 1 | $A$ | 131.6° | 84.39° | 38.71° | 10 | $W\dfrac{h_o^2}{\mu N a^3}$ | 0.0180 | 0.0671 | 0.1349 |
| 2 | $\alpha$ | 30° | 60° | 90° | 11 | $F\dfrac{h_o}{\mu N a^2}$ | 0.0980 | 0.1948 | 0.2890 |
| 3 | $\alpha/\beta$ | 0.50 | 0.50 | 0.50 | 12 | $\lambda\dfrac{a}{h_o}$ | 5.458 | 2.902 | 2.142 |
| 4 | $c$ | 0.7615 | 0.4904 | 0.3645 | 13 | $\lambda\dfrac{l}{h_o}$ | 5.716 | 6.078 | 6.731 |
| 5 | $h_m/h_o$ | 2.072 | 2.056 | 2.021 | 14 | $H\dfrac{h_o(10)^6}{\mu N^2 a^3}$ | 1.555 | 3.091 | 4.586 |
| 6 | $h_1/h_o$ | 1.140 | 1.148 | 1.163 | 15 | $H\dfrac{(10)^5}{WNh_o}$ | 8.660 | 4.604 | 3.399 |
| 7 | $\eta/h_o$ | 4.193 | 1.962 | 1.574 | 16 | $\dfrac{WNh_o}{H(10)^5}$ | 0.1155 | 0.2172 | 0.2942 |
| 8 | $\theta_1$ | 163.0° | 147.8° | 135.7° | 17 | $\lambda\Big/\sqrt{\dfrac{\mu N}{p_o}}$ | 0.5171 | 0.5317 | 0.5564 |
| 9 | $\phi$ | 161.6° | 144.4° | 128.7° | | | | | |

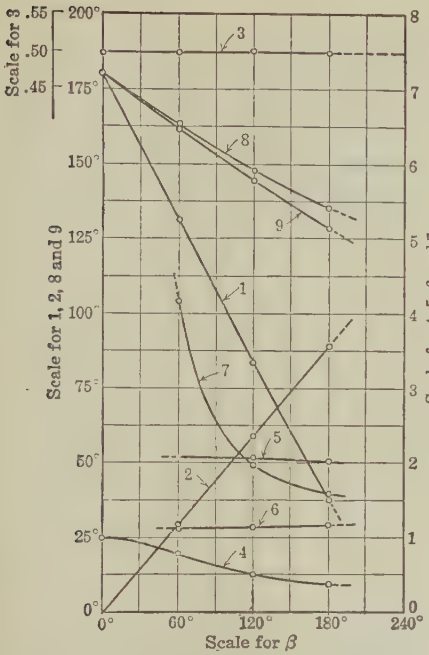


FIG. 48. Geometrical Optima for Minimum Friction Coefficient for Class C Journal Bearings. See Table 8.

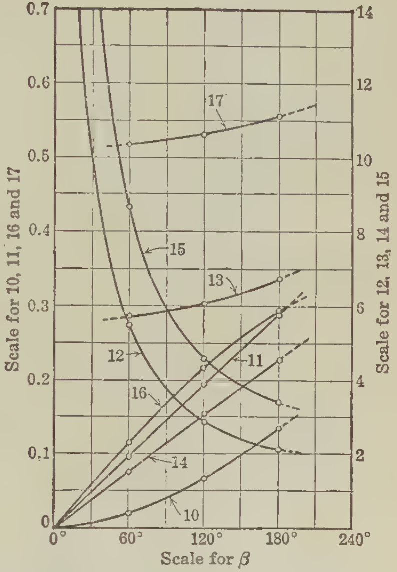
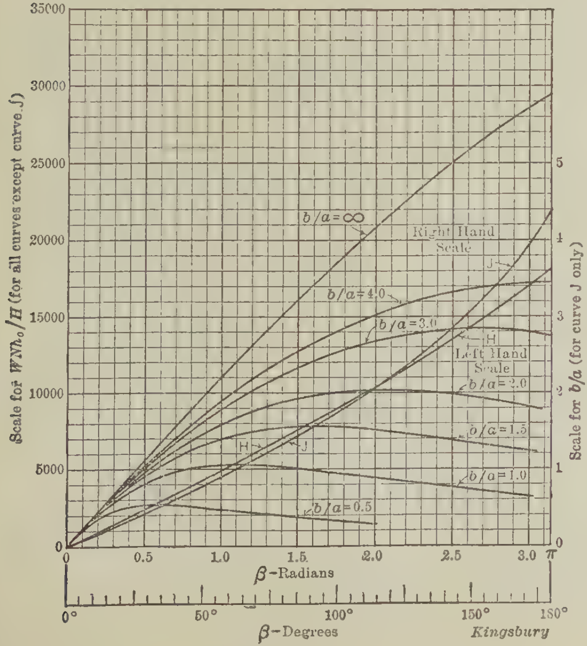


FIG. 49. Dynamical Optima for Minimum Friction Coefficient for Class B Journal Bearings. See Table 8.



*Kingsbury*

FIG. 50. Side Leakage Chart for Class C Clearance Bearings, for Minimum Friction Coefficients. See Fig. 47.

This chart shows the values of the non-dimensional group $(W\,N\,h_o/H)$ as affected by side leakage with film forms for the minimum coefficient of friction. Curve *J* gives the value of $b/a$ at which group $(W\,N\,h_o/H)$ is greatest for each value of $\beta$. See Figs. 48 and 49 and Tables 7 and 8 for optimum conditions, assuming constant viscosity and no side leakage. When the curve for a given ratio of $b/a$ is chosen, each ordinate of it will be correct for the optimum relation of the symbols in this group. Hence the correction factors for side leakage are already allowed for in the heights of those ordinates.

*(Continued from p. 13-17)*

Clearance $\eta$ is determined as follows: The ratio of $\eta/h_o$ is found from curve 7, Fig. 40, to be 1.72, corresponding to $\beta = 153.6$, when $\eta = 1.72\ h_o = 1.72 \times 0.003 = 0.00516$ in.

Bore diameter of bearing $= 2(a + \eta) = 2(4.564 + 0.00516) = 9.138$.

Clearance ratio $\eta/a = 0.00516/4.564 = 0.00113$.

To ascertain required oil viscosity coefficient, correction factors must be determined by means of Fig. 38. Arc length of bearing surface $l = \beta$ (radians) $\times\ a = 2.68 \times 4.564 = 12.23$ in. Width $b = (b/a) \times 4.564 = 13.69$, and $l/b = 0.893$. At the points on the several curves in Fig. 38 corresponding to $l/b = 0.893$, the following correction factors are found: $L_1 = 0.487$; $F_1 = 0.916$; $L_1/F_1 = 0.532$. To ascertain the mean viscosity in the oil film, refer to group 10 and curve 10, Table 4 and Fig. 41. For an infinitely wide bearing and $\beta = 153.6$, curve 10 shows the value of $W(h_o^2/\mu Na^3)$ to be 0.117. Applying the correction factor, $W(h_o^2/\mu Na^3)$, $L_1 = 0.117 \times 0.487 = 0.05698$. The actual mean load per unit width of bearing is $W = 25{,}000/13.69 = 1826$ lb. Substituting this and other known values in group 10, we find

$\mu = (1826 \times 0.003^2)/(0.05698 \times 1500 \times 4.564^3) = 2.022 \times 10^{-6}$ Reyn.

$= 69{,}000 \times 2.002 \times 10^{-6} = 0.1395$ poise, corresponding to a light machinery oil at 130° to 140° F. For corresponding commercial viscosities, see p. 13-55.

The coefficient of friction $\lambda$ is found by means of curve 12 and group 12, Fig. 41, and Table 4. Let $m$ = the value on curve 12 corresponding to value of $\beta$. In the present case, $\beta = 153.6$, whence $m = 2.00$. Then

$$\lambda = (F_1/L_1) \times m \times (h_o/a) = (2.00 \times 0.003)/(0.532 \times 4.564) = 0.00247.$$

Inspection of Fig. 42 shows that, other things being equal, an increase of the ratio $b/a$ will reduce the relative power loss. See Example 2. The possible increase in $b/a$ is limited by considerations of space, cost, and strength or elasticity of the journal. In Example 1, $\beta$ might be increased, following the curve $b/a = 3$, Fig. 42, to obtain a greater bearing area, with, however, a relatively greater friction loss. Other data for the bearing in question can be found from Figs. 40 and 41, and Tables 3 and 4; for example, leading angle $\alpha$ from group 3 and curve 3.

EXAMPLE 2.—Assume the same data as for Example 1, excepting that $b/a = 4$. Then using the same curves and tables we have,

$a = \sqrt{125/8} = 3.953$ in.; $\beta = 180.8$ deg. $= 3.155$ radians for least power loss.
Total power loss $= (25{,}000 \times 1500 \times 0.003)/20{,}700 = 5.435$ Hp.
Clearance $\eta = 1.622\ h_o = 0.004866$ in., $= 0.00123$ in. per inch of diameter.
Length $l = 3.155 \times 3.953 = 12.47$ in. Width $b = 4 \times 3.953 = 15.81$ in.
Ratio $l/b = 0.789$. This determines the correction factors:
$L_1 = 0.533$; $F_1 = 0.924$; $L_1/F_1 = 0.577$.
$W = 25{,}000/15.8 = 1581$ lb.
$\mu = (1581 \times 0.003^2)/(0.1517 \times 0.533 \times 1500 \times 3.953^3) = 1.90 \times 10^{-6}$ Reyn, corresponding to 0.131 poise.
$\lambda = (F_1/L_1) \times m \times (h_o/a) = (1.762 \times 0.003)/(0.577 \times 3.953) = 0.002318$.

The charts and tables also may be used to determine proper radial clearance or oil viscosity for a known set of other conditions. See Example 3.

**B. Fitted Partial Bearings** (Fig. 43).—The journal and bearing have the same radius of curvature. The line of application of the load does not pass through the center of the bearing arc, and $\alpha > \beta/2$. Figs. 44 and 45 and Tables 5 and 6 show optimum conditions for values of $\beta$ from 0 to 150 deg. The procedure followed in design is the same as that for Class A bearings, except that reference is made to the tables and curves for Class B bearings corresponding to those used for Class A bearings in Examples 1 and 2. See also Example 5.

**C. Centrally-loaded Partial Bearings with Running Clearance** (Fig. 47).—The radius of curvature of the bearing is greater than that of the journal. The line of application of the load passes through the center of the bearing arc, and $\alpha = \beta/2$. Figs. 48 to 50 and Tables 7 and 8 show optimum conditions for values of $\beta$ from 0 to 180 deg. Fig. 51 shows general conditions for values of $\beta$ from 60 to 180 deg. Bearing problems may be solved by the same procedure as used for Examples 1 and 2 except that reference should be made to the curves and tables of Class C bearings. Example 3 shows the application of the curves to the selection of the correct radial clearance $\eta$.

EXAMPLE 3.—Determine the best radial clearance $\eta$ for a bearing in which the total load $R =$ 25,000 lb., $N = 1500$ r.p.m., $\beta = 120$ deg., $p_o = 125$ lb. per sq. in., ratio of width to radius, $b/a = 4$, oil viscosity, $\mu = 3.4 \times 10^{-6}$ Reyn.* Bearing is centrally loaded.

*Solution.*—By procedure outlined in Example 1 and from the foregoing data, $2a = 10$ in., $b =$ 20 in., $W = 1250$, $l = 10.47$ in., $l/b = 0.524$.

Correction factors from Fig. 38 will be $L_1 = 0.672$, $F_1 = 0.947$, $L_1/F_1 = 0.710$. Then from group 10 in Table 8, by introducing correction factor $L_1$, we have $Wh_o^2/\mu Na^3 = 0.0672\ L_1 = 0.0451$. Substituting the known values in this equation, and solving for $h_o$, we find $h_o = 0.0048$. In group 7, Table 8, the value of $\eta/h_o$ for $\beta = 120$ deg. is 1.962, whence $\eta = 1.962 \times h_o = 1.962 \times 0.0048 =$ 0.0094. Clearance ratio $= \eta/a = 0.0094/5 = 0.00188$. Friction coefficient factor $\lambda a/h_o$ is found

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* See Viscosity-temperature conversion chart, p. 13-56, to obtain corresponding commercial viscosities.

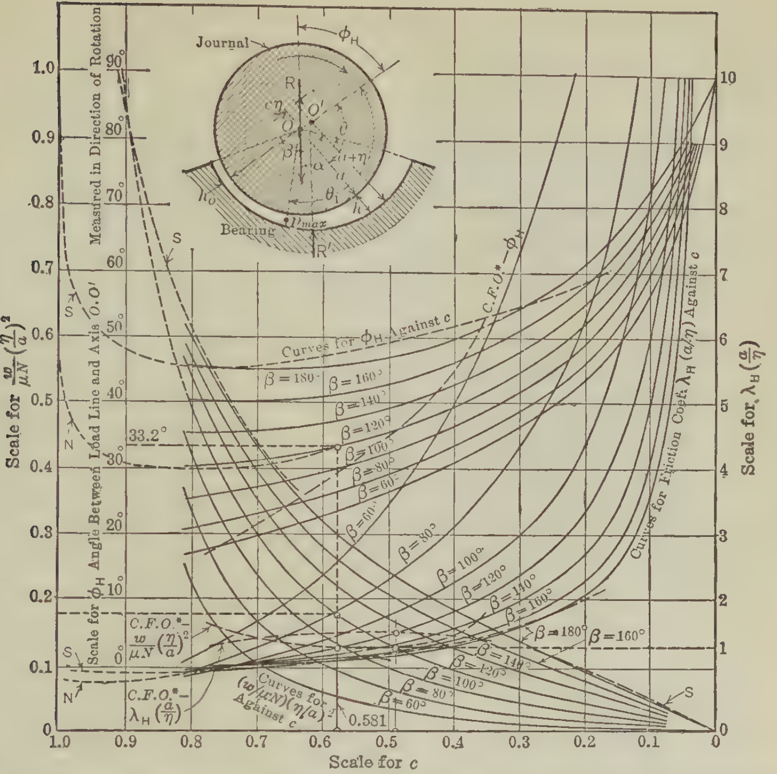

FIG. 51. General Characteristics of Centrally-loaded Clearance Bearings, Class C,† assuming constant velocity and no side leakage

C. F. O. = Class C friction optima. Where the dotted curves cut the corresponding full line curves the values are identical with the optimum conditions for minimum friction coefficient in Figs. 48 and 49 and Table 8.

from group 12, Table 8 to be 2.902, to which the correction factor $L_1/F_1$ must be applied. Hence,

$$\lambda = (h_o/a) \times 2.902 \times \{1/(L_1/F_1)\} = (0.0048/5) \times 2.902 \times (1/0.710) = 0.00392.$$

For conditions other than optimum the charts Figs. 51 to 56 may be used. The side leakage correction factors given in Fig. 38 may be applied to these curves with fair approximation. See Example 4.

EXAMPLE 4.—Given a centrally-loaded 120-deg. bearing with a total load $R$ = 10,000 lb., and actual nominal pressure $p_o$ = 125 lb. per sq. in. Journal diameter is 8 in. and bearing width is 10 in. Clearance ratio $\eta/a = 0.001$, and speed $N$ = 400 r.p.m. Determine the running position of the journal when using two oils of viscosity coefficient $\mu = 3.4 \times 10^{-6}$ Reyn, and $10.2 \times 10^{-6}$ Reyn, respectively.

*Solution.*—Referring to Fig. 51, the scale at the left represents a symbol group in which all values are known except $w$, which is the ideal nominal unit pressure that the bearing would sustain if side leakage could be neglected. This ideal pressure is reduced to actual pressure $p_o$ by side leakage, and $p_o = L_1 w$. Correction factor $L_1$ is found from Fig. 38. The ratio $l/b = \{(120/360)\pi 8\}/10 = 0.839$. Entering Fig. 38 at $l/b = 0.839$, the correction factors are found to be as follows: $L_1 = 0.51$; $L_1/F_1 = 0.555$; $F_1 = 0.92$. Then $W = 125/0.51 = 245$, and

$$(w/\mu N)(\eta/a)^2 = (245 \times 1 \times 10^{-6})/(\mu \times 400) = (0.613 \times 10^{-6})/\mu.$$

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[*] Curve C. F. O. – $(w/\mu N)\,(\eta/a)^2$ is the locus for the loading that is proper for these optima for various values of $\beta$. Curve C. F. O. – $\lambda_H(a/\eta)$ is the locus of the corresponding friction coefficients. Curve C. F. O. – $\phi_H$ is the locus of the angles that determine the journal running positions. The three points for each angle $\beta$ are on a vertical line through the corresponding eccentricity factor on the base scale, as for example the point marked for $c$ = 0.49 and $\beta$ = 120°.

[†] This chart has been adapted from earlier one by Howarth (7) and contains added curves and points obtained from the work of other investigators. For dotted curves $S$, see Sommerfeld (14). For dotted curves $N$, see Needs (15). For dotted curves, marked C. F. O., see Kingsbury (5). These numbers refer to the bibliography, p. 13–58.

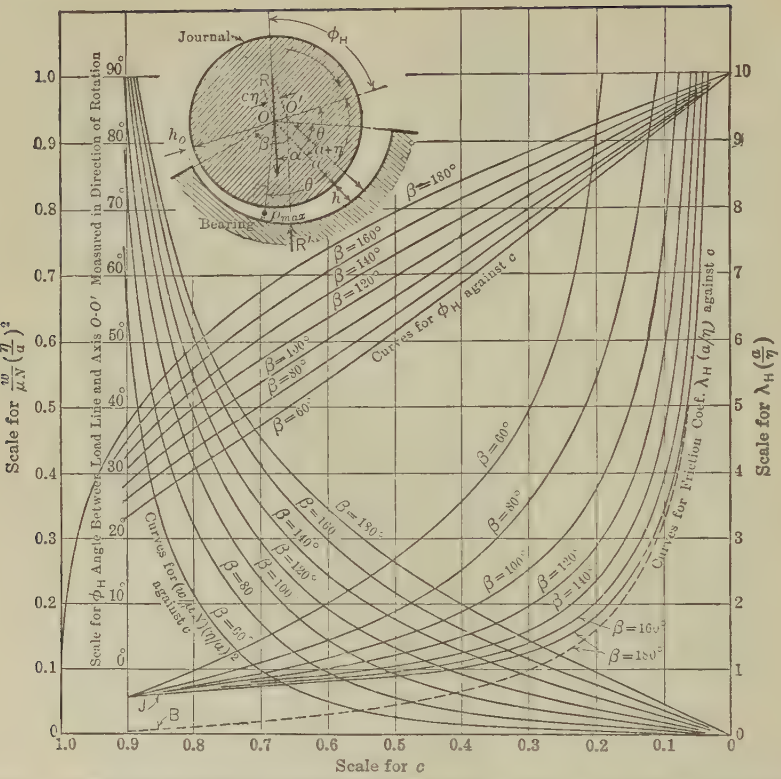


FIG. 52. General Characteristics of Offset Clearance Bearings, Class D, Assuming Constant Viscosity and No Side Leakage. See Fig. 53 for Corresponding Ratios of $\alpha/\beta$.

This class of bearings differs somewhat from the eccentrically-loaded bearings in Class A.

The next step is to substitute in the above expression the two values of $\mu$ given in the statement of the problem, and with the resulting values enter Fig. 51 at the left. First substituting $\mu = 3.4 \times 10^{-6}$, $(w/\mu N)(\eta/a)^2 = 0.180$. Locating this point on the load-capacity curve, $\beta = 120$ deg. (the group of curves starting at the lower right) find $c = 0.581$. The ordinate of $c = 0.581$ intersects the friction coefficient curve for $\beta = 120$ deg. (group of curves starting at the lower left) at $\lambda_H a/\eta = 1.29$. This ordinate also intersects the $\phi_H$ curve (group of curves starting at the upper right) for $\beta = 120$ deg. at $\phi_H = 33.2$ deg.

The actual friction coefficient will be $\lambda = (F_1/L_1)\lambda_H = 1.29 \times 10^{-3}/0.555 = 0.00233$.

The journal running position will be on the line $\phi_H = 33.2$ deg. and its distance from the center of the bearing will be $c\eta = 0.581 \times (8/2) \times 0.001 = 0.002324$ in. The minimum film thickness will be $h_o = (1 - c)\eta = 0.001676$ in. Optimum eccentricity is shown by the dotted C. F. O. loading curve $(w/\mu N)(\eta/a)^2$ to be 0.49 for a 120-deg. bearing, with a corresponding value of $h_o = 0.00204$. Hence, with $\mu = 3.4 \times 10^{-6}$, this bearing is not quite so safe as the optimum would be.

Solving the problem with $\mu = 10.2 \times 10^{-6}$, the value of $(w/\mu N)(\eta/a)^2$ for entering the chart is 0.060. The following values then are established:

$$c = 0.28;\ \phi_H = 46 \text{ deg.};\ \lambda_H a/\eta = 2.54;\ \lambda = 0.00458;\ h_o = 0.00288.$$

Friction and minimum film thickness are greater with the heavier oil. An intermediate oil would cause the journal to run with optimum eccentricity. The decision then rests on which of the three oil film thicknesses $h_o$ = 0.00168, 0.00204, or 0.00288 in. best suits the required shaft alignment and other conditions. Heavier oil is best where shaft alignment conditions are not ideal. See Shaft Deflections, p. 13–12, and also the film thickness noted in Table 11.

**D. Offset Partial Bearings with Running Clearance.**—These are eccentrically loaded bearings that differ somewhat from the eccentrically loaded partial bearings of Class A. The line of application of the load does not pass through the center of the bearing arc and

$\alpha > \beta/2$. The relation of journal and bearing is shown in the inset in Fig. 52. In this series a particular value of $\alpha/\beta$ refers to each value of $\beta$ and each running position (eccentricity). The characteristics of a bearing with a given value of $\beta$ and $\alpha/\beta$ ratio are shown by one point only on Fig. 53 and by another on Fig. 52, but are not available for a combination of $\beta$ and $\alpha/\beta$ throughout any range of running positions. If on the other hand, entry is made through Fig. 52, in the same manner as explained in Example 4, and a suitable eccentricity is determined, Fig. 53 will give the corresponding ratio of $\alpha/\beta$. Side leakage corrections may be applied as in Example 4. If the useful film has an angular length $\beta$ that is less than the film space provided, or if the angle $\beta$, whether constant or variable with eccentricity, changes its $\alpha/\beta$ ratio with change of eccentricity, Fig. 52 will prove useful for comparing theory with actual practice.

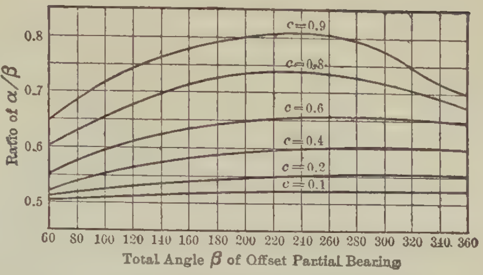
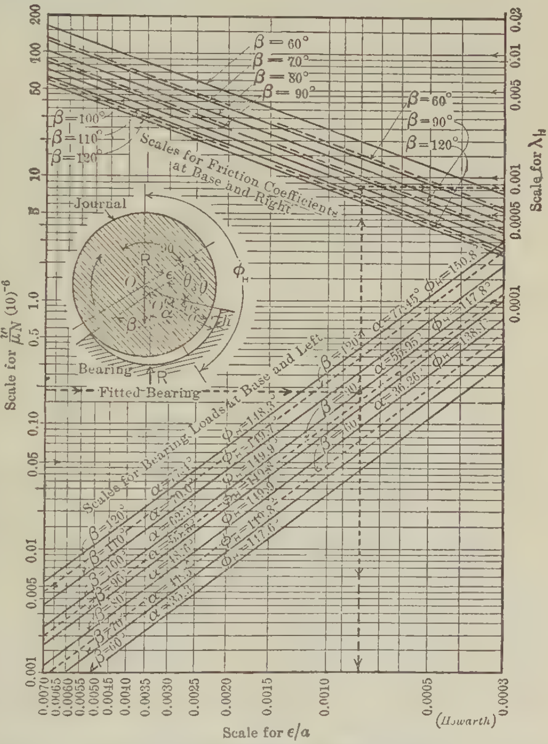


FIG. 53. $\alpha/\beta$ Values of Class D Journal Bearings. For other characteristics, see Fig. 52.



(Howarth)

FIG. 54. General Characteristics of Fitted Bearings, Class E. This series differs slightly from Class B as indicated by the broken lines for $\beta$ = 60 deg., $\beta$ = 90 deg., and $\beta$ = 120 deg., which represent the optima for minimum friction coefficient.

**E. Fitted Partial Bearings.**—The radius of curvature of the bearing is the same as that of the journal. The line of application of the load does not pass through the center of the bearing arc, and $\alpha > \beta/2$. The relations of bearing and journal are shown in the inset in Fig. 54. Bearings of this class differ slightly from Class B bearings. In Fig. 54, which gives general conditions for a definite series having variable $\alpha/\beta$ ratios with values of $\beta$ from 60 to 120 deg., the nearest equivalent to a clearance ratio is the scale $\epsilon/a$, in which $\epsilon$ is the actual eccentricity of the running position of the journal. Angle $\phi_H$, marked on the diagonal for the loading of the particular bearing of angle $\beta$ that is under consideration, shows the direction of shift of the journal center. See Example 5.

EXAMPLE 5.—Given a 100-deg. fitted bearing of journal diameter 5 in., and width 6 in., carrying a total load $R$ of 3000 lb., running at $N$ = 300 r.p.m., and using oil of viscosity $\mu = 3 \times 10^{-6}$ Reyn. Find the minimum film thickness.

*Solution.*—$l/b = (5\pi/6) \times (100/360) = 0.727$, from which (see Fig. 38) $L_1 = 0.565$, and $w = 3000/(6 \times 5 \times 0.565) = 177$. Then $(w/\mu N) \times 10^{-6} = (177 \times 10^{-6})/(3 \times 10^{-6} \times 300) = 0.197$, which value is used to enter Fig. 54. The intersection of this value with the lower diagonal $\beta$ = 100, gives a value of $\epsilon/a$ of 0.0008. The minimum thickness of oil film $h_o$ will be at the trailing edge of the bearing, and is, numerically, $h_o = \epsilon \sin \theta$. From inset in Fig. 54, $\theta = 270° + \beta - (\phi_H + \alpha) = 157.6°$. Hence $h_o = 0.0008 \times 2.5 \times 0.381 = 0.00076$.

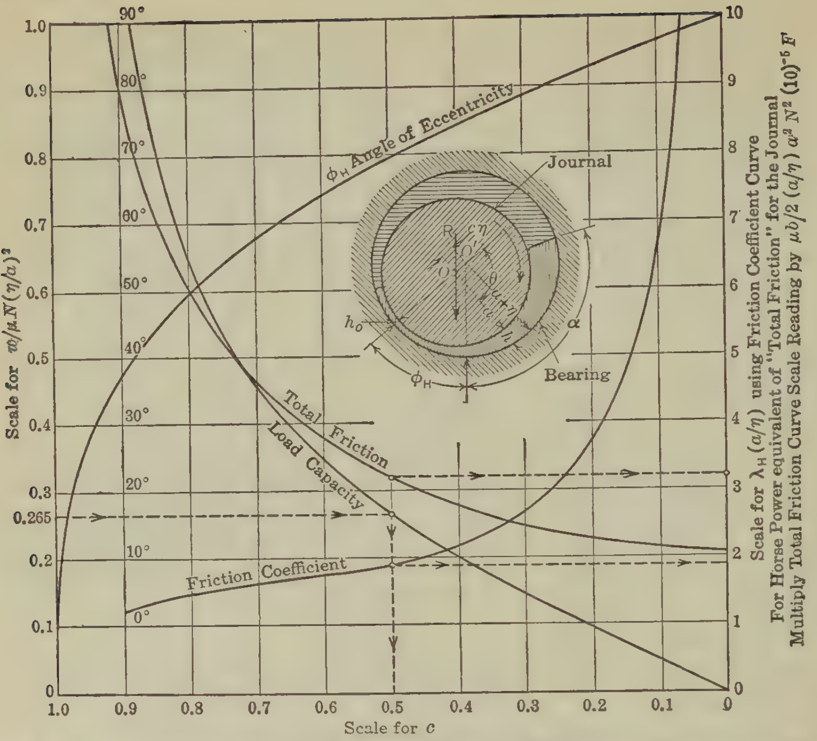


FIG. 55. General Characteristics of Full Bearings, Class F, with Running Clearance, Assuming Constant Viscosity and No Side Leakage.*

Other data that may be obtained from Fig. 54 are: Uncorrected coefficient of friction $\lambda_H = 0.00082$. Applying correction factor $L_1/F_1 = 0.61$ (Fig. 38), the actual coefficient of friction

$$\lambda = \lambda_H/(L_1/F_1) = 0.00082/0.61 = 0.00134.$$

**F. Full Bearings with Running Clearance.**—This bearing which is shown in the inset in Fig. 55 is treated as a half-bearing so far as carrying capacity and running position are concerned. As regards friction, it is treated as a full bearing in which the running position of the journal is the same as for a half-bearing. The line of application of the load, when the film is in running position would not pass through the center of the supporting film arc. The useful film of the bearing, therefore, may be treated as that of an offset bearing, Class D. In Fig. 55, the load capacity curve is taken from Fig. 52 for $\beta = 180$ deg., Class D bearings. The pressure film in full bearing is not apt to exceed 180 deg. in angular extent. The experiments of Stanton (16), Bradford (17), and McKee (18)† indicate shorter films for certain conditions. For other arcs than 180 deg., the appropriate curve for load capacity in Fig. 52 can be used to find the eccentricity. Fig. 55 then can be entered with the eccentricity so found, the approximate total friction determined and the horsepower loss estimated. The friction coefficient curve in Fig. 55 can be used only with the load capacity curve in the same illustration. Example 6 shows the use of side leakage correction factors in a full bearing.

EXAMPLE 6.—Determine the friction power loss and friction coefficient of a full bearing of diameter $2a = 4$ in., width $b = 8$ in., running at $N = 3600$ r.p.m., and carrying a total journal load $R = 3600$ lb. The clearance ratio $\eta/a = 0.003$, and the oil viscosity $\mu = 2 \times 10^{-6}$ Reyn.

*Solution.*—In order to find $L_1$, assume $\beta = 180$ deg. Then $l/b = (4\pi/8) \times (180/360) = 0.79$, and from Fig. 38 $L_1 = 0.53$. $F_1 = 0.925$, and $L_1/F_1 = 0.575$.

$$w = R/(2a \times b \times L_1) = 3600/(4 \times 8 \times 0.53) = 212 \text{ lb. per sq. in.}$$

---

* The load capacity curve corresponds to that for an offset (Class D) bearing for which $\beta = 180$ deg. (See explanations on page 13–26.) The friction curves assume that for the purpose of calculating this loss the bearing film may be considered as extending all around the journal. This assumption is arbitrary but appears to be well on the safe side.

† See Bibliography, p. 13–58.

Fig. 55 is entered with the value of the symbol group

$$(w/\mu N)(\eta/a)^2 = (212 \times 9 \times 10^{-6})/(2 \times 10^{-6} \times 3600) = 0.265.$$

This value on the load capacity curve corresponds to a value of eccentricity factor $c = 0.5$. For this value of $c$ the value of $\lambda_H a/\eta$ is read from the friction coefficient curve as 1.88. The total friction factor is read from the total friction curve as 3.2.

$$\text{Total power loss} = \text{Total friction factor} \times (\mu b/2)(a/\eta)a^2 F_1 N^2 \times 10^{-5} = 4.1 \text{ Hp.}$$

$$\text{Actual coefficient of friction } \lambda = \lambda_H/(L_1/F_1) = \lambda_H/0.583 = 0.0098.$$

Checking back $\lambda_H$ to the total power loss, the latter is found to be the same if the correction factor is properly introduced in $\lambda = (F_1/L_1)\lambda_H$.

**GENERAL CHARACTERISTICS OF JOURNAL BEARINGS.**—Fig. 57 may be used to study the influence on running position, in all classes of 120-deg. clearance bearings, of such changes in design proportions as the $(\alpha/\beta)$ ratio, and the clearance ratio $(\eta/a)$. Side leakage correction factors may be applied as in Example 4, but only for $L_1$ as in $p_0 = w\, L_1$. The reliability of factor $L_1$, decreases with increased departure from good film proportions. Kingsbury's optimum points for maximum load and minimum friction for Class A bearings are designated by ALO and AFO, respectively. Similar maximum load and minimum friction points for Class C bearings are designated by CLO and CFO, respectively. These points permit the determination of the deviation from optimum of the actual conditions in a given bearing. Fig. 57 also gives the angular location, $\theta_1$, of the maximum film pressure. These curves may be used to determine whether there is a possibility of negative pressures existing in the oil film. All pressures in the film are positive between the lines $M$ and $N$. An unpublished investigation by S. J. Needs shows that negative pressure is to be expected in a finite bearing where the bearing of infinite width would show all pressures to be positive. Hence, when the data of a problem show that the running position approaches the upper limit line of Fig. 57, some negative pressure is probable. If such conditions exist, it is advisable to place the problem in some other class of bearing, using $\alpha = 60$ deg. as suggested below.

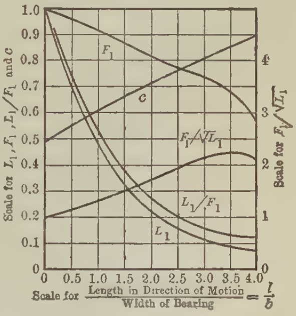


Fig. 56. Correction Factors for Influence of Side Leakage on Load Carrying Capacity and Friction for 120-deg. Class C Journal Bearing. See Fig. 38.

S. J. Needs has determined (15), for 120-deg. centrally-loaded bearings, the optimum eccentricities and the correction factors, for length-width ratios up to 4.0. His results are best shown in Fig. 56 which is in a form comparable with Kingsbury's chart, Fig. 38. The eccentricity for optimum friction condition is shown to vary from 0.49 for $l/b = 0$, to 0.90 for $l/b = 4.0$, with approximately straight line interpolation between. He further concludes that side leakage factors depend on bearing loads as well as length-width ratios, and that the length-width ratio influences the eccentricity for optimum operating conditions. See discussion in the paper referred to.

If conditions of operation are such that negative pressures may occur at the trailing end, 120-deg. bearings are best studied as if they were Class A, eccentrically-loaded bearings, or Class D, offset bearings with $\beta < 120$ deg. In both cases, leading angle $\alpha$ should be taken as 60 deg., allowing the trailing angle $(\beta - \alpha)$ to be determined by the data in the class used. See Example 7.

Example 7.—Determine whether negative pressure would exist in a journal of diameter $2a = 6$ in., and width $b = 10$ in., carrying a total load $R = 600$ lb. at $N = 1000$ r.p.m., taking $\alpha/\beta$ ratios as follows: [1]. $\alpha/\beta = 0.44$; [2]. $\alpha/\beta = 0.52$; [3]. $\alpha/\beta = 0.62$. Clearance ratio $\eta/a = 0.002$, and the oil viscosity $\mu = 4.0 \times 10^{-6}$ Reyn.

*Solution.*—Determine correction factor $L_1$ corresponding to $l/b = (6\pi/10) \times (120/360) = 0.628$. From Fig. 38, $L_1 = 0.613$, and $w = 600/(6 \times 10 \times 0.613) = 16.30$. Since $\mu = 4.0 \times 10^{-6}$, the group

$$(w/\mu N)(\eta/a)^2 = [16.3/\{(4 \times 10^{-6}) \times 1000\}] \times 0.002^2 = 0.0163.$$

If an ordinate be erected at 0.0163 on the scale $(w/\mu N)(\eta/a)^2$ in Fig. 57 it will intersect the curves for $\alpha/\beta = 0.44$, 0.52, and 0.62 at the points [1], [2], and [3] respectively. Hence, with ratios of $\alpha/\beta$ of 0.44 and 0.62 negative pressure is quite likely to exist at the trailing and leading ends, respectively, of the bearing. With a ratio of $\alpha/\beta = 0.52$ the bearing probably will not have negative pressure.

Other items that may be determined from Fig. 57 are the total lift of the bearing and the point of maximum pressure in the oil film. See Example 8.

Example 8.—Determine the lift of the journal when in running position, and the point of maximum pressure in the oil film of a centrally-loaded, type Class C, 120-deg. clearance turbine bearing without a cap, with ratio of $\alpha/\beta = 0.56$; journal diameter $2a = 15$ in.; length $b = 30$ in.; vertical load $R = 67{,}500$ lb.; speed $N = 1500$ r.p.m. Clearance ratio $\eta/a = 0.001$ in. The oil at operating temperature has a viscosity $\mu = 3.4 \times 10^{-6}$ Reyn.

*Solution.*—Ratio $l/b = (15\,\pi/30) \times (120/360) = 0.524$, whence from Fig. 38, $L_1 = 0.673$.

$$w = R/(2a \times b \times L_1) = 67{,}500/(15 \times 30 \times 0.673) = 223.$$

$$(w/\mu N)(\eta/a)^2 = \{223/(3.4 \times 10^{-6} \times 1500)\} \times (0.001)^2 = 0.0437,$$

with which value Fig. 56 is entered. The ordinate erected at $(w/\mu N)(\eta/a)^2 = 0.0437$ intersects the curve $\alpha/\beta = 0.56$ at $\phi_H = 107$ deg., which corresponds to a value of $c = 0.3$.

$$\text{Lift} = \eta(1 - c \cos \phi_H) = 0.075\,[1 - \{0.3 \times (-0.29237)\}] = 0.0684 \text{ in.}$$

The point of maximum pressure $p_m$ is the value of $\theta_1$ at the point where the ordinate $(w/\mu N)(\eta/a)^2 = 0.0437$ intersects the curve $\alpha/\beta = 0.60$, which is $\theta_1 = 83$ deg. $p_m$, therefore, will be on the radius of the journal in running position, 83 deg. in the direction of rotation from the line $O-O'$ joining the center of the journal and the center of curvature of the bearing.

**SPEED LIMITATIONS OF OIL FILM BEARINGS.**—The upper limit of applicability of oil films in thrust and journal bearings has not yet been attained (1937). Tests conducted in 1934 on two opposed double thrust bearings measuring 6 1/4 in. diam. over the shoes, and two journal bearings, 2 1/2 in. diam. by 30 in. long, in the same mountings, showed that the bearings operated perfectly up to 12,000 r.p.m. The thrust capacities reached 2000 lb. per sq. in. before breakdown. Maximum peripheral speed of the thrust bearing was 330 ft. per sec.

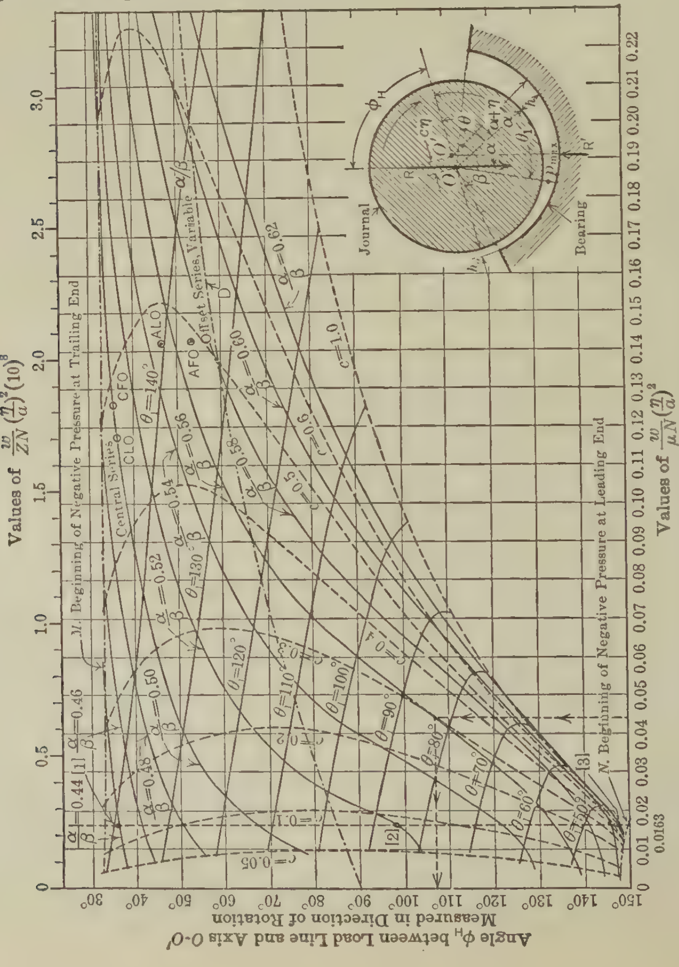


FIG. 57. General Capacity Characteristics of 120-deg. Clearance Bearings, Including Classes A, C and D, Assuming Constant Viscosity and No Side Leakage. Angular Length of Bearing $\beta$ = 120 deg.

## 4. CURRENT PRACTICE IN BEARING LOADING AND CLEARANCE

Data on current (1937) practice in the loading and clearance of bearings are useful for comparison with theoretical considerations in design. Data were collected from the following manufacturers, and engineering textbooks, and incorporated in Figs. 58–60.

*Manufacturers*

1. Westinghouse Electric & Mfg. Co.
2. Newport News Shipbuilding Co.
3. Electric Boat Co.
4. E. I. DuPont De Nemours Co.
5. Wm. Sellers Co.
6. General Electric Co.
7. De Laval Separator Co.
8. U. S. Navy Specification S-45-1
9. Kingsbury Machine Works, Inc.

*Textbooks, Handbooks and Engineering Periodicals*

10. Kimball and Barr, Elements of Machine Design, 1923
11. F. A. Halsey Handbook, 1913.
12. Hutte, 26th Edition, 1926
13. *Machinery's* Handbook, 1930
14. C. Pfleiderer, *Die Kreiselpumpen*, 1932
15. Prescott Poole, *Journal*, S.A.E. Oct., 1931.

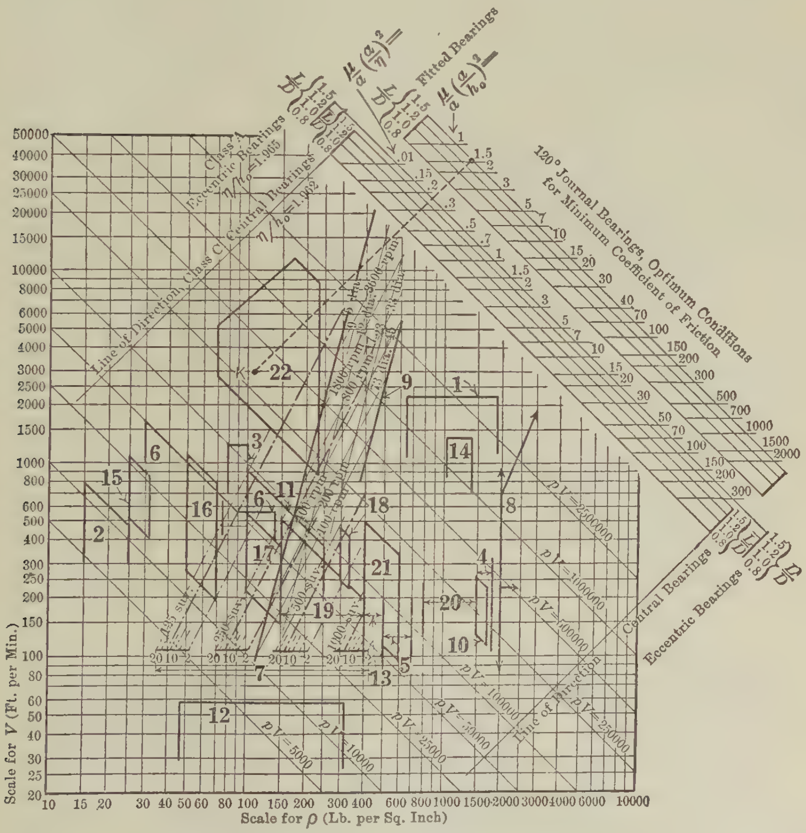


Fig. 58. Current Practice in Loading of Bearings

1. Aircraft Engine Crank-pins [15]
2. Cast Iron Bearings [10]
3. Centrifugal Pumps [4 14]
4. Gas Engine Crank-pins [10]
5. Gas Engine Main Bearings [10]
6. Generators [10]
7. Kingsbury Journal Bearings [9]
8. Roll Neck Bearings
9. Kingsbury Thrust Bearings [9]
10. Locomotive Crank-pins [10]
11. Machine Tools [5]
12. Machine Tools, Slow Running [5]
13. Marine Engine Bearings [10 11 12]
14. Marine Gas Engine Crank-pins [3]
15. Marine Lineshaft Bearings [2]
16. Marine Thrust Collars [11 12]
17. Ordinary Bearings—Bronze or Babbitt [10 12]
18. Railway Cars [10 12]
19. Stationary Engine Main Bearings [10 11]
20. Stationary Engine Overhung Crank-pins [10]
21. Stationary and Marine Engine Crank-pins [10 11 12]
22. Steam Turbines and Reduction Gears [1 2 4 6 10 12 13]

References are to list of manufacturers and textbooks above.
S. U. V. = Saybolt Universal Viscosity.

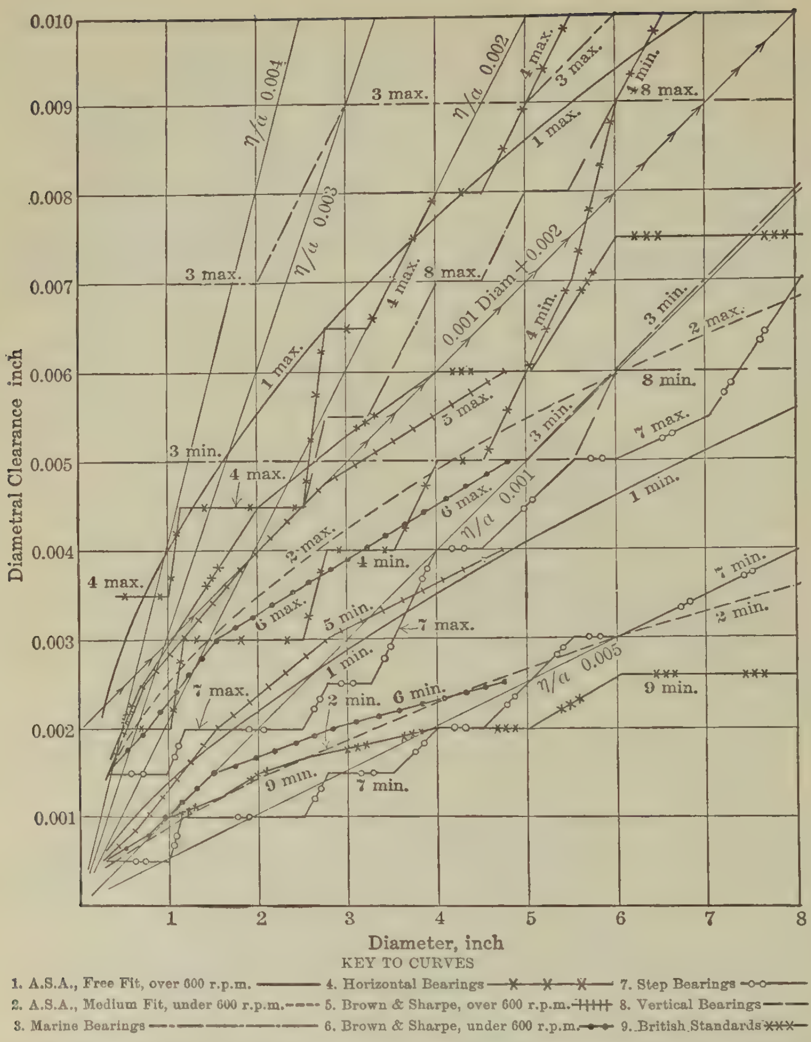


Fig. 59. Current Practice (1937) in Bearing Clearance, Shaft Diameter, 0 to 8 in. See Fig. 60 for Shafts 6 to 24 in. Diameter.

**BEARING PRESSURES AND SPEEDS.**—Fig. 58 gives the relation of (pressure × velocity), together with other data that are useful for analytical studies, on bearings covering a wide range of services. For journal bearings, the nominal unit pressure $p_o$ has been determined by the equation $p_o = R/2\,a\,b$, where $R$ = total load on bearing, lb.; $a$ = radius of bearing, in.; $b$ = width of bearing, in. Logarithmic scales are used for $p_o$ and $V$ so that straight lines will represent simple exponential relations. The diagonal lines sloping downward to the right give values of the product of $p_o\,V$ for any point on the chart. The heavy lines indicate, in general, the range of the data.

The diagonal lines slanting downward to the left, "Lines of Direction" and lines that may be drawn parallel to them, are used to study journal bearings. They afford means of obtaining the optimum relation between viscosity, journal diameter and clearance. For example, the dotted line from point $K$, extending across the scales of Kingsbury's optima for Classes A, B and C, leads to numerical values therein for equating with the symbol groups $(\mu/a) \times (a/\eta)^2$ and $(\mu/a)\,(a/h_o)^2$, where $\mu$ = coefficient of viscosity, Reyns; $a$ = radius of journal, in.; $\eta$ = radial clearance, in.; $h_0$ = minimum film thickness, in. The scale values of the symbol group are given for $l/b$ ratios from 0.8 to 1.5 for optimum journal bearings for minimum friction coefficient, based on a bearing arc $\beta = 120$ deg. Side leakage influence is allowed for the (length/width) ratios marked on the scales, based on Fig. 38.

**BEARING CLEARANCES.**—A study of bearing clearances indicates wide divergences in practice. Fig. 59 gives clearance data for marine bearings, horizontal bearings, vertical bearings, step bearings, Brown & Sharpe practice for over and under 600 r.p.m., respectively, the American Standards Association standards and British standards, up to a total clearance of 0.010 in. and diameters up to 8 in. Fig. 60 extends this range to larger clearances and diameters. The line marked 0.001 diam. + 0.002 shows a rule used by Kingsbury for standard duty journal bearings. In order that the other data may be interpreted with reference to the clearance ratio $\eta/a$, reference values of it are given from 0.0005 to 0.004. If clearances are chosen from these curves, they must for best results, be supplemented by a suitable oil.

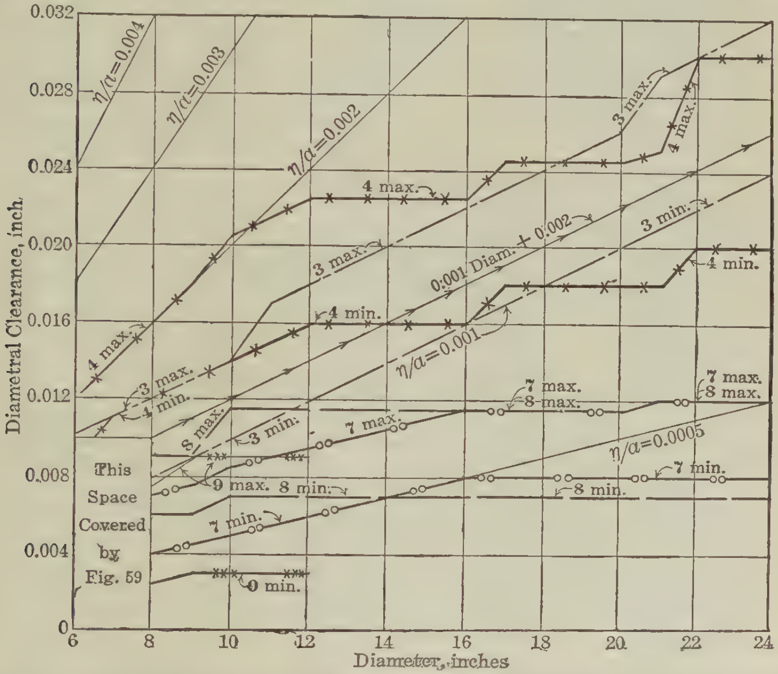


FIG. 60. Current Practice (1937) in Bearing Clearance, Shaft Diameter, 6 to 24 in. See Fig. 59 for Shaft Diameters 0 to 8 in., and key to curves.

## 5. THRUST BEARINGS

**THE THRUST PROBLEM.**—More or less axial load is encountered with every vertical shaft. With horizontal shafts, which are the most common, there usually is no real thrust load and only shaft shoulders are needed to hold the machine parts in proper axial relation.

**Heavy Horizontal Loads** are encountered in unbalanced centrifugal pumps and steam turbines, in marine propeller shafts, in industrial machines, and in some machine tools. When thrust loads are encountered at very low speed, the wear and the power lost in friction often are ignored and very inefficient bearings are used, although it is better practice to use rolling contact bearings in such service. With relatively high speeds and large shafts, as for marine propellers, the power loss is important, and efficient durable thrust bearings are used. For very high speeds with heavy thrust loads, as in steam turbines, the thrust bearing also must be efficient and durable.

**Light Horizontal Loads.**—Horizontal shafts, with little or no thrust load, often require close control of endwise position. This may be done by a collar on the shaft end as in Fig. 61. One collar in one bearing mounting then controls end-play. Using the shaft shoulder, as in Fig. 20, requires two bearing mountings. See Thrust Bearing Design and Construction below.

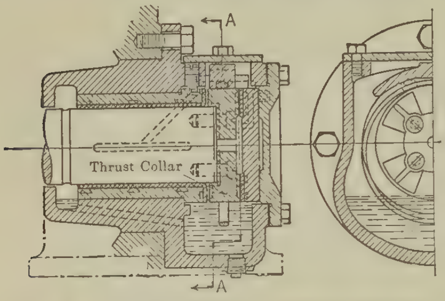


Fig. 61. Ring-oiled Journal and Thrust Bearing

**Vertical Loads.**—A vertical shaft in a machine always has a thrust load, due to the weight of the machine elements that in a horizontal shaft would be carried by journal bearings. However, a vertical shaft has no side loads on its journals except those produced by the operation of the machine itself. The thrust load on a vertical shaft either may be increased or reduced by loads produced by operation of the machine. Thus, in a hydroelectric generating unit, or a vertical centrifugal pump, hydraulic thrust in the runner increases the thrust load due to the weight of the revolving parts. The importance of the thrust problem depends on the combination of load and speed.

**THRUST BEARING DESIGN.**—Thrust bearings may be divided into two main groups, *viz.*, imperfectly lubricated and perfectly lubricated. The degrees of imperfection or perfection will vary. Among bearings that are perfectly lubricated, a bearing of one design may run with several times more friction than another, under otherwise identical conditions, and with no metallic contact of bearing faces during operation. Similarly, with imperfect lubrication, some bearings will run with higher friction than others, with the same oil, load and speed conditions, but with more or less metallic contact of the bearing faces always present.

**IMPERFECTLY LUBRICATED THRUST BEARINGS.**—Imperfect lubrication implies lack of provision for the functioning of adequate wedge-shaped films rather than lack of lubricant. See Perfect Lubrication, p. 13–44. For very low speed, heavy oils and hardness of bearing materials have been relied on to keep the rate of wear within reasonable limits. Such bearings should be placed in oil baths, if possible, so that lubricant may be continuously available. Grooving may be designed to keep oil from wasting, or to produce circulation that will continually change the oil on the bearing faces.

**Vertical Shafts.**—For this service the plain pivot, the multiple-disc-supported pivot, the plain ring thrust, the washer-supported ring thrust, Figs. 62 to 65, are used. The grooves in these bearings, if submerged in oil, should be carefully formed, usually in the upward faces, and with rounded edges, as in Fig. 66, in the harder surface. If oil is present in only minute quantities, grooves should be in the downward face. Further improvement

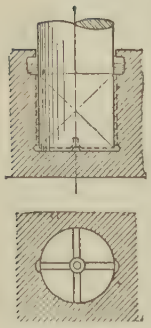


Fig. 62. Plain Pivot Thrust Bearing

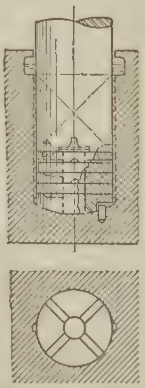


Fig. 63. Multiple-disc-supported Pivot Thrust Bearing

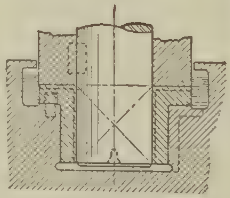


Fig. 64. Plain Ring Thrust Bearing

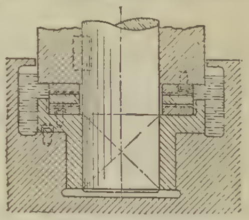


Fig. 65. Washer-supported Ring Thrust Bearing

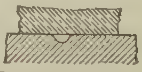


Fig. 66. Form of Oil Groove

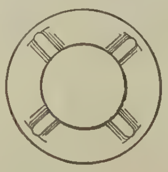


Fig. 67. Arrangement of Groove in Thrust Washer

can be made by slightly tapering the grooved bearing face for a distance at least equal to the width of the oil groove, on each side of the groove, but using not more than half the total surface for grooves and tapers. Two to four oil grooves usually are enough. For large diameters, or areas where radial width is small compared with the circumference, more grooves may be used. The size of the groove will vary, increasing with size of the bearing, sluggishness of the lubricant, and speed of rotation. If an oil bath cannot be used, it is best to stop the grooves short of the circumference of the disc or washer, and feed oil to the grooves from the center outward. See Fig. 67.

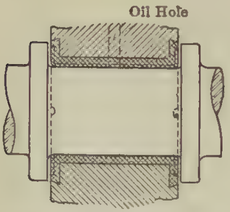


Fig. 68. Bearing to Take Light Thrust in Two Directions

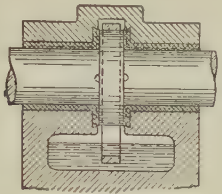


Fig. 69. Ring-oiled Two-direction Bearing for Light Thrust Load

**Horizontal Shafts.**—Conditions usually are quite different from those of vertical shafts. An oil bath is less easily provided. Also, because of end-play, it is difficult to prevent oil leaking away from the thrust surface, especially when thrust action is intermittent. For a simple construction like Fig. 68, thrust face grooves may be most easily cut in the stationary bearing face, although the harder face is preferable. Grooves should stop short of the outer diameter of the thrust collar. Sometimes a thrust collar, integral with or fastened to the shaft, is placed between two short journal bearings whose adjacent sides are enlarged to present reasonable thrust surfaces to the collar. If of sufficient diameter, this collar may be used to lubricate all bearing surfaces from a well below the shaft. Otherwise a ring may be used as in Fig. 69.

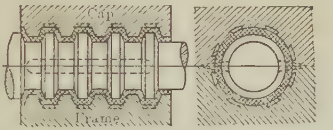


Fig. 70. Multiple-collar Bearing Sometimes Used in Woodworking Machinery to Position Spindle

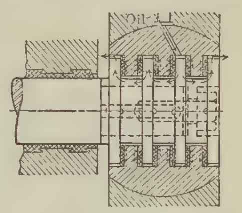


Fig. 71. Multi-collar Thrust Bearing for Light Loads

For data on friction coefficients of imperfectly lubricated thrust bearings, see p. 8–24. In such bearings the coefficient will, if bearing is adequately oiled, vary from 0.20 downward to about 0.02. This lower value would be considered very high for a perfectly lubricated bearing surface. See p. 8–28 for friction coefficients possible with perfect lubrication.

**Multi-collar Thrust Bearings** are used in several forms, more or less well made. In woodworking machinery, a bearing-metal lining is poured into the frame and cap, around a grooved shaft (Fig. 70). Oil is best applied at the sides. Such a bearing if lightly loaded, and not too tightly clamped on the shaft, may serve as a combined journal and thrust bearing, but it is not considered good practice. Sometimes the cap of such a bearing is made axially adjustable for control of end-play. Fig. 71 shows a multi-collar bearing used only to carry light thrust loads. It is adjacent to an adequate journal bearing. Oil under pressure may be applied as marked, and caused to flow through suitable grooves back and forth across the faces of the collars. If carefully made, and when oil grooves are properly designed with adjacent surfaces properly beveled, these bearings can carry moderate loads even at high speed.

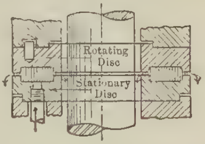


Fig. 72. Oil Pressure Thrust Bearing

One serious difficulty with all multi-collar thrust bearings is the inequality of distribution of load among the collars. To help overcome this in large bearings for marine service, crescent shaped elements, called horseshoes, with soft metal faces, are placed between the shaft collars. Sometimes grease is used for lubrication, but usually oil is supplied from a bath below the shaft into which the collars dip. The shoes sometimes are water-cooled by passages, but, for high-speed bearings, a water coil usually is placed in the bath below the collars. Horseshoes in marine practice are loaded to from 30 to 70 lb. per sq. in. They now are seldom used, because more efficient and more durable types of bearings are available, that operate with about one tenth the power loss. See pp. 13–36 to 13–39.

**THE OIL PRESSURE THRUST BEARING** employs two discs constructed as in Fig. 72. Oil is pumped under high pressure into the stationary disc at a definite rate, lifting the upper or rotating disc until oil discharge rate equals supply rate. Accumulators are required to protect against stoppage of the pumps. Rollers sometimes are provided to prevent contact of narrow faces of discs when oil supply fails. If the discs are permitted to rub they will "flash" and be damaged. Friction in the bearing itself is very low, but considerable power is required to drive the pumps. These bearings, therefore, are uneconomical, and are little used to-day.

**PERFECTLY LUBRICATED THRUST BEARINGS.**—See Perfect Lubrication, p. 13–44, for a discussion of the principles underlying the formation of the tapered oil film. Such films, in various forms, exist between the bearing faces of the thrust bearings described below.

**The Fixed Wedge Film** type of thrust bearing was suggested in 1888 by Prof. Archibald Barr, when discussing Beauchamp Tower's then recent tests on plain collar thrust bearings. It is obvious that part of the thrust bearing face must be flat to permit the collar to start and stop under load without undue concentration of pressure. The wedge-shape film in a bearing of this kind is unable to lift its load until a fair speed is attained. Conversely, it is apt to stop suddenly when slowing down. The fixed wedge may be combined with a flat surface in two ways: 1. Alternate bevels and flats with grooves *G* between, Fig. 73; 2. A supply groove *G* at the thick end of the bevel, omitting the groove between the bevel and the flat in the direction of motion, Fig. 74. This construction is used in the Gibbs thrust bearing, built by S. Morgan Smith Co., York, Pa., for both horizontal and vertical shafts. Bearing pressures range from 200 to 300 lb. per sq. in.

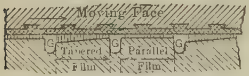
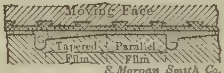
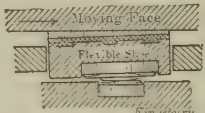


Fig. 73. Fixed Wedge-film Double Grooved Thrust Bearing



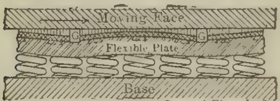
S. Morgan Smith Co.

Fig. 74. Fixed Wedge-film Gibbs Thrust Bearing



Kingsbury

Fig. 75. Free Wedge-film Thrust Bearing



General Electric Co.

Fig. 76. Reist Thrust Bearing

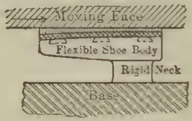
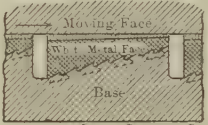


Fig. 77. Thrust Bearing with Flexible Shoe on Rigid Neck



Sperry Gyroscope Co.

Fig. 78. Restrained Wedge-film Bearing

A characteristic of these bearings is that the slope of the wedge remains constant for all conditions of load, speed, oil viscosity, and temperature for a given bearing. The film thickness, however, will vary automatically as these conditions change, provided the speed is great enough to separate the flat faces.

**The Free Wedge Film** (Fig. 75) type of thrust bearing (Kingsbury Thrust Bearing) was conceived, constructed and tested in 1896 by Prof. Albert Kingsbury. It is built by the Kingsbury Machine Works, Inc., Philadelphia. A parallel, but independent, development of the similar Michell thrust bearing has taken place in England. The bearing members (shoes) in this type are tiltably mounted, and are free to form automatically the most efficient natural oil film required by each different combination of load, speed, viscosity and temperature to which the bearing is subjected.

**The Restrained Wedge Film** type of thrust bearing appears in several forms. In this type the elements forming the films are not fully free, but do respond, in a limited degree, to change of load, speed, viscosity and temperature. One of these, Fig. 76, invented by H. G. Reist about 1915, is built by the General Electric Co. Its stationary element is a babbitted, relatively thin, steel ring, with one radial gap, mounted on many small stiff precompressed springs. Radial grooves are cut in both of the opposing bearing faces to supply cool oil to the bearing faces. Wedge film formation is restrained by the stiffness inherent in the flexible babbitted plate, which is finished flat. During operation, a film of oil thick enough to prevent metallic contact is maintained between the rotating and stationary plates. The film is not sensitive to changes of speed or loading, nor will it be disrupted by a small amount of sludge or small particles. Both stationary and rotating grooves are chamfered to direct the oil between the surfaces, and the film forms when half the distance between grooves has been traversed. The flexibility of the structure insures the formation of a film of the most effective shape. In the event of misalignment, the flexible support automatically deflects to equalize the pressure distribution over the entire bearing. The operation of the bearing depends on the combined effects of the grooved rotating and stationary plates and of the flexible support.

Fig. 77 shows a second form of restrained-wedge-film bearing, inventor unknown. Restraint of wedge film formation is imposed by the rigid neck and the stiff portions of the adjacent shoe body. This is an integral structure. The bearing faces of the shoe bodies are finished flat. The flexing of the overhung portion takes place under the pressures incident to the formation of the film.

A third and rather unusual restrained wedge film, invented by Elmer D. Sperry, is shown in Fig. 78. The supporting base is faced with bearing metal of thickness that varies considerably. Babbitt metal has a coefficient of elasticity much greater than iron or steel. Hence if the thickness is greater where the pressures are lower a wedge shaped film may be formed, though considerably restrained.

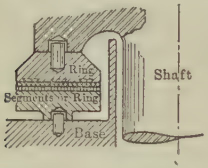


Fig. 79. Radial Section of Thrust Bearing Indicating Radial Tilting

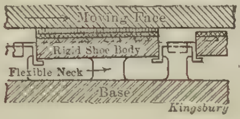


Fig. 80. Thrust Bearing Shoe on Flexible Neck

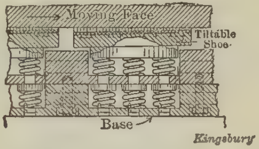


Fig. 81. Thrust Bearing with Duplex Spring-supported Shoes

Radial tiltability may be given to relatively thick bearing rings by applying the load to their backs through annular supporting ribs. See Fig. 79. Also the lower ring may be made in separate segments that have freedom only for radial tilting, in which case the ribs supporting the segments may be tangential. Such elements may have fixed bevel bearing faces or flexible leading portions, as already described.

**The Semi-free Wedge Film** is formed when the tilting shoe is mounted on a flexible neck, Fig. 80. A fairly efficient form of thrust bearing of simple integral structure results. A certain amount of freedom must be sacrificed to preserve the requisite ruggedness.

Another form in which this semi-free film appears is that in which separate shoes are mounted on springs or other elastic means. See Fig. 81. This particular design contains other features important for resiliently supported bearings, *i.e.*, means for preventing and for damping vibration, without interference with the normal functioning of the bearing under variable load.

**Film Forms.**—When the relative motion is circular, instead of straight-line, the flat faced wedge film is not geometrically a flat wedge in the developed section of the path of relative motion, but is a sine curve wedge, as in journal bearings. There also are changes in film shape due to elasticity of materials, and to temperature variations. Thrust bearing films, therefore, usually are of a composite nature. Occasionally a composite form is purposely given the film, as when beveling is added to the leading or trailing portions of the bearing face of the freely tilting shoe. A variable composite film appears in the Reist bearing, Fig. 76, in which both bearing faces are provided with oil grooves. The double grooving tends alternately to build up and to break down the pressure in the films, but it helps the radial flow of oil across the bearing faces.

**CONSTRUCTION DETAILS.—Reversible Bearings.**—In general a thrust bearing may be made suitable for both directions of rotation, *i.e.*, reversible, by making its supporting feature symmetrical with respect to the bearing face segments or grooves. For example, the free-wedge type may have its shoes supported in the middle, and it loses little efficiency thereby. The fixed-wedge type requires the provision of an additional beveled portion, tapered in the opposite direction. When the limit of dead load supportable by the flat parallel portion of the bearing face is reached, the reversible fixed-wedge bearing area would have to be larger than if made for one direction only.

**Aligning and Equalizing Means for Thrust Bearings.**—When the machine is so accurately made that alignment of shafting is perfect within the limits of the film thicknesses in the bearings and the elasticity of the component parts, no aligning means need be provided. This also is true when the bearings are so placed with respect to the machine frame that the slight elasticity of the frame itself enables the bearings to preserve their proper alignment. However, it often is desirable to provide some aligning means within the thrust bearing, and the methods cited below are employed.

The design of aligning means should be governed by the misalignment to be corrected, as initial, subsequent, or continuous or rapidly recurring. The durability of the aligning means, therefore, is an important consideration in the selection of a thrust bearing. Special forms have been devised for such service as gyro-stabilizers and for thrust collars that are out of square with the shaft.

1. **The Spherical Washer,** Fig. 83, is the commonest form and has several advantages, one of which is ruggedness. When used with three freely tilting shoes it tends to equalize the load distribution among them. For best efficiency, the mean angle of the spherical faces should be about 45 deg. Even so, when once well loaded, an eccentricity of the load equal to more than half the projected radius to the mean zone of the spherical fit, is required before slipping will take place. Hence, this method of equalizing or aligning is not sensitively responsive to variations in alignment under load.

2. **Adjustable Bearings.**—When freely tilting shoes are mounted on jack screws, the latter may be used for initial setting of height, alignment and equalization of load, but are not useful for preserving alignment when the supporting structure shifts.

3. **Elastic Support** is an approximate aligning means, and also an approximate equalizing means for various portions of a bearing when little correction need be made. With the stiff springs necessarily employed, carefully preliminary setting of the thrust deck is essential. Means to prevent or control vibration must be employed. See discussion of Figs. 76 and 81.

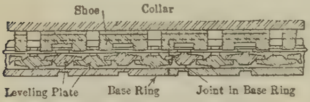


FIG. 82. Equalizing Supports of Kingsbury Thrust Bearing

4. **Equalizing Supports.**—Systems of rocking levers are used effectively in Kingsbury bearings for alignment and equalization. They continue effective, even in the presence of considerable subsequent displacement of the thrust deck. Their range and accuracy are considerable. See Fig. 82. This form has been adapted, with roller contacts, to very large hydroelectric bearings.

**DATA ON STANDARD KINGSBURY THRUST BEARINGS.**—Because of the relatively recent development of these oil film bearings their standardization is not as widely known as that of ball and roller bearings. It has, therefore, been thought desirable to place herein the principal dimensions of one important generally useful line of Kingsbury bearings and to illustrate a number of types of mountings in which they are used.

**Internal Parts.**—The self-aligning equalizing types of Kingsbury thrust bearings consist mainly of the shoes and their supporting structures shown in Figs. 83 and 84. The overall dimensions of these 3-shoe and 6-shoe elements are made the same for interchangeability. See Table 10. These elements may be used either horizontally or vertically, but usually they rely on a ring or on a surrounding wall of the mounting to hold the shoes radially in place.

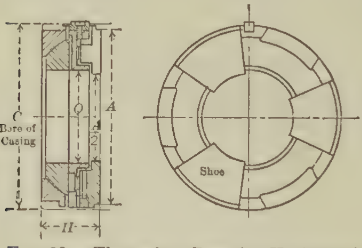


FIG. 83. Three-shoe Base for Horizontal or Vertical Thrust Bearing

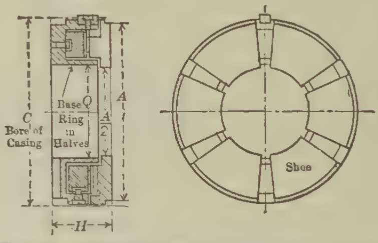


FIG. 84. Six-shoe Base for Horizontal or Vertical Thrust Bearing

Commonly-used combinations of these elements are: 1. Horizontal 3-shoe base and 6-shoe base with thrust collar between, a combination useful for thrust loads that are heavier in one direction than the other; 2. Single horizontal bearing for one-way thrust with 6-shoe base and separate collar; 3. Horizontal double 6-shoe bearing for two-way thrust with separate collar (see Fig. 85); 4. Horizontal double 6-shoe bearing with collar integral with shaft (See Fig. 86). In this construction the 6-shoe bases are made in halves; 5. Horizontal double two-way bearings with bases in halves, with one 2-shoe tilting ring type base, and one 6-shoe base, the major thrust being taken by the 6-shoe bearing; 6. Vertical 3-shoe thrust bearing, spherically seated base (See Fig. 83), and runner; 7. Vertical 6-shoe vertical bearing and runner; 8. Vertical 3-shoe spherically seated bearing with spherically seated shoe-retaining base, and runner (See Fig. 87); 9. Vertical 6-shoe thrust bearing with shoe retaining base and runner (see Fig. 88); 10. Vertical 6-shoe thrust bearing with runner and shoe-retaining straps.

**THRUST BEARING CAPACITIES.**—Table 10 is based on the practice of Kingsbury Machine Works for average conditions in bearings with freely tilting shoes of proportions shown in Table 11. For similar information on other bearing types, consult the manufacturers concerned. Capacity depends on mechanical strength, materials used, surface speed, oil viscosity, and temperature. It may be increased with speed until a limit is reached, imposed by other considerations, such as starting and stopping conditions. It is assumed that safety while running depends on the maintenance of the oil film above some minimum thickness that experience has proved to be adequate. Any capacity table must, therefore, have its limitations stated, because bearings can be designed and operated under vastly greater loads than indicated in Table 10, which applies to thrust bearings as usually applied to steam turbines, centrifugal pumps, ordinary industrial machinery, marine propulsion, and hydroelectric units.

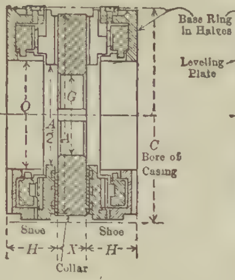


FIG. 85. Double Thrust Bearing with 6-Shoe Bases and Separate Collar

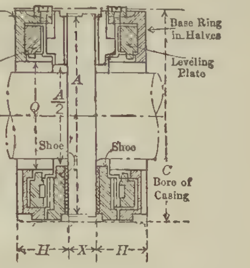


FIG. 86. Double Horizontal Thrust Bearing with 6-Shoe Bases and Integral Collar

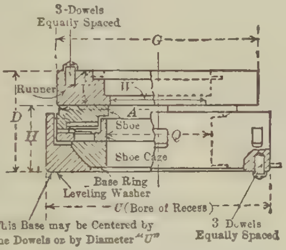


FIG. 87. Three-shoe Vertical Thrust Bearing with Runner and Shoe-retaining Base

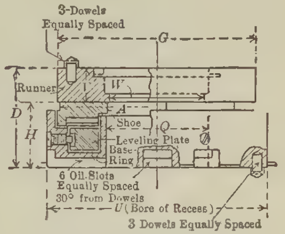


FIG. 88. Six-shoe Vertical Thrust Bearing with Runner and Shoe-retaining Base

**LOADING OF THRUST BEARINGS WITH PERFECT LUBRICATION.**—See also Viscous Flow, p. 13–44, and Perfect Lubrication, p. 13–44. Of the various possible film forms, one is the *optimum for friction* and another the optimum for capacity. Minimum

**Table 9.—Dimensions of Standard Three-shoe and Six-shoe Thrust Bearings**

(Kingsbury Machine Works, Inc., Philadelphia)

See Figs. 85 to 88

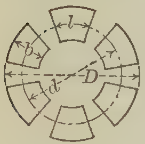
| Nominal Size, *A* | Net Area, sq. in. | | *B* (Bore) | Dimensions, inches | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 6 shoes | 3 shoes | | *C* | *D* | *G* | *H* | *Q* | *U* | *V* | *W* | *X* |
| 5 | 12.5 | 6.3 | 1.750 | 5.375 | 2 5/8 | 5 1/8 | 1 3/4 | 2 3/4 | 6.000 | 7/8 | 2 3/8 | 7/8 |
| 6 | 18.0 | 9.0 | 2.125 | 6.375 | 3 1/16 | 6 1/8 | 2 1/16 | 3 1/4 | 7.000 | 1 | 2 7/8 | 1 |
| 7 | 24.5 | 12.3 | 2.500 | 7.375 | 3 5/8 | 7 1/8 | 2 3/8 | 3 3/4 | 8.000 | 1 1/4 | 3 3/8 | 1 1/4 |
| 8 | 32.0 | 16.0 | 3.000 | 8.375 | 4 1/16 | 8 1/8 | 2 11/16 | 4 5/16 | 9.125 | 1 3/8 | 3 7/8 | 1 3/8 |
| 9 | 40.5 | 20.3 | 3.500 | 9.375 | 4 1/2 | 9 1/8 | 3 | 4 7/8 | 10.125 | 1 1/2 | 4 3/8 | 1 1/2 |
| 10 1/2 | 55.1 | 27.6 | 4.125 | 11.000 | 5 1/8 | 10 11/16 | 3 3/8 | 5 11/16 | 11.750 | 1 3/4 | 5 1/16 | 1 3/4 |
| 12 | 72.0 | 36.0 | 4.750 | 12.500 | 5 3/4 | 12 3/16 | 3 3/4 | 6 1/2 | 13.375 | 2 | 5 13/16 | 2 |
| 13 1/2 | 91.1 | 45.6 | 5.375 | 14.000 | 6 1/2 | 13 11/16 | 4 1/4 | 7 5/16 | 14.875 | 2 1/4 | 6 9/16 | 2 1/4 |
| 15 | 112.5 | 56.3 | 6.000 | 15.500 | 7 1/8 | 15 3/16 | 4 5/8 | 8 1/8 | 16.500 | 2 1/2 | 7 5/16 | 2 1/2 |
| 17 | 144.5 | 72.3 | 6.625 | 17.625 | 8 | 17 1/4 | 5 1/4 | 9 3/16 | 18.625 | 2 3/4 | 8 1/4 | 2 7/8 |
| 19 | 180.0 | ...... | 7.500 | 20.250 | 7 3/4 | 19 1/4 | 4 3/4 | 10 5/8 | ...... | 3 | 9 1/4 | 3 1/4 |
| 21 | 220.0 | ...... | 8.500 | 22.250 | 8 1/4 | 21 1/4 | 5 1/4 | 11 3/4 | ...... | 3 | 10 1/4 | 3 5/8 |
| 23 | 264.0 | ...... | 9.375 | 24.500 | 8 11/16 | 23 1/4 | 5 11/16 | 12 3/4 | ...... | 3 | 11 1/4 | 3 7/8 |
| 25 | 312.0 | ...... | 10.000 | 26.500 | 9 9/16 | 25 1/4 | 6 3/16 | 14 | ...... | 3 3/8 | 12 1/4 | 4 1/4 |
| 27 | 364.0 | ...... | 11.000 | 28.750 | 10 1/16 | 27 1/4 | 6 11/16 | 15 | ...... | 3 3/8 | 13 1/4 | 4 5/8 |
| 29 | 420.0 | ...... | 11.750 | 30.750 | 10 1/2 | 29 3/8 | 7 1/8 | 16 1/4 | ...... | 3 3/8 | 14 1/8 | 5 |
| 31 | 480.0 | ...... | 12.500 | 33.000 | 11 | 31 3/8 | 7 5/8 | 17 1/4 | ...... | 3 3/8 | 15 1/8 | 5 1/4 |
| 33 | 545.0 | ...... | 13.375 | 35.000 | 11 7/8 | 33 3/8 | 8 1/8 | 18 1/2 | ...... | 3 3/4 | 16 1/8 | 5 5/8 |
| 37 | 684.0 | ...... | 15.000 | 40.000 | 12 3/4 | 37 1/2 | 9 | 20 3/4 | ...... | 3 3/4 | 18 | 6 3/8 |
| 41 | 840.0 | ...... | 16.750 | 43.750 | 14 | 41 1/2 | 10 | 23 | ...... | 4 | 20 | 7 |
| 45 | 1012.0 | ...... | 18.500 | 48.000 | 15 1/8 | 45 1/2 | 11 | 25 | ...... | 4 1/8 | 22 | 7 5/8 |

## Table 10.—Capacities of 6-Shoe Standard Duty Horizontal and Vertical Thrust Bearings

(Kingsbury Machine Works, Inc., Philadelphia, Pa.)

Based on viscosity of 150 sec. Saybolt at operating temperatures. Capacities given may be increased from 10% to 25%, if viscosity is increased in same proportion. Maximum operating pressure, 400 to 500 lb. per sq. in. of segment area. Over and above these bearings have a factor of safety of about 4.

| Bearing Size | Area, sq. in. | Revolutions per Minute | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 70 | 100 | 150 | 200 | 300 | 400 | 500 | 700 | 800 | 900 | 1200 | 1800 | 2500 | 3600 |
| | | Safe Load, Thousands of Pounds | | | | | | | | | | | | | |
| 5 | 12.5 | ..... | 1.44 | ..... | 1.7 | ..... | 2.0 | ..... | ..... | 2.4 | ..... | 2.6 | 2.9 | 3.2 | 3.5 |
| 6 | 18.0 | ..... | 2.30 | ..... | 2.7 | ..... | 3.2 | ..... | ..... | 3.8 | ..... | 4.2 | 4.6 | 5.0 | 5.5 |
| 7 | 24.5 | ..... | 3.30 | ..... | 3.9 | ..... | 4.7 | ..... | ..... | 5.6 | ..... | 6.2 | 6.8 | 7.4 | 8.0 |
| 8 | 32.0 | ..... | 4.60 | ..... | 5.5 | ..... | 6.6 | ..... | ..... | 7.8 | ..... | 8.6 | 9.6 | 10.4 | 11.4 |
| 9 | 40.5 | ..... | 6.20 | ..... | 7.4 | ..... | 8.8 | ..... | ..... | 10.4 | ..... | 11.4 | 13.0 | 14.0 | 15.0 |
| 10½ | 55.1 | ..... | 9.20 | ..... | 10.8 | ..... | 13.0 | ..... | ..... | 15.4 | ..... | 17.0 | 19.0 | 21.0 | 22.0 |
| 12 | 72.0 | ..... | 12.80 | ..... | 15.2 | ..... | 18.0 | ..... | ..... | 21.0 | ..... | 24.0 | 26.0 | 28.0 | 29.0 |
| 13½ | 91.1 | ..... | 17.20 | ..... | 20.0 | ..... | 24.0 | ..... | ..... | 29.0 | ..... | 32.0 | 35.0 | 36.0 | 36.0 |
| 15 | 112.5 | ..... | 22.00 | ..... | 26.0 | ..... | 32.0 | ..... | ..... | 37.0 | ..... | 41.0 | 45.0 | 45.0 | .... |
| 17 | 144.5 | ..... | 30.00 | ..... | 36.0 | ..... | 43.0 | ..... | ..... | 51.0 | ..... | 57.0 | 58.0 | 58.0 | .... |
| 19 | 180 | 37.0 | 40.00 | 44.0 | 48.0 | 53.0 | ..... | 60.0 | 65.0 | ..... | 69.0 | ..... | ..... | ..... | .... |
| 21 | 220 | 47.0 | 51.00 | 57.0 | 61.0 | 68.0 | ..... | 77.0 | 84.0 | ..... | 89.0 | ..... | ..... | ..... | .... |
| 23 | 264 | 59.0 | 65.00 | 72.0 | 77.0 | 85.0 | ..... | 97.0 | 105.0 | ..... | 106.0 | ..... | ..... | ..... | .... |
| 25 | 312 | 73.0 | 80.00 | 88.0 | 95.0 | 105.0 | ..... | 119.0 | 123.0 | ..... | 123.0 | ..... | ..... | ..... | .... |
| 27 | 364 | 88.0 | 97.00 | 107.0 | 115.0 | 127.0 | ..... | 144.0 | 146.0 | ..... | 146.0 | ..... | ..... | ..... | .... |
| 29 | 420 | 106.0 | 116.00 | 128.0 | 137.0 | 152.0 | ..... | 168.0 | 168.0 | ..... | ..... | ..... | ..... | ..... | .... |
| 31 | 480 | 125.0 | 137.00 | 151.0 | 162.0 | 180.0 | ..... | 192.0 | 192.0 | ..... | ..... | ..... | ..... | ..... | .... |
| 33 | 544 | 146.0 | 160.00 | 177.0 | 189.0 | 210.0 | ..... | 220.0 | 220.0 | ..... | ..... | ..... | ..... | ..... | .... |
| 37 | 684 | 195.0 | 215.00 | 235.0 | 250.0 | 275.0 | ..... | 275.0 | ..... | ..... | ..... | ..... | ..... | ..... | .... |
| 41 | 840 | 250.0 | 275.00 | 305.0 | 325.0 | 335.0 | ..... | 335.0 | ..... | ..... | ..... | ..... | ..... | ..... | .... |
| 45 | 1012 | 315.0 | 345.00 | 385.0 | 405.0 | 405.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .... |



## Table 11.—Thrust Bearing Capacities and Minimum Film Thicknesses, Freely Pivoted Shoes

(Kingsbury Machine Works, Inc., Philadelphia)

For average proportions of thrust bearings $d = 3b$. For any ratio $b/d$, if $l = b$, bearing area $= 6\,l^2$ for 6 shoes, and total capacity will be (tabular value × No. of shoes). Table is based on $l/b = 1$. For other proportions multiply tabular value by correction factor below.

| Face Area, sq. in. | | Surface Speed, Feet per Second, $= U/12 = \pi dN/(12 \times 60)$ | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 5 | 10 | 20 | 40 | 60 | 80 | 100 | 120 | 140 |
| | | Capacity, Pounds per Square Face | | | | | | | | |
| 4 | Capacity, lb...... | 855 | 1015 | 1210 | 1440 | 1590 | 1710 | 1810 | 1890 | 1960 |
| | Film Thickness, in. | 0.00036 | 0.00047 | 0.00061 | 0.00079 | 0.00092 | 0.00102 | 0.00112 | 0.00119 | 0.00126 |
| 16 | Capacity, lb...... | 4060 | 4830 | 5750 | 6850 | 7550 | 8150 | 8600 | 9000 | 9350 |
| | Film Thickness, in. | .00047 | .00061 | .00079 | .00103 | .00119 | .00132 | .00145 | .00155 | .00163 |
| 36 | Capacity, lb...... | 10,100 | 12,000 | 14,300 | 17,000 | 18,750 | 20,200 | 21,400 | 22,400 | 23,200 |
| | Film Thickness, in. | .00055 | .00071 | .00092 | .00119 | .00138 | .00154 | .00168 | .00180 | .00192 |
| 64 | Capacity, lb...... | 19,300 | 23,000 | 27,400 | 32,500 | 36,000 | 38,600 | 40,900 | 42,800 | 44,500 |
| | Film Thickness, in. | .00061 | .00079 | .00102 | .00133 | .00155 | .00172 | .00188 | .0020 | .00215 |
| 100 | Capacity, lb...... | 31,900 | 38,000 | 45,200 | 53,500 | 59,500 | 64,000 | 67,500 | 70,700 | 73,500 |
| | Film Thickness, in. | .00066 | .00086 | .00112 | .00145 | .00168 | .00186 | .00205 | .0022 | .00235 |
| 144 | Capacity, lb...... | 48,000 | 57,200 | 68,000 | 81,000 | 89,500 | 96,500 | 102,000 | 106,500 | 111,000 |
| | Film Thickness, in. | .00071 | .00092 | .00119 | .00155 | .00180 | .0020 | .0022 | .00235 | .0025 |
| 196 | Capacity, lb...... | 68,000 | 81,000 | 96,000 | 114,500 | 126,500 | 136,000 | 144,000 | 150,000 | 156,000 |
| | Film Thickness, in. | .00075 | .00098 | .00127 | .00163 | .00190 | .0021 | .0023 | .0025 | .00265 |
| 256 | Capacity, lb...... | 92,000 | 109,000 | 130,000 | 155,000 | 171,000 | 184,000 | 194,000 | 203,000 | 211.000 |
| | Film Thickness, in. | .00079 | .00103 | .00133 | .00172 | .0020 | .0022 | .00245 | .0026 | .0028 |
| 324 | Capacity, lb...... | 119,000 | 142,000 | 168,000 | 201,000 | 222,000 | 239,000 | 252,000 | 264,000 | 274,000 |
| | Film Thickness, in. | .00083 | .00108 | .00138 | .00180 | .0021 | .0023 | .00255 | .0027 | .0029 |
| 400 | Capacity, lb...... | 152,000 | 181,000 | 215,000 | 256,000 | 283,000 | 304,000 | 321,000 | 336,000 | 350,000 |
| | Film Thickness, in. | .00086 | .00112 | .00145 | .00188 | .0022 | .0024 | .00265 | .0028 | .0030 |

Load Correction Factors

| $l/b$ = | 0.50 | 0.67 | 1.00 | 1.33 | 1.50 | 2.00 |
|---|---|---|---|---|---|---|
| Correction Factor = | 1.557 | 1.357 | 1.000 | 0.732 | 0.630 | 0.423 |

film thickness is a criterion of bearing safety. Table 11, based on the experience of Kingsbury is a basic loading table that shows minimum film thicknesses to be expected under given conditions for average thrust bearings of steam turbines, centrifugal pumps, ordinary industrial machinery, hydro-electric units, and marine propulsion, where the oil is not too heavy. Values in the table are based on a mean viscosity coefficient, within the film at operating temperature, of $3.4 \times 10^{-6}$ Reyn = 0.23 poise, equivalent to 125 sec. Saybolt Universal viscosity with oil of 0.875 specific gravity. Film thickness is taken on the median line of the trailing edge of the shoe. Load correction factors are based on a study of side leakage. See p. 13–16.

**FRICTION IN THRUST BEARINGS.**—See *Mech. Engg.*, April 1927, p. 375. The friction coefficient for a perfectly lubricated surface moving in a straight line on tiltable rectangular shoes (approximately the condition in Kingsbury and Michell bearings) is given by

$$\lambda = Fb/R = 1.81\,k\sqrt{\mu v / l p_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [13]$$

where $k$ is a factor given by Fig. 89 based on Michell's data (9). Other notation is as given on p. 13–15.

**POWER LOSSES IN THRUST BEARINGS.**—Table 12 gives the relations of speed, capacity and friction of Kingsbury thrust bearings when the mean viscosity of oil in the films is $3.4 \times 10^{-6}$ Reyn, the ratio of $l/b$ of the shoes is 4/3, and the outer diameter $D$, in., of the bearing is $4\,d/3$, where $d$ = mean diameter. For a 6-shoe bearing the net area of segments is $^1/_2 D^2$. Friction coefficients and power losses are based on a value of side leakage correction factor of $k$ = 1.6. Corrections can be made for other conditions by application of formula [13]. For similar data on thrust bearings of other types, the several bearing manufacturers should be consulted.

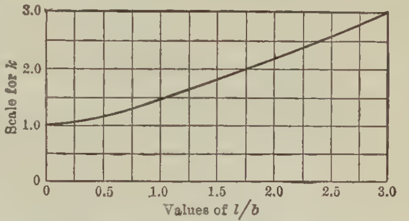


Fig. 89. Friction Coefficient Factor $k$, Interpreted from Michell's Analysis of Flat Bearing Surfaces

**THRUST BEARING MOUNTINGS.**—The following paragraphs describe typical methods of mounting thrust bearings.

**Combined Vertical Thrust and Guide Bearings.**—The combined flat thrust and cylindrical guide bearing, Fig. 90, usually has three or six thrust shoes $T$, the thrust runner $R$ being submerged. Oil, lifted by centrifugal action, fills the space around the pivoted guide bearing shoe $G$ and at the

**Table 12.—Capacity, Power Loss and Coefficient of Friction for Kingsbury Thrust Bearings**
(6 shoes, 51 deg. subtended angle, viscosity = $3.4 \times 10^{-6}$ Reyn)

| Capacity and Power Loss | | | Coefficient of Friction and Correction Factor for Load | | | Correction Factor for Load* | |
|---|---|---|---|---|---|---|---|
| Outer Diameter of Bearing, in. | 350 lb. per sq. in. Pressure; Shoe Bore = $^1/_2$ Outside Diam. | | Speed, r.p.m. | Speed Factor for Power Loss* | Coefficient of Friction at 350 lb. per sq. in. Pressure | Pressure, lb. per sq. in. | Correction Factor for Coefficient of Friction* |
| | Capacity, lb. | Power Loss, Hp. at 100 r.p.m. | | | | | |
| 5 | 4,380 | 0.0128 | 50 | 0.354 | 0.00069 | 100 | 0.63 |
| 6 | 6,300 | .0220 | 60 | .465 | .00076 | 175 | .71 |
| 7 | 8,600 | .0350 | 80 | .716 | .00088 | 250 | .85 |
| 8 | 11,200 | .0522 | 100 | 1.00 | .00098 | 300 | .93 |
| 9 | 14,200 | .0744 | 120 | 1.32 | .0011 | 350 | 1.00 |
| $10\,^1/_2$ | 19,300 | .118 | 160 | 2.02 | .0012 | 400 | 1.07 |
| 12 | 25,200 | .176 | 200 | 2.83 | .0014 | 500 | 1.19 |
| 15 | 39,500 | .344 | 250 | 3.95 | .0016 | ......... | .......... |
| 17 | 50,800 | .501 | 300 | 5.20 | .0017 | ......... | .......... |
| 21 | 77,300 | .944 | 400 | 8.00 | .0020 | ......... | .......... |
| 25 | 109,500 | 1.59 | 500 | 11.2 | .0022 | ......... | .......... |
| 29 | 147,500 | 2.49 | 600 | 14.7 | .0024 | ......... | .......... |
| 33 | 191,000 | 3.66 | 800 | 22.6 | .0028 | ......... | .......... |
| 37 | 240,000 | 5.17 | 1000 | 31.6 | .0031 | ......... | .......... |
| 45 | 355,000 | 9.30 | 1250 | 44.2 | .0035 | ......... | .......... |
| 53 | 493,000 | 15.2 | 1500 | 58.1 | .0038 | ......... | .......... |
| 61 | 652,000 | 23.2 | 2000 | 89.4 | .0044 | ......... | .......... |
| 73 | 935,000 | 39.7 | 3000 | 164.0 | .0054 | ......... | .......... |

* Multiply the power losses in third column by these factors to correct for speed and load.

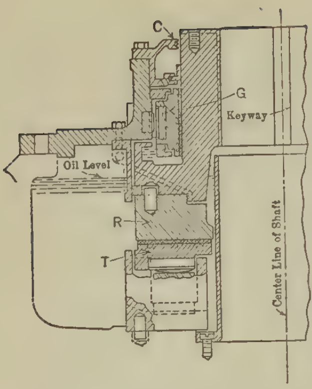


FIG. 90. Combined Vertical Guide and Thrust Bearing

top flows back to the oil bath. The upper special seal ring *C* prevents aeration. Load on thrust shoes is equalized as in the bearing Fig. 82. A cooling coil is used if required.

The self-aligning submerged guide bearing Fig. 91, has guide shoes similar to those in Fig. 90, but the mounting and oil well are so designed as to submerge the shoes. The arrangement shown is suitable for moderate speeds. For high speeds, the bearing is arranged to circulate oil from the bottom of the oil well and to keep the shoe cavity under moderate oil pressure to prevent foaming. In an alternative arrangement, Fig. 92, the guide shoes *G* bear directly against the shaft. Oil is lifted to the bearing by a centrifugal or viscosity pumping device on the shaft. The shoes dip in oil retained in the basin of the pumping device and thereby have capillary oiling for starting.

**SPHERICAL THRUST BEARING,** Fig. 93, performs in one unit the functions of both the flat thrust and the guide bearing, Fig. 90. Six pivoted shoes *T* are seated in the solid base ring *B*, the shoe faces being concave against the spherical-faced runner *R*. The bearing will sustain a radial load of about 15% of the downward load, and unlike a cylindrical guide bearing, it requires no radial running clearance. The shaft, therefore, will run accurately centered. For high speeds, an air seal ring *C* around the runner minimizes foaming. A narrow bronze bushing *S*, surrounding, and loose on, the thrust block hub *H* holds the parts together in handling. For temporary radial overloads, or lifting of the shaft, an auxiliary guide bearing may be applied above the spherical thrust bearing, using the same guide parts as in Fig. 90. The guide bearing shoes then are bored with a larger running clearance, and act only if the shaft lifts. Four 30-in. bearings of this type, operating at 500 r.p.m. are used in condensers at the Leaside Station, Toronto, Ont.

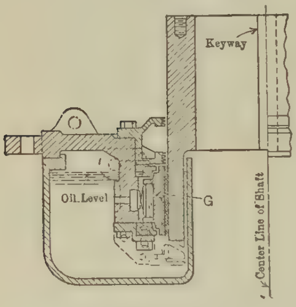


FIG. 91. Self-aligning Submerged Type Combination Guide and Thrust Bearing

**MARINE TYPE AUTOMATICALLY LUBRICATED THRUST MOUNTING,** Fig. 94, is commonly used with Kingsbury thrust bearings abaft ship propulsion machinery, where the shaft diameter is large and the full area of the thrust collar is not required to carry the load. Two thrust shoes in each end of the mounting are adjusted by jack screws and hardened thrust pins. The adjacent journal bearing has a removable split shell. End closures commonly are of non-wearing types. See Figs. 21 and 22. Oil lifted by the thrust collar is directed to the bearing surfaces.

The disc-oiled propeller shaft bearing Fig. 95 has a short self-aligning bearing shell and an oiling disc *D* operating on the viscosity pumping principle. The disc delivers oil to a hopper *H* in the cap of the bearing shell. A crown ring closure (Fig. 22) prevents leakage at the housing opening.

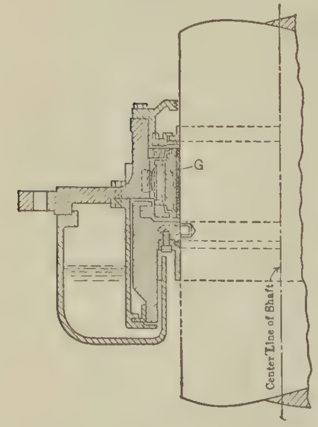


FIG. 92. Self-aligning Guide and Thrust Bearing with Shoes Bearing on Shaft

**HORIZONTAL MOUNTINGS FOR HIGH-SPEED COMBINED THRUST AND JOURNAL BEARINGS,** Fig. 96, are used in large high-speed centrifugal pumps. The double thrust elements have two, three or six shoes, depending on the load. Oil circulation is maintained by a viscosity pump 5 on the periphery of the thrust collar, and is automatically reversible with change in direction of rotation. With rotation in the direction of the arrow, oil is drawn from reservoir at *C*, flows through passages *B* toward the shaft, and thence outward between the shoes. It passes to the cooler through space *D* and passage *E*. Holes *A* are used for rotation in opposite direction. Passage *G* delivers oil to the journal. The

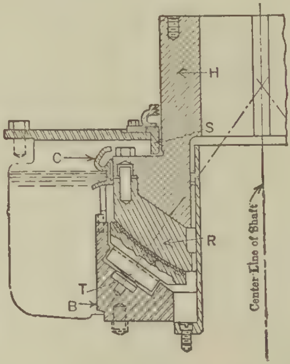


FIG. 93. Spherical Thrust Bearing

mounting may be capped at *L* for an oil supply to a separate journal bearing, with return at end.

**THRUST BEARINGS FOR HYDROELECTRIC UNITS.**—The field in which the largest thrust bearings are required is that of hydroelectric units of the vertical type. The turbines for large hydroelectric units usually are made by one manufacturer and the generators by another. Thrust bearings usually are supplied with the generators. Examples of practice in thrust bearing application follow.

**Boulder Dam.**—Two General Electric generators, operating at 180 r.p.m., are carried on Kingsbury equalizing thrust bearings 85 1/2 in. diameter. The thrust load is 1,800,000 lb. Two other General Electric generators, also operating at 180 r.p.m., are carried on General Electric thrust bearings (Fig. 76). The thrust load is 1,786,000 lb. An Allis-Chalmers generator operating at 257 r.p.m., with a thrust load of 680,000 lb. is carried on a Kingsbury adjustable-type thrust bearing 57 in. diam. Two Westinghouse generators are carried on 84-in. Kingsbury guide-thrust bearings with eight babbitt-lined shoes, each supported by separate adjusting screws which permit alignment and equalization of load. The thrust load is 1,700,000 lb.

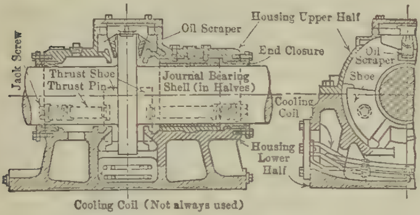


FIG. 94. Marine Type Automatically Lubricated Thrust Bearing

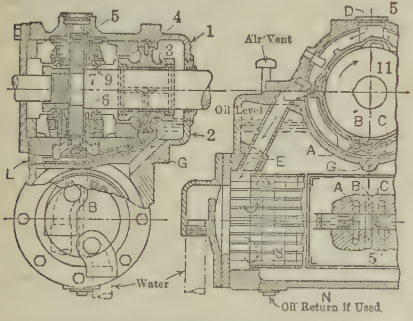


FIG. 96. Horizontal High-speed Thrust Bearing Mounting

1. Housing Upper Half
2. Housing Lower Half
3. Bearing Shell (in halves)
4. Bearing Shell Dowel
5. Pumping Ring
6. Thrust Shoe
7. Collar
8. Oil Cooler
9. Seal Ring

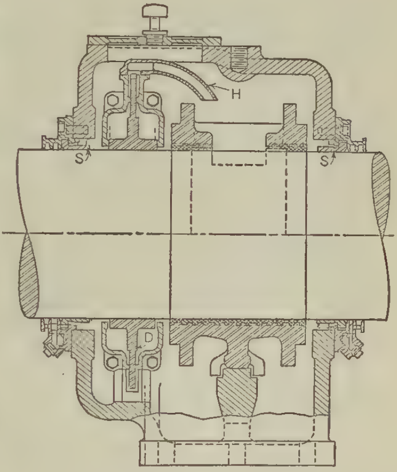


FIG. 95. Disc-oiled Propeller Shaft Bearing

**McIndoes Plant,** New England Power Construction Co., has four Gibbs bearings, on vertical hydroelectric generators of 3500 and 5120 Hp., respectively, operating at 150 r.p.m. The load is 220,000 lb. The Bellows Falls, Vt., plant of the same company has three 17,500-Hp. vertical hydroelectric generators operating at 85.7 r.p.m. They are fitted with Gibbs bearings taking a thrust load of 520,000 lb.

**Pickwick Landing (T.V.A.).**—The 46,000-kva. umbrella-type Westinghouse generators are carried on Kingsbury thrust bearings 105 in. diam., designed to support a total thrust load of 2,500,000 lb., or a unit pressure of 405 lb. per sq. in., with a guide bearing adjacent to the thrust bearing. Each thrust bearing has 10 screw-adjusted shoes, supported in a 10-arm bracket. The rigid bearing produces a restraining effect on the rotor, insuring satisfactory operation, even if the upper guide bearing is removed.

**Bonneville Dam.**—Two 48,000-kva. General Electric generators operating at 75 r.p.m. are carried on General Electric thrust bearings. The thrust load is 3,000,000 lb.

**Safe Harbor.**—Three General Electric generators operating at 109 r.p.m. are carried on Kingsbury adjustable thrust bearings 82 in. diam., adapted to the continuous girder type of support. The thrust load is 1,750,000 lb.

**Guntersville.**—Two General Electric generators, normal speed 69.2 r.p.m., are carried on Kingsbury thrust bearings 87 in. diam., of the same type as used at Safe Harbor. Thrust load is 2,000,000 lb.

**Rosetilla Station,** Chihuahua, Mexico, has a General Electric generator, operating at 120 r.p.m., with a 45-in. Kingsbury spherical thrust bearing, (Fig. 93) carrying a thrust load of 300,000 lb.

**Russian Hydroelectric Stations.**—Among the largest thrust bearings are the General Electric bearings used in nine 77,500-kva. generators in Russia, operating at 88 r.p.m. They carry thrust loads of 2,000,000 lb.

## 6. PERFECT LUBRICATION OF BEARINGS

**VISCOUS FLOW.**—Perfect lubrication is based on the principles of viscous flow, *i.e.*, each particle moves in a direction parallel to the boundary of the enclosing channel. The force that opposes viscous flow of a liquid is known as the resistance to distortion. Let $V_u$, $V_l$ = respectively, velocity of the upper and lower surfaces of a film, and $T$ = thickness. Then $(V_u - V_l)/T = U/T$ = rate of distortion of the film. Let $f$ = force acting on the upper surface of a film to produce a velocity $U$. Then the coefficient of viscosity $\mu$ is

$$\mu = f/(U/T); \quad f = \mu(U/T) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [14]$$

These are the fundamental formulas of the study of viscous flow. Equation [14] can be used to approximate the frictional resistance to rotation of a journal in a bearing when surface speed, film thickness and viscosity are known.

**Friction and Viscosity.**—The relation between friction and viscosity was first expressed by Petroff (1), who assumed that a shaft runs concentrically with its sleeve bearing. This assumption is valid for high-speed, moderately-loaded bearings whose journals run nearly concentrically in practically full-sleeve bearings. Let $a$ = radius of journal; $b$ = width of journal; $N$ = rev. per min.; other notation as before. (See p. 13-15.) Then the tangential friction $F$, resisting rotation at the surface of the journal, is $F = 2\,\pi\,a\,b\,f$, and the surface speed of the journal is $2\,\pi\,a\,N/60$. Combining these two expressions

$$F = \mu\,\pi^2\,a^2\,b\,N/15\,T \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [15]^*$$

The friction force $F$ can be combined with speed and expressed as power loss.

Viscosity coefficient $\mu$ must be expressed in Reyns, $\mu_E$, if dimensions are in inches, and in poises $\mu_M$ or centipoises $Z$ if they are in the metric system. The following relations hold: $\mu_M/\mu_E = 69{,}000$; $\mu_E/\mu_M = 0.0000145$; $Z/\mu_M = 100$; $Z/\mu_E = 6{,}900{,}000$.

**Viscous Flow through Tubes.**†—If it be assumed that there is no molecular slippage of liquid at the tube surface, the laws of viscous flow may be applied to determine the volume and velocity of flow in tubes. Let $V_t$ = volume of flow; $t$ = time, seconds; $P_1$, $P_2$ = respectively, the pressure acting on the liquid at the entering and leaving ends of the tube; $h$ = diameter of tube or thickness of film; $L$ = length of tube; $u$ = linear velocity of flow, and $y$ = fraction of $h$ at which velocity $u$ is determined. Then

$$V_t/t = \{\pi\,(P_1 - P_2)\,h^4\}/128\,\mu\,L \quad . \; . \; . \; . \; . \; . \; . \; . \quad [16]$$

$$u = \{(P_1 - P_2)\,(h\,y - y^2)\}/4\,\mu\,L \quad . \; . \; . \; . \; . \; . \; . \quad [17]$$

**Viscous Flow between Parallel Plates,**† neglecting the influence of the sides, if the plates form a closed channel, is expressed by

$$V_p/t = \{b\,h^3\,(P_1 - P_2)\}/12\,\mu\,L \quad . \; . \; . \; . \; . \; . \; . \; . \quad [18]$$

$$u = \{(P_1 - P_2)\,(h\,y - y^2)/2\,\mu\,L \quad . \; . \; . \; . \; . \; . \; . \quad [19]$$

where $b$ = width of plates, other notation being as before. In equations [16] to [19] only one system of units can be employed (English or metric) and the viscosity coefficient $\mu$ must be in the same system.

**Viscous Flow from Approaching Plates.**—The time rate at which two parallel surfaces will approach when the space between is filled with a viscous liquid, under the conditions of elliptical plates with no central cavity, no oil supply, and the upper plate moving downward under a constant normal force $R$, was found by Osborne Reynolds (2) to be as follows:

$$t = \{(3\mu\,\pi\,a^3c^3)/2R(a^2 + c^2)\}\,\{(1/h_1^2) - (1/h_2^2)\} \quad . \; . \; . \; . \quad [20]$$

$$P_4 = 2R/\pi\,a\,c \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [21]$$

$$P_5 = -(2R/\pi\,a\,c)\,\{(x^2/a^2) + (z^2/c^2) - 1\} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [22]$$

where $a$, $c$ = respectively, semi-major and semi-minor axes of the ellipse; $x$, $z$ = co-ordinates of any point, measured respectively from $c$ and $a$; $R$ = total pressure on plate, lb.; $h_1$, $h_2$ = distance between plates at start and finish, respectively, of the interval considered; $P_4$ = pressure of oil at center of plate; $P_5$ = pressure of oil at any point with co-ordinates $x$ and $z$.

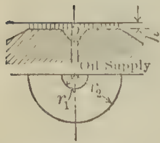


FIG. 97. Separation of Faces by Pressure Oil

**Separation of Circular Plates by High Pressure Oil.**—See Fig. 97. If oil be forced into the recess $D$ under sufficient pressure to cause outward radial flow, the plates will be separated. Two cases may be considered.

**Notation.**—$P_1$ = pressure of oil supply to recess $D$; $P_2$ = oil pressure at any radius $r$; $P_o$ = specific pressure on plate produced by load = $R/\pi\,r_2^2$; $h$ = distance plates are separated; $R$ = total load on plate, lb.; $r_1$, $r_2$ = respectively, radius of cavity and of plate; $G$ = oil supply, gal. per min.; $t$ = time, sec., for plates to move from height $h_2$ to height $h_1$; $\mu$ = viscosity, Reyns. Dimensions are inches; pressures in lb. per sq. in.

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* Petroff's formula was more complicated because he assumed slippage of oil on the bearing surfaces.

† Influence of acceleration at entrance is not included.

Case 1. Circular Plate, load carried by continuous flow of oil under pressure:

$$G = 0.0867h^3R/\mu\,(r_2^2 - r_1^2) \qquad [23]$$

$$P_1 = 2R \log_e (r_2/r_1)/\pi\,(r_2^2 - r_1^2) = 7.35\,\mu G \log_e (r_2/r_1)/h^3 \qquad [24]$$

$$P = 2\,R \log_e (r_2/r)/\pi\,(r_2^2 - r_1^2) \qquad [25]$$

$$P_1/P_o = 2r_2^2 \log_e (r_2 - r_1)/(r_2^2 - r_1^2) \qquad [26]$$

Case 2. Circular Plates, oil supply stopped, plates moving together; oil flow radially outward:

$$t = \{3\mu\pi\,(r_2^4 - r_1^4)/4R\}\,\{(1/h_1^2) - (1/h_2^2) \qquad [27]$$

$$P_2 = 2R/\pi\,(r_2^2 - r_1^2) \qquad [28]$$

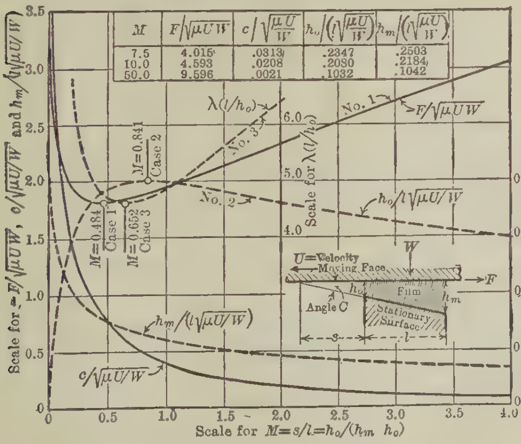
$$P = 2R\,(r_2^2 - r_1^2)/\pi\,(r_2^4 - r_1^4) \qquad [29]$$

**THE TAPERED OIL FILM.**—In 1883 Beauchamp Tower discovered the existence of an oil film completely separating a journal from its bearing. In 1896, Albert Kingsbury (4) discovered that the film was wedge shaped. The hydrodynamical theory, on which modern lubricating practice is based was developed from Tower's work by Osborne Reynolds (2). A good review of Reynolds analysis is given in Kingsbury's paper (4).

**Side Leakage.**—Reynolds neglected the influence of side leakage on pressure and friction in the oil film, and determined the maximum pressures that a bearing could develop with a given film, under ideal conditions. He, therefore, set an upper limit or standard of comparison for actual conditions. However, all other conditions being the same, the narrower a bearing surface, perpendicular to the direction of motion, the lower will be the pressures developed in a film of given form and thickness, due to a larger percentage of the oil drawn in at the front edge leaking from the sides of the narrower bearing. See p. 13–16.

**Film Thickness.**—The factor of safety of a bearing depends largely on film thickness and shape, and minimum thickness is an excellent criterion of bearing safety. Film thickness is difficult to measure. Hence it usually is calculated for reference purposes from data of bearings that have proved successful in service. See Table 11 for calculated capacities of minimum film thicknesses.

**Viscosity Change within Oil Film.**—The friction generated in a bearing whose surfaces are completely separated by an oil film, for the usual range of speeds, is mainly produced within the film itself. It depends on the viscosity of the film, the film thickness, and form, the speed, and the extent and shape of the bearing surfaces covered by the film. See Viscous Flow above. Some of the heat generated in the oil flowing through a bearing film, with consequent rise of temperature which reduces viscosity, passes into the bearing faces. The rest passes out with the oil discharged from the sides and at the far end of the film. The maximum possible rise of temperature in a perfect film may be simply calculated from the mean film thickness, the speed, and the specific heat of oil, assuming that all the heat is absorbed by the film, and that the flow through it is at half the relative speed of the bearing surfaces. In bearing calculations the *mean* viscosity within the film usually is taken to be that of the whole film.



| $M$ | $F/\sqrt{\mu UW}$ | $c/\sqrt{\frac{\mu U}{W}}$ | $h_o/(l\sqrt{\frac{\mu U}{W}})$ | $h_m/(l\sqrt{\frac{\mu U}{W}})$ |
|---|---|---|---|---|
| 7.5 | 4.015 | .0313 | .2347 | .2503 |
| 10.0 | 4.593 | .0208 | .2080 | .2184 |
| 50.0 | 9.596 | .0021 | .1032 | .1042 |

Fig. 98. Effect of Deviations from Optimum Conditions for a Bearing with Flat Wedge Films

**THE FLAT WEDGE FILM** (Fig. 99).—A study of the flat wedge film by means of the Reynolds equations assumes the oil flow to be all parallel to line A-A in the direction of motion, although the actual flow follows the curved lines. See Kingsbury (5). The analysis discloses some important oil film characteristics. The stationary bearing surface may be assumed to be placeable in any desired relation to the upper or moving face, thus varying the relative proportions of $s$ and $l$ and the thicknesses $h_o$, $h_m$. The load $W$ per unit of width $b$, the length $l$ of the bearing face, the speed $U$, and the mean viscosity $\mu$, all may be assumed as fixed. It is assumed that the position of the support of the inclined face, if pivoted, may be altered at will to satisfy these imposed conditions. Three cases, limited to the conditions in which the influence of side leakage on pressure is negligible, as with infinitely wide bearings, are considered below:

**Case 1. Minimum Friction Condition.**—Frictional resistance to motion will be a minimum when $s/l = 0.484$. Film proportions then will be $h_m/h_o = (s + l)/s = 3.065$.

**Case 2. Optimum Load Condition.**—Film thickness at $h_o$ will be greatest when $s/l = 0.841$. This condition was determined by Reynolds. Film proportions will be $h_m/h_o = (s + l)/s = 2.190$.

**Case 3. Optimum Friction Condition.**—Friction, per unit of least film thickness $h_o$, will be minimum when $s/l = 0.652$. Film proportions will be $h_m/h_o = (s + l)/s = 2.533$. Kingsbury (5) has selected this as the *optimum friction condition* for flat-surface bearings.

The following discussion is based on the curves, Fig. 98, for which the above case data are critical points. They are plotted from the Reynolds equations. Case conditions are marked on curves having the same numbers.

The critical points of the curves are found to be so grouped that the range of variations of least film thickness and of friction does not exceed 4%. Case 3 is the best choice from the combined viewpoint of bearing efficiency and factor of safety, although any one of the case criteria would be reasonable.

**Deviations from Optimum.**—Side leakage effects are assumed negligible. Referring to Fig. 98, curve 1, the minimum friction conditions exist with a value of the ratio $M = s/l = 0.484$ (Case 1). Decrease of this ratio increases the angularity of the bearing faces, increases friction and decreases minimum film thickness $h_o$ (curve 2), tending toward metallic contact. Increase of the ratio $M = s/l$ tends toward parallelism of the bearing faces, increases the friction slowly, and decreases minimum film thickness $h_o$ more slowly. These changes bring the conditions closer to those of Cases 2 and 3. By means of tabulated figures at the top of Fig. 98, the curves can be extended beyond the limits of the chart to show friction coefficients, etc., in bearings with nearly parallel faces, far removed from the best. Thus when least film thickness $h_o$ (Curve 2) reaches a value one-quarter of the best value assumed in Case 2, the friction, $-F/\sqrt{\mu U W}$, is over five times the least obtainable. Nevertheless, if great care is used in making and maintaining the bearing surfaces, they might run with perfect lubrication and without wear.

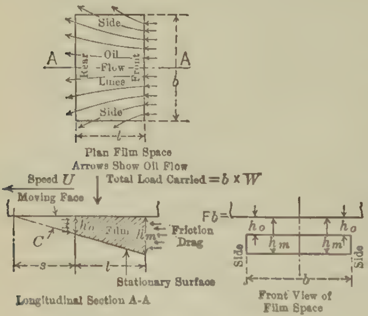


Fig. 99. The Flat Wedge Film

Side leakage correction factors set up for optimum conditions may be used with a fair degree of accuracy for quite different proportions, if the deviation is in the direction of near parallelism of bearing surfaces.

**THE CURVED WEDGE FILM.**—Oil film shapes in journal bearings, when geometrically accurate and simple, are of two forms, *i.e.*, clearance bearing films and fitted bearing films. Both may be expressed by a sine curve equation and shown in developed section when the journal is represented by a straight line. See Figs. 100 and 101. For comparison, the angular extent $\beta$ of the bearing is made equal to the length $l$ of the flat wedge film, Fig. 99. Film thickness at entering end is denoted by $h_E$; that at leaving end by $h_L$.

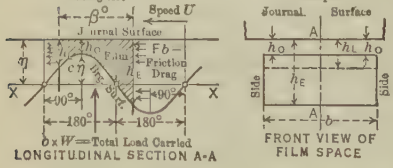


Fig. 100. Curved Wedge Film of Clearance Bearing

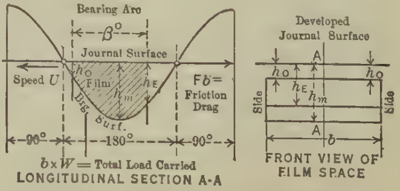


Fig. 101. Curved Wedge Film of Fitted Bearing

**Clearance Bearing Films** (Fig. 100).—The sine curve axis is placed at a distance $\eta$ = radial clearance from the sine curve median line X-X. If $c$ = eccentricity factor, minimum film thickness $h_o = \eta - c\,\eta = \eta\,(1 - c)$. Thickness $h_E$ may be less than maximum film thickness for some film proportions.

**Fitted Bearing Film** (Fig. 101).—The running clearance is zero. Angular contact $\beta$ cannot much exceed 90 deg., and $\beta$ = 180 deg. is geometrically impossible. Maximum film thickness $h_m$ is apt to be in the film, and not at the entering end. Minimum film thickness will be at the far end of the film, *i. e.*, $h_o = h_L$.

**Optimum Conditions in Journal Bearings.**—See discussion, tables and curves relating to optimum bearing classes A, B and C in Loading and Friction of Journal Bearings, pp. 13–18 to 13–25.

**General and Special Conditions.**—See discussion, general tables and charts for Classes C, D, E and F, Loading and Friction of Journal Bearings, pp. 13–24 to 13–28.

**Deviation from Optimum.**—Many series of journal bearing films are possible. The present study is limited to a centrally-loaded 120-deg. clearance bearing. The film is assumed to be a curved wedge, Fig. 102, extending from $h_{\max.}$ at the entering edge to $h_o$ where the film thickness is least, for which the wedge length is $l_o$. Length $l_o$ usually will be a little less than 120-deg., depending on operation conditions.

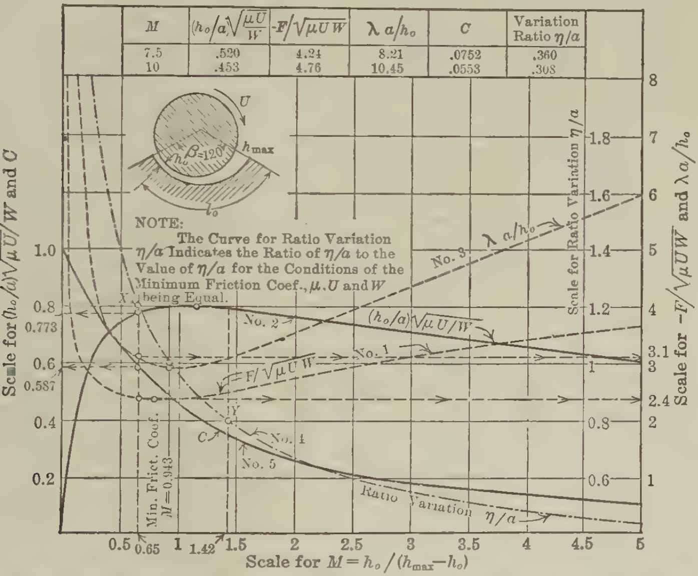

| $M$ | $(h_o/a)\sqrt{\mu U/W}$ | $-F/\sqrt{\mu U W}$ | $\lambda\, a/h_o$ | $C$ | Variation Ratio $\eta/a$ |
|---|---|---|---|---|---|
| 7.5 | .520 | 4.24 | 8.21 | .0752 | .360 |
| 10 | .453 | 4.76 | 10.45 | .0553 | .308 |

Fig. 102. Effect of Deviation from Optimum Friction Coefficient Conditions in a 120-deg. Clearance Bearing

Fig. 102 is a series of curves, showing the relation of certain dimensionless groups in a 120-deg. centrally-loaded bearing, which will be useful in arranging tolerances for journal and bearing diameters. The curves which are plotted against a scale of $M = h_o/(h_{\max.} - h_o)$, are as follows: 1, Friction drag, $-F/\sqrt{\mu U W}$; 2, Bearing capacity, $h_o/(a\sqrt{\mu U/W})$; 3, Friction coefficient, $\lambda a/h_o$; 4, Clearance ratio $\eta/a$; 5, Eccentricity factor $c$. See p. 13–15 for notation. Curves 1 to 3 correspond to those with the same numbers in Fig. 98. The application of the curves can be illustrated by an example.

Example.—Determine the effect of an increase of 20% and a decrease of 20% in clearance ratio from optimum for optimum friction condition of a 120-deg. clearance bearing.

*Solution.*—Minimum friction is shown by Curve 3 to correspond to a value of $M$ = 0.943. Optimum clearance ratio $\eta/a$ corresponding to this has been taken as unity in plotting Curve 4. An increase of 20% corresponds to point X on Curve 4 with a corresponding value of $M$ = 0.65. The ordinate through X intersects the other curves to give the following values: Friction drag, Curve 1, 2.4; bearing capacity, Curve 2, 0.773; friction coefficient, Curve 3, 3.1; eccentricity factor, Curve 5, 0.587.

A decrease of 20% in clearance ratio $(\eta/a)$ corresponds to the point Y on Curve 4, with a corresponding value of $M$ = 1.42. From the intersection of this ordinate with the several curves, the following values are found: Friction drag, 2.53; bearing capacity, 0.788; friction coefficient, 3.1; eccentricity factor, 0.348.

The tabulation above Fig. 102 permits the extension of the curves beyond the limits of the chart. With a value of $M = 10$, the other values are such that an accurately made bearing would function properly with pure film lubrication.

The curves of Fig. 103 have been derived from those of Fig. 102 to show more directly the influence of deviation of clearance ratio from optimum. Small deviations have little influence, but if clearance ratio variation is large, the efficiency of the bearing is sensibly affected, especially if clearance is reduced. The friction coefficient factor increases more with a reduction of clearance ratio than with an increase. If it is not feasible to design a bearing with a clearance ratio corresponding to optimum conditions, a larger rather than a smaller clearance ratio should be provided. The relation of curvature to safety of starting and stopping under heavy loads should not be overlooked, as it leads in the other direction, *i.e.*, the choice of a smaller clearance, or to the fitting of a portion of the bearing surface to the journal. (See Curve 4.)

**EFFECT OF SIDE LEAKAGE.**—The influence of side leakage on the capacity of a perfectly lubricated thrust bearing, and other bearings with wedge-shaped films, is considerable. Fig. 104 compares the results of several investigators. Abscissas are ratios of (length/width) *i.e.*, $l/b$. Ordinates are capacity factors $L_1$, which show the percentage of the *ideal load* that can be carried on a film with finite dimensions of length and width. The ideal load with which the bearing of finite length and width is compared, is one of the same finite length $l$ (in direction of motion), but which is understood to be a part of one whose width $b$ is so great (infinite) that there is no side leakage to reduce the capacity. The points determined by Michell (9), by Martin (10), and by Duffing (12), apply to a flat wedge film whose proportions are two to one, *i.e.*, $h_m/h_0 = 2.0$. Kingsbury extended the investigation to cover curved wedge films such as occur in journal bearings of the fitted and clearance types. The correction factors in Fig. 38 were developed from this study.

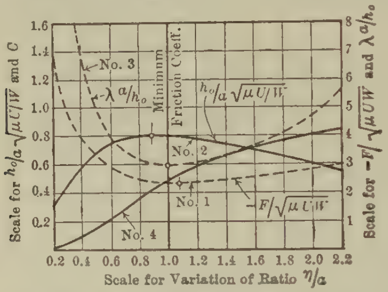


FIG. 103. Effect of Deviation from Optimum Friction Coefficient Conditions in a 120-deg. Clearance Bearing

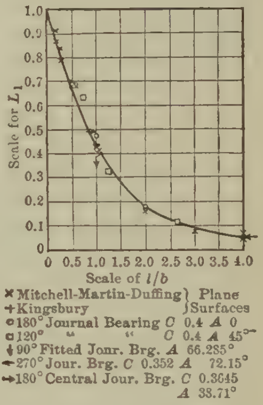


FIG. 104. Influence of Side Leakage on Carrying Capacity of Oil Film as Determined by Analyses of Michell (9), Martin (10), Duffing (12), Kingsbury (6).

## 7. MISCELLANEOUS BEARINGS

Study of the oil film type of journal bearing has indicated the possibility of their successful application within dimensions identical with those now used for ball and roller bearings. In the Nomy bearing, separate pivoted shoes are mounted on a ring that revolves with the shaft. The shoes bear on the inside of a stationary outer race. Provision is made for automatic lubrication. The Fast bearing has shoes integrally formed on the inner ring and connected thereto by stems. The bearing is quite rugged.

In neither of these bearings is the revolving element integral with the shaft. It, therefore, must be carefully fitted and clamped against the shaft, usually against a shoulder, and keyed or otherwise held to prevent slippage at starting. Otherwise these bearings are mounted in the same way as ball or roller bearings. A comparison of a bearing of this type and the more usual construction in which a bearing shell about one diameter long supports the shaft journal directly, shows the space occupied to be about the same in both cases for similar service. Manufacturers should be consulted for detailed information.

**ROLL-NECK BEARINGS.**—Roll necks in mills have long presented a serious problem, especially in steel mills. Early practice, still widely followed, was crude and very wasteful of power. The bearings were undependable. Wear on necks and bearings was considerable. The bearing equipment acted as a bar to the construction of larger mills. To remove this handicap, anti-friction bearings have been used with considerable success, but exhibited certain limitations. Since 1930, the oil film type of bearing has been successfully applied to roll necks. One of these, the Dahlstrom roll neck bearing, marketed by the Morgan Construction Co., Worcester, Mass., under the trade name of Morgoil Bearing, is described in *Trans.* A.S.M.E., IS 55–2, 1933.

The Morgoil bearing comprises a ground and polished steel sleeve or journal fitted over the roll neck, the latter being tapered at an angle of about 12 deg. The journal thus can be removed easily from the neck, which also can be made of larger diameter than is ordinarily possible. The sleeve is keyed to the neck. The bearing comprises a casing, into which is inserted a steel or bronze liner, faced with centrifugally-cast babbitt. Oil for lubrication is pumped into the bearing through holes in the casing, and flows through appropriate grooves to a drain pipe. Tests described in the paper above cited show this bearing to be capable of withstanding loads in excess of the roll-neck strength, and to effect considerable savings in power.

In Kingsbury roll-neck bearings installed on a wide strip mill of the Inland Steel Co., 1933, load was carried directly on the neck of the roll. The bearing shell was water-cooled. Oil was fed to the bearing by a gravity system to make up leakage at the seals. High-pressure oil was introduced to reduce starting friction, permitting operation at any speed, irrespective of the load on the bearings. The friction was about the same as for roller bearings for all conditions.

**OILLESS BEARINGS.**—Although bearings are offered that are lubricated by other media than oil or grease, it is assumed here that the term Oilless Bearing applies only to bearings to which no lubricant is expected to be applied. These comprise a wide variety of bearing shell materials with which lubricating substances are combined by one means or another in the process of manufacture. Among these are: 1. Fiber, plain or impregnated with lubricant; 2. Wood, plain or impregnated with lubricant, or with cavities filled with hard lubricant; 3. Special highly graphitic alloys; 4. Metal bushings with insets of hard lubricant; 5. Press-moulded mechanical mixtures of bearing materials and lubricants. Some of these materials will carry light loads at moderate speeds without serious wear. They are best suited to intermittent service or for very low speeds. Manufacturers should be consulted for detailed information concerning them.

## 8. BEARING MATERIALS

**METALS.**—**Hardened Steel** will work well on hardened steel at low speed under very heavy loads. It is fairly durable even if imperfectly lubricated. The same metals work well at high speeds when carrying moderate loads if well designed, finished and lubricated. Hard-surface bearings are least subject to abrasion by operation under adverse conditions. They are much used for small parts, but are too costly for large ones.

**Gray Cast Iron** is useful for bearing faces, especially if chill-cast. For moderate loads, such iron is excellent for both bearing surfaces, even for high-speed bearings, if properly lubricated. Cast iron works well with softer bearing materials, as babbitt metals and bronzes when properly lubricated, and can be made to carry high pressures.

**Soft Steel** will run well with babbitt metals and other white metal bearings, and will carry considerable pressure if properly lubricated. It also will run directly on cast iron. Soft steel on soft steel is not recommended, because of the tendency to seizure. Hard steel on soft steel sometimes is useful. A carbon content of 0.30 to 0.40 is recommended for shafts running on soft metal bearings. Soft copper-base alloys containing lead are useful for moderate speed and heavy pressure. There is a tendency, however, to line brass and bronze bearings with a thin coating of white bearing metal. This avoids the tendency of high-copper alloys, when overheated in bearings, to destroy the finish of the opposing cast-iron or steel surface by roughening, plating it, or producing surface cracks.

**Bearing Metal Alloys** are discussed more fully on pp. 4–62 to 4–67. In general, they are classified as bearing bronzes and babbitts. A more exact classification is low-lead bronze, high-lead bronze, lead-base, and tin-base bearing metals. In the last group is the bearing metal produced by Isaac Babbitt, containing Sn, 83 $^1/_3$, Sb, 8 $^1/_3$, Cu, 8 $^1/_3$. This alloy, and others with varying compositions of the same metals, still are called Babbitt Metal. Kingsbury uses a composition of Sn, 85, Sb, 10, and Cu, 5. The U. S. Navy composition is Sn, 88 $^1/_2$, Sb, 7 $^1/_2$, Cu, 4.

**Non-Metals.**—*Fiber* (pressed paper) washers often are used for thrust bearings for low speed and considerable load, or high speed with light load when oil lubricated. *Jewels* or ruby or other crystals are used with hardened polished steel pivots in timepieces and instruments, oil lubricated. *Ivory* bushings are found in the best old clocks, with hardened steel pivots. *Hardwood* bearings have been used, lubricated by tallow and soaps.

**Lignum Vitae.**—In the best applications lignum vitae is used with grain perpendicular to the bearing face. It is used to line stern tubes of ships, with fresh or salt water lubricant. The bearing in a vertical-shaft hydraulic turbine often has a lining of lignum vitae blocks in longitudinal grooves in the bearing shell. Adjustment is required to compensate wear. In this service they are lubricated with fresh water. Lignum vitae has been used in footstep bearings under water in water-wheels.

**Rubber** is used to line bearings where only water is available as lubricant and where bearing pressure is light. It is useful in guide bearing bushings for vertical revolving shafts, and in stern tube bearings in ships.

**Resin-bonded Bearings.**—Mr. L. M. Tichvinsky of the Westinghouse Elect. & Mfg. Co. gives the following résumé of literature on this bearing material:

The rapid development of the phenolytic or resin-bonded bearings was directly connected with the steel mill industry. According to their structure and manufacture, the synthetic resin-bonded bearings can be classified into three groups: 1. Manufactured from plain or graphitic molding powders; 2. Manufactured from various felts of impregnated non-oriented fibers; 3. Manufactured from laminated products.

The bearings of Group 1 are manufactured in great quantities for low load application, and for guide bearings of high-speed machines. High brittleness limits their application.

Due to different bases, bearings of Group 2, are tougher and more rigid than the bearings of Group 1. Their application is also for low or medium loads. Group 3 covers the most important type of bearings bonded with synthetic resin. They also are called laminated bearings because of their base consisting of woven textile fabrics. Occasionally, for less important performances, they are made from laminated paper products. In manufacturing these bearings high pressures and high temperatures are applied for a predetermined period of time during which the powdered synthetic resin liquifies. On cooling, a well-bonded product is obtained. These bearings are marketed under various trade names.

For more severe operations, as in steel mills, these bearings perform most efficiently with water alone for both lubricating and cooling. The average physical and mechanical properties of a good laminated bearing material are as follows: Specific gravity, 1.25 to 1.40; weight (approx.), 0.05 lb. per cu. in.; Brinell hardness, 30–42; tensile strength, lb. per sq. in. (along grain), 8,000–15,000; (across agrain), 6,000–11,000; modulus of elasticity, 1,000,000; compressive strength, lb. per sq. in., (perpendicular to laminae) 40,000–45,000, (parallel to laminae), 20,000–24,000; water absorption, 7-day test, 1–3%; oil absorption, negligible.

## 9. MAKING AND FINISHING OF METAL BEARING SURFACES

**ANCHORING.**—For holding a soft metal lining to harder bearing surfaces the ordinary dovetail circular anchor is much used, but the T-head anchor is better because shrinkage of its neck holds the metal faces together. Anchors should be small and sparingly used, because they remove the backing of the soft bearing face by shrinking away from it. Sometimes they are omitted entirely. Small T-slot grooves are useful, especially if machined. Simple anchors sometimes are formed by a drill point, or by plain grooves.

**TINNING.**—In order that heat may pass freely from the bearing lining to its backing, the latter often is tinned. The surface to be tinned should be machined or ground smooth and must be clean. The shell should be heated, flux applied, and the tinning metal then spread thoroughly by scrubbing. It is best to tin the surface with the lining material. Usually with the residual preheat the shell is then ready to receive its soft metal lining. Some metals tin more easily than others. Steel and ordinary bronze are more easily tinned than cast iron and the aluminum- and other special bronzes. Copper plating before tinning improves the bond.

**PEENING.**—The value of peening or burnishing in the preparation of soft metal bearing faces is an open question. The resulting texture may be less uniform, even though sometimes harder. Peening holds the lining tightly to dovetail anchors, but peening a face that has been applied to a tinned surface is apt to loosen the tinned bond. Some alloys are hardened more by peening or burnishing than are others.

**FITTING UNDER LOAD** is useful for large vertical thrust bearings, and often is used to compensate deflections produced by loads. In this process, a few turns at very low speed are made to spot the softer surfaces by the harder. A fairly heavy coating of lubricant will prevent overheating during the process. The hard face then is lifted and the rubbed spots on the soft face scraped down lightly. High spots on the harder face may be reduced by fine stoning. Usually all fitting is done on the softer face. The process is repeated until a fit of the desired extent and accuracy is obtained. The oil used with this fitting should be only heavy enough to get the desired marking on the bearing surface without injury or waste of time.

**RUNNING-IN** of bearings often is used as a system of inspection or of preliminary service. It sometimes is rather brutal, the journal being used to ream or burnish the bearing. It should not be taken as a substitute for good design and workmanship. See

Fitting under Load, above. Running-in, if carefully done, will improve the bearing surface smoothness. It may be used in lieu of fitting under load with small machines. It is best, however, to pre-machine the bearing elements as nearly correct as possible, and to use designs and clearances suitable for immediate service.

**FORMING.**—The hardest surfaces can be finished by grinding, and softer materials by machining, grinding or scraping. The smoother and truer the surface the better will the bearing operate with proper lubrication. Lapping, after grinding, or over a smooth tool finish, is one of the best ways to produce accurately cylindrical, flat, or spherical surfaces on steel and cast iron. See p. 21–87. Final finishing may be by rotation under a lightly pressing block of wood faced with leather, using fine abrasive and oil. Scraping, stoning, and filing are all useful for finishing and for fitting one surface to another. Smooth file finishing on a revolving a shaft is common, but it will not true a surface initially out of round. Careful final finishing or fitting of the bearing surface is essential. A very smooth and accurate tool or grinder finish may not need further treatment. In important work, fitting of the softer bearing surface to the other or to a master surface by hand scraping is common. Thus, a bearing may be fitted to a mandrel of a diameter equal to that of the journal plus the desired running clearance.

## 10. LUBRICATION OF BEARINGS

Bearings are lubricated by the following systems: 1. Automatic; 2. Intermittent; 3. Bath; 4. Splash; 5. Pressure; 6. Mechanical.

**AUTOMATIC LUBRICATION.**—The rotation of the shaft causes oil to be fed to the bearing surfaces. Lubrication ceases when the shaft stops turning. The systems in common use are: Ring oiling, chain oiling, disc oiling, capillary oiling, waste pack oiling, vacuum oiling, and oiling by means of a shaft-driven pump of any type. All these systems lift oil to the bearing from a well below it, and are capable of continuous operation for long periods, because oil leaking from the ends of the bearing or elsewhere returns to the well and is used over again. See Figs. 20 and 28.

**INTERMITTENT LUBRICATION.**—Oil is applied to the bearing at intervals from a source independent of the rotation of the shaft. The simplest, but least dependable system, is hand oiling from an oil can. Cup oiling is better, as it has the possibility of being practically continuous and uniform, with visible and adjustable rate. Wick oiling, with textile or metal wicks, is, next to hand oiling, the simplest and cheapest system. The number of wick strands determines the rate of supply. The wicks filter the oil. The general faults of intermittent systems include usually complete loss of oil after it leaves the bearing, frequent refilling of cups and wells, and inadequate, or wasteful oiling by reason of ignorance or carelessness.

**BATH LUBRICATION.**—The bearing faces are completely immersed in a bath of oil at all times. This system frequently is used for vertical shafts, and is simplest for the lower end of the shaft, where thrust and radial bearings can be combined. At other places on the shaft, the well may be formed by a revolving bowl, or the well may have a tube that extends upward around the shaft, and inside of a sleeve that extends downward from a hub to form the journal of the radial bearing. See Fig. 31. This last system is commonly used for thrust bearings placed anywhere along a vertical shaft. Packing may be used in stuffing boxes to hold oil in any kind of a recess about a bearing, horizontal or vertical, that will make it a bath-oiled bearing. Oil will leak away, even though slowly, from such a bearing. There also is the danger of setting up the packing so tightly that it will run hot and the shaft will be worn within it.

**SPLASH LUBRICATION** was common in the crank cases of internal combustion engines and still is considered a simple way to oil all the bearings of a trunk-piston engine. The splash is produced by the lower ends of the connecting rods dipping into the oil in the crank case. Various refinements are used to control the splash. This might be considered a continuous system, because most of the oil is returned to the crank case to be used again, although partly contaminated or broken down by exposure to cylinder walls. The system is used in compressors and other crank machines.

**PRESSURE LUBRICATION.**—Pipes lead oil to the bearings from a source of oil under pressure. The pressure is available at the point of application of oil to the bearing and will therefore tend to keep oil passages open. Bearings with this system usually are simple when the whole machine is so enclosed that oil will drain to the sump from which an oil pump redistributes it to the bearings. The rate of oil supply to each bearing may be regulated by making the size of the supply hole to suit the pressure in the system. The source of oil may be independent of the operation of the machine, as when a separate motor-driven oil pump is used. It may otherwise be driven by some shaft in the machine, and, therefore, operates only when the machine is running. Each method has its drawbacks, and judgment must be used in selection.

**MECHANICAL OILERS.**—A variety of mechanical devices are on the market for producing what may be styled centrally-controlled, intermittent-lubrication of bearings. They eliminate oil cups, but introduce a multitude of small oil pipes, through which oil is fed or forced intermittently, in measured amounts. These devices usually operate by ratchets driven from the shaft of the machine whose bearings they oil.

**COOLING OF BEARING OIL.**—To keep oil in a bearing at a safe temperature, the heat of friction due to operation must be dissipated. Safe temperature depends on the kind of oil used and the conditions against which the bearing must be protected. Thickness of oil film and heat of friction both increase with viscosity of oil.

**Direct Air-cooling.**—The oil housing surrounding the bearing should be sealed tight to prevent leakage. The oil then is allowed to warm up sufficiently to dissipate friction heat to the surrounding air by radiation and convection. As the oil heats it becomes less viscous and the friction loss in the bearing decreases. The rate of radiation and convection increases as the oil gets warmer. When heat generation and heat dissipation become equal the oil temperature will remain quite uniform, being affected only by changes in the surrounding air temperature. In air-cooled bearings the oil level should be watched and new oil added, as required, to replace that lost by leakage and evaporation.

**Direct Water-cooling** is applied in three ways: 1. Water-cooling coils in the oil within the housing; 2. Water-cooled shoes; 3. Water-cooled collars. With all of these applications water must be circulated fast enough to remove the heat of friction from the oil and keep it at a safe temperature. Means must be provided to prevent leakage of water into the oil, and leakage of oil from the housing. New oil must be added from time to time to make up loss from leakage and evaporation. When water-cooling coils are used in the housing, the oil should have a pour test point considerably lower than the temperature of the coldest cooling water, to prevent congealing on the coils.

**Self-contained Unit Circulating System.**—Each machine in which a bearing is installed has an individual oiling system that circulates oil through all bearings. Within the system are means for air cooling or water cooling the oil, depending on the amount of friction heat to be dissipated. When the pump is driven directly by the machine itself, oil supply to the bearings begins when the machine is started. Hence, if the oil in the bearing housing has leaked away during a shut-down and exposed the rubbing surface, the bearing will burn out before the pump has supplied enough oil to bring its level up above the bearing faces. This danger must be removed by inspection before starting the machine. Siphoning of oil back through the inlet pipe should be prevented by placing its opening above the oil level, or by providing an air vent hole.

**Central Station System.**—With this system oil should be circulated through the housing of each bearing at the rate recommended for each case, to lubricate the bearing and carry away the friction heat developed. The piping in such a system often is so extensive that the oil will be sufficiently air cooled in making the circuit. When necessary, however, the system is provided with water-cooling coils in the central station.

# LUBRICANTS

By J. C. Geniesse.

A lubricant is a liquid or soft solid substance, as oil, grease or graphite, interposed between moving parts to reduce friction and wear. Liquids are almost universally used where rubbing or bearing surfaces are designed to retain most of the lubricant within an enclosed system. Greases and solids are used where leakage is too high to retain liquid. The most important property of a liquid lubricant is its viscosity at normal operating temperature, since it determines the friction and whether there shall be any wear. All other requirements imposed on a viscous lubricant are to insure that it remain chemically inactive under all conditions of operation for a reasonable length of time. Requirements for liquid, plastic and solid lubricants are the same, *viz.*, that they must produce low friction and wear. These requirements are not so easily defined in terms of properties of the plastic and solid lubricants, since they usually are used where metal-to-metal contact is inevitable, and where wear is almost unavoidable.

## 1. TYPES OF LUBRICANTS

**MINERAL OILS.**—Lubricating oils of petroleum origin are almost universally used, because of availability and stability. They are complex mixtures, primarily of many hydrocarbons. Eastern base, or Pennsylvania oils, contain a large proportion of paraffin hydrocarbons; Western base oils generally contain more naphthene and less paraffin hydrocarbons. Lubricating oils from Midcontinent crudes lie between these extremes. Coastal oils from states adjoining the Gulf of Mexico may be similar to those from California or the Midcontinent area. Specific gravity, and change of viscosity with both temperature and pressure, are lower for paraffin than naphthene type oils. Carbon residue, color and heat stability depend more on type and degree of refining than on source of the oil. However, by means of solvent and other special methods of refining, naphthenic and unstable compounds may be removed from a lubricating stock, leaving a fraction predominantly paraffinic. The paraffinicity of a naphthenic or mixed-base lubricating stock may be increased, producing an oil of viscosity and gravity equivalent to those of Pennsylvania oils.

**ANIMAL, VEGETABLE AND FISH OILS** frequently are called fixed oils, because they are not volatilized without decomposition. Distillation causes them to decompose, leaving a carbonaceous residue. The oils most frequently used are: *Animal*, lard, tallow, neatsfoot; *Fish*, sperm; *Vegetable*, castor and rape seed. With the exception of sperm oil they consist mainly of glycerides of saturated and unsaturated long-chain acids. Sperm oil is chiefly myricyl palmitate, a long-chain ester of a monohydric alcohol. All contain unsaturated compounds which oxidize more easily than petroleum oils. This fact has retarded their general use at high temperatures.

Fixed oils adhere more tenaciously to metal than straight mineral oils. Consequently, they are used for compounding purposes where water vapor has a tendency to condense on bearing surfaces. Where humidity is high, as in marine service, most oils are blended with 5 to 10 % of fixed oil to overcome the detrimental effect of water. They have better viscosity-temperature slopes than mineral oils, and are used as a medium for introducing other agents, as sulphur, since they have greater solubility and chemical affinity for such materials.

**GREASES,** in general, are mixtures of mineral oil and soap. Other compounds, as fixed oils, graphite, etc., may be introduced to obtain specific properties. The soap reduces flow or loss of lubricant. In general, viscosity of the mineral oil is chosen on the assumption that no leakage will occur; the soap then is added to prevent leakage. At high rates of shear, friction offered by a given grease is not much higher than that of straight oil of the same viscosity as that incorporated in the grease; at low rates of shear friction is considerably higher.

**MISCELLANEOUS.**—Graphite, wax, soapstone and other soft solids are used for lubrication, particularly where rubbing speeds are low, and where the type of construction makes it difficult to retain lubricant on bearing surfaces. Water also has been used extensively for lubricating bearings. Steel mill roll-necks with bakelite or other resin bearings can be lubricated quite successfully with water. If operation is intermittent, the water must be removed when not in use, to avoid excessive corrosion.

## 2. PROPERTIES OF LUBRICANTS

**VISCOSITY.**—All lubricating oils decrease in viscosity as temperature increases, the relation being complex. The A.S.T.M. has published charts for both kinematic and Saybolt viscosity whose scales are constructed to give straight lines when data are plotted on them. See Fig. 107. Paraffinic oils change less with temperature than do naphthenic oils; vegetable and animal oils vary still less.

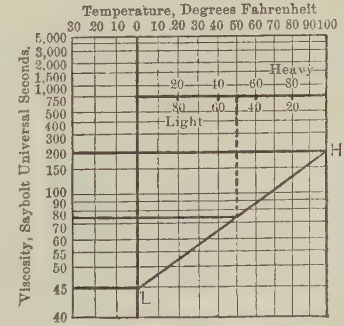


Fig. 105. Viscosity Blending Chart

Viscosity usually is measured at temperatures of 100°, 130° or 210° F. in the Saybolt instrument; the time in which a given volume of oil flows through an orifice under specified conditions is taken as the viscosity. More recently, capillary kinematic methods have been developed to measure viscosity more accurately. Refer to latest A.S.T.M. Standards for detailed procedure. Various attempts have been made to numerically express the viscosity-temperature slope for purposes of convenience. Dean and Davis use an index of 100 for Pennsylvania oils and 0 for extremely napthenic oils. On this scale, Midcontinent oils average 75. See Fig. 107 for a convenient graphical method for estimating the viscosity index of lubricating oil.

Blending often is employed to produce oils with intermediate viscosities. Fig. 105 illustrates the procedure generally used to predict the viscosity of a blend. Any 100° temperature range on a viscosity-temperature chart may be used for this purpose, if the upper and lower temperature limits are used, respectively, for the heavy and the light components.

All oils also change markedly in viscosity when high pressures are imposed on them.

Hersey has shown that the equation $\log (Z_1/Z_0) = K_1P + K_2$ may be used to depict this relationship. In this equation, $Z_0$ = viscosity at atmospheric pressure; $Z_1$ = viscosity at pressure $P$; $K_1$, $K_2$ = constants. Paraffinic oils vary less with pressure than naphthenic oils, and fixed oils still less.

**CONSISTENCY** defines the resistance to shear of plastic substances, as grease, just as viscosity is applied to viscous liquids. Consistency of grease varies both with temperature and rate of shear. At high rates of shear the consistency is lower than at lower rates. Consequently, the force required to shear grease in a given bearing does not decrease proportionately with speed. This permits use of a product that does not readily leak, and yet offers small resistance to shear at high speeds. Consistency ordinarily is measured by a penetrometer. A double-pitched cone of standardized weight and dimensions is dropped into the product, maintained at a definite temperature. The depth of penetration, measured in hundredths of a centimeter, is reported as the penetration of the material. See A.S.T.M. Tentative Standards.

**SPECIFIC GRAVITY,** *per se*, is of no value in predicting quality of a lubricating oil, although it frequently has been used to determine or specify the source of the oil. It varies linearly with temperature. See Fig. 106. Paraffinic oils are the lightest; naphthenic, intermediate, and the animal and vegetable oils the heaviest. Compounding, particularly with lead soaps, increases specific gravity of oil.

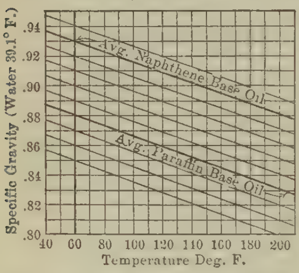


Fig. 106. Variation of Specific Gravity with Temperature of Oil

**POUR TEMPERATURE** of an oil is the lowest temperature at which it will flow, under conditions prescribed by A.S.T.M. It is used to predict the lowest temperature at which an oil will flow freely to bearings or oil pump. Paraffinic and some naphthenic crudes contain wax which congeals at low temperatures, and must be dewaxed for use in cold weather.

**FLASH AND FIRE.**—The A.S.T.M. flash point is determined by heating the oil in a vessel under prescribed conditions until sufficient vapor is driven off to flash when exposed to a flame. A.S.T.M. fire point is the temperature at which sufficient vapor is given off to give a continuous flame. Only under unusual circumstances are fire and flash points of value in predicting quality of an oil, except to indicate its source. In general, flash and fire points of paraffinic oils are higher than those of naphthenic oils.

**COLOR** of lubricating oils is determined in the A.S.T.M. Union Colorimeter by comparing the density of light transmitted through the oil with that transmitted through glass color standards. An arbitrary scale of 0 to 8 is used to designate color. Commercial crank-case oils range from 1 to 6 and heavy gear oils from 6 to 8. Color is of value only to determine the extent of refining, when considered in conjunction with the source, since naphthenic oils are, in general, lighter than paraffinic oils. Solvent refining methods, however, have upset the value of color as an indicator of degree of refining.

**CARBON RESIDUE** is percent of residue remaining after destructive distillation of a weighed quantity of oil under prescribed conditions. It originally was developed to compare carbon-forming qualities of lubricating oils for internal combustion engines. Correlation of laboratory tests with engine tests has been only moderately successful. Modern refining methods tend to make this test less useful for predicting carbon-forming tendency of motor oils.

## 3. LUBRICATING OIL REQUIREMENTS

**INTERNAL COMBUSTION ENGINES.**—Requirements of crank-case lubricating oil are more difficult to meet than those of most lubricants, because of the wide range of temperatures to which it is subjected. At the lowest atmospheric temperatures, viscosity must be below 40,000 Saybolt seconds in order that starting torque be within the capacity of the starting motor. At the highest operating temperatures, which may be 300° F., it must have sufficient viscosity to maintain an oil film in the bearings. No lower limit of viscosity has been established, and probably would have no practical value, since oil consumption, which depends on viscosity under a given set of conditions, becomes excessive before bearing failures develop. This should not be construed to mean that viscosity is the controlling factor determining consumption. Motor speed, and mechanical condition, are responsible for far greater variations.

To simplify lubricating oil viscosity requirements, the S.A.E. has promulgated a viscosity classification to cover atmospheric temperatures and types of operation ordinarily encountered. The ranges (see Fig. 107), are classified at low temperatures for winter oils and higher temperatures for summer oils.

Pour point must be approximately equal to or below the lowest anticipated atmospheric temperature, to assure uninterrupted supply to pump and cylinder walls.

For reasonable life of oil it must not oxidize so rapidly as to form sludge or carbonaceous matter which ultimately may interfere with lubrication. There are no infallible quantitative tests for this determination, but almost any well-refined product marked by a reliable concern is satisfactory. Carbon deposits in combustion chamber and on valves are not so serious a problem as in the past, since refining methods have made possible the marketing of oils which deposit very little carbon in combustion chambers.

**GEARS.**—The requirements for gear lubricants depend on load and speed of the gears, and on construction of the gear box. If load and speed are not very high, oils of viscosity of 80 to 200 Saybolt sec. at 210° F., depending on operating temperature, are generally recommended. If the gear box is not tight enough to prevent leakage, oil of the same viscosity should be incorporated in a grease. If speeds are high and gears machined to a good fit, oils in the range S.A.E. 20 to S.A.E. 40 may be used.

Many automotive rear axles operate at very high loads and with appreciable rubbing speeds. To avoid scoring, they must be lubricated with special products. Two types of oil have been developed: 1. Mild extreme pressure lubricants; 2. Powerful extreme pressure lubricants. They contain active chemicals which presumably act as anti-weld agents at points of excessive load. For hypoid gears, where rubbing speeds are high, more powerful lubricants, containing more active ingredients, are necessary. Extreme pressure lubricants should be free of insoluble foreign materials and have the same viscosities as those used on other gears.

Gears in an industrial establishment are designed to operate at high loads. It then is necessary to use mild extreme pressure lubricants. No rigid rule is applicable to determine the type of lubricant necessary for a given set of gears, but, in general, if wear is high, leading to frequent replacements, an extreme pressure lubricant is advisable.

**STEAM TURBINES.**—The most important requirement of a turbine lubricant is that it be sufficiently stable to avoid forming oxidation products which tend to form stable water-oil emulsions. No fool-proof laboratory test exists for predicting this quality. Experience is the best guide. The viscosities normally used range from 150 Saybolt sec. at 100° F. to 100 sec. at 210° F., depending on speed and operating temperature.

**AIR COMPRESSORS.**—Lubricating oils must not form large carbon deposits, particularly on valve seats. The resultant valve sticking may lead to hot spots which ignite the air-oil vapors causing an explosion that may be disastrous both to equipment and life. Fixed oils must not be used, except in small quantities, as they are prone to form carbonaceous residues. Viscosities normally used range from 250 to 750 Saybolt sec. at 100° F., depending on amount of compression and the resulting temperatures.

**STEAM CYLINDERS.**—Condensation of moisture during the expansion stroke in steam cylinders makes necessary the use of compounded lubricating oils, which are effective in the presence of water. If high superheat is used, little condensation occurs, and not over 5% of the compounding agent is necessary; 10% is advisable with saturated steam. Acidless tallow is the compounding material generally used, although many vegetable and animal oils are satisfactory. The function of the compounding agent is to form an emulsion with water which will be carried to the cylinder walls to form a tenacious lubricating film. Viscosities generally used range from 90 to 200 Saybolt sec. at 210° F.

**RAILWAY CAR JOURNAL** lubricants must not be too viscous at the lowest operating temperatures, or the cotton waste will not carry oil to the brasses. In general, summer oils should have a stable viscosity of approximately 225 sec. at 130° F., and a pour point of 32° F. Winter oils should be 175 sec. at 130° F., and pour at 0 to 10° F. Wear reduction of brasses is possible by compounding, but since they are easily replaced, it is generally believed that the increased cost of such oils is unwarranted.

## 4. THE SELECTION OF MINERAL LUBRICATING OIL *

In the discussion of bearing design the viscosity of lubricating oil has been emphasized. Viscosity varies with temperature, and the viscosity-temperature relation is not the same for all oils. These variations have been classified by means of the viscosity index numbers proposed by Dean and Davis (generally known as V. I. numbers). The V. I. number for paraffin-base oil averages 100 on the Dean and Davis scale, and for naphthene-base oils 0. Special oils are now available with V. I. numbers higher than 100. The relations of the viscosities to the Dean and Davis numbers, and to various classification scales and Saybolt Universal viscosities (S.U.V.), are given in the chart, Fig. 107.

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* Contributed by H. A. S. Howarth.

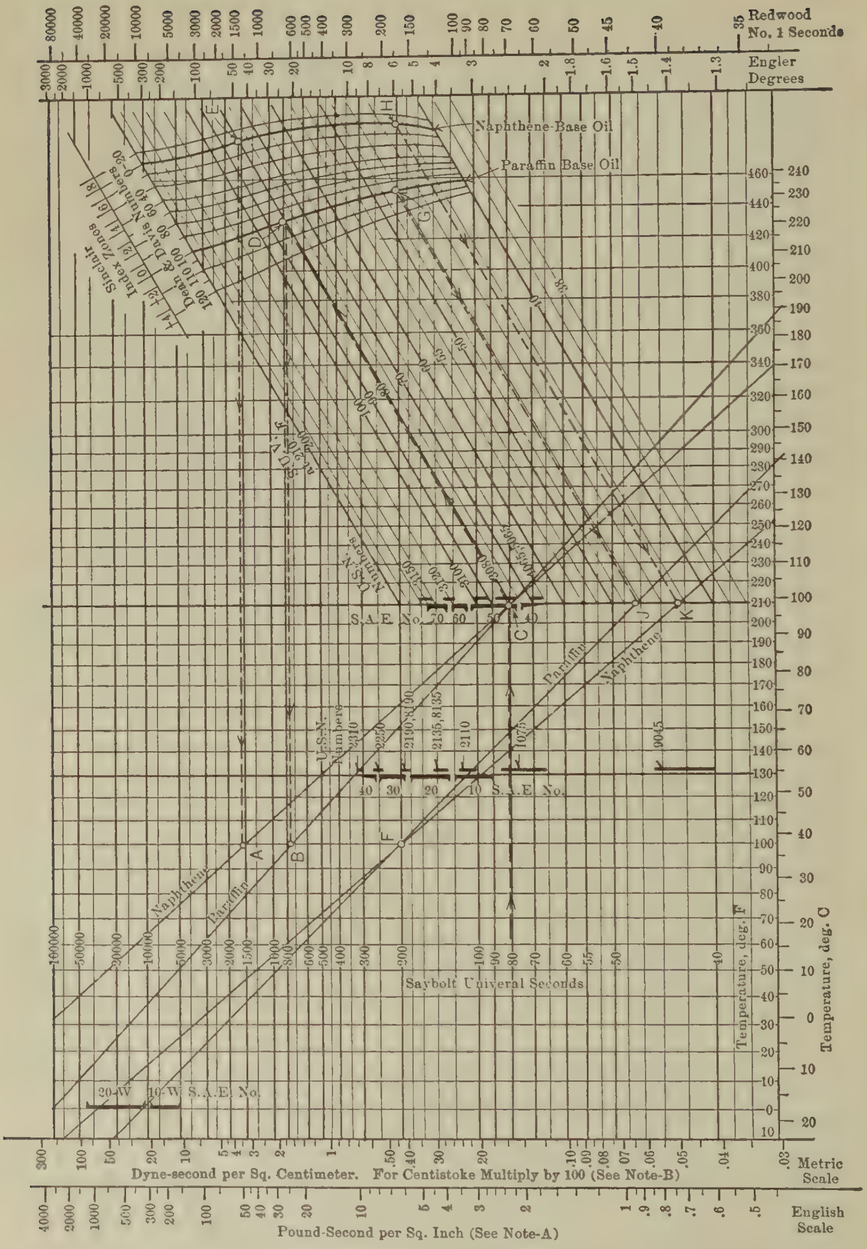

Fig. 107. Viscosity-temperature and Viscosity-index Chart

Note A. For absolute English viscosity coefficient (Reyns) multiply English scale reading by $10^{-6}$ and specific gravity referred to water at 39.1° F.

Note B. For absolute metric viscosity coefficient (poises or centipoises) multiply metric scale reading by specific gravity referred to water at 39.1° F.

Note C. For approximate Saybolt Furol seconds, take 0.1 × Saybolt Universal seconds.

Example 1.—Lay off viscosity-temperature lines for a naphthene- and a paraffin-base oil of the same viscosity, 80 Saybolt sec. at 210° F. Extend the line across the chart, Fig. 107. Note their relations to the various systems of oil classification. *Procedure.*—Follow the abscissa through 80 on the vertical scale of Saybolt Universal seconds until it intersects the ordinate through 210° F. at *C*. Follow the diagonal 80 on the S.U.V. scale at 210° F. upward to the right until it intersects the heavy V.I. = 100 line at *D*, and the V.I. = 0 line at *E*. The abscissas through *D* and *E* intersect the ordinate through 100° F. at *B* and *A* respectively. Draw lines through *B* and *C* and *A* and *C*, extending them across the chart. Line *AC* shows viscosity-temperature relation for the naphthene-base oil, and line *BC* shows it for the paraffin-base oil. Both will have the same viscosity, 80 S.U.V., at 210° F. At all temperatures above 210° F., viscosity of the paraffin-base oil will be higher than that of the naphthene-base. At all temperatures below 210° F. viscosity of the naphthene-base oil will exceed that of the paraffin base. The paraffin base oil lies in Navy Classification zone 2310 (on ordinate through 130° F.) Both oils will lie in Navy Classification zone 3080 (on ordinate through 210° F.). At 210° F., both oils lie in S.A.E. zone 50. At 0° F. both oils have viscosity above S.A.E. zone 10–W and 20–W (upper left corner).

Example 2.—Select an oil of viscosity 200 S.U.V. at 100° F. (point *F*, Fig. 107). The viscosity-temperature lines for the naphthene-base and paraffin-base oils satisfying this condition are determined as follows: Follow the abscissa through *F* (S.U.V. = 200) until it intersects the paraffin line V.I. = 100 at *G*, and the naphthene line for V.I. = 0 at *H*. From *G* and *H* follow the diagonals down to the left until they intersect the 200° F. ordinate at *J* and *K* respectively. Draw lines *FJ* and *FK*, extending them across the chart. Line *FJ* shows the viscosity-temperature relation for the paraffin-base oil, and line *FK* for the naphthene-base oil. The paraffin-base oil line crosses the 0° F. ordinate in the lower range of S.A.E. 20–W. At 130° F., it crosses S.A.E. zone 10, and U.S.N. zone 2110. The naphthene line crosses the upper range of S.A.E. 20–W at 0° F., and at 130° F. crosses S.A.E. zone 10 and lies between U.S.N. zones 2110 and 1075.

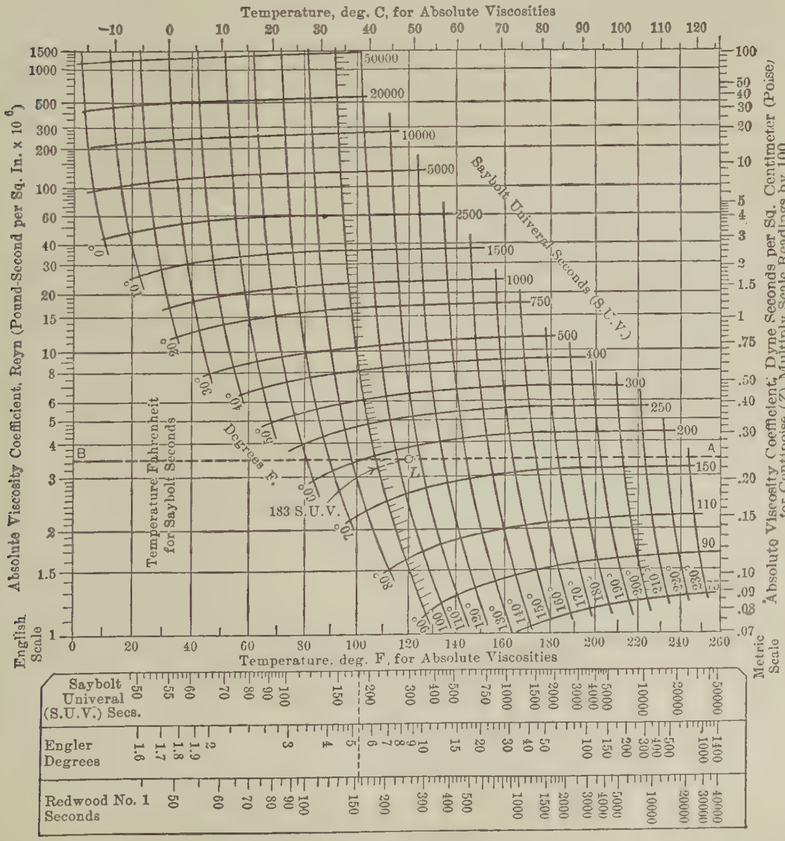

Fig. 108. Viscosity Conversion Chart for Mineral Oil Specification.

A study of Examples 1 and 2 show that if the naphthene and paraffin-base oils are selected at a given temperature to be identical in Saybolt viscosity, the viscosities will differ at temperatures above and below the temperature chosen, irrespective of the temperature used as a basis for comparison.

Saybolt viscosity is not a true measure of the friction-producing quality and load-sustaining value of oil, both of which depend on the specific gravity of the oil as well as its commercial viscosity. The *absolute viscosity* which takes both of these into account, usually appears in the answers to problems in bearing design and lubrication. Absolute viscosity may be expressed in poises in the metric system or Reyns in the English system. See p. 13–44. See notes A and B on Fig. 107, for methods of conversion from one system to the other. Specific gravities used in such conversions must be those of the temperature under consideration, and be expressed in relation to water at 39.1° F. Specific gravities of oils decrease with temperature. At 100° F., 0.865 represents an average value for paraffin-base oils, and 0.915 an average value for naphthene-base oils. Hence the absolute viscosity of two such oils, whose Saybolt viscosities are the same at 100° F., will vary in the ratio of 0.865/0.915. It is possible, however, to find a different temperature at which the absolute viscosities of these two oils are identical, and *vice versa*. That is, oils identical as to viscosity and temperature in the usual commercial units will be identical in absolute viscosity units at some other temperature. Fig. 108 affords a means of working intelligently from one system to the other for mineral oils. The chart is so constructed as to permit oil to be specified in commercial units in order to obtain the desired oil with the absolute viscosity assumed, and at the same time not be concerned as to whether the oil is of one base or the other. The principles underlying this chart are explained in Howarth, Current Practice in Pressure Speeds, Clearances and Lubrication of Oil Film Bearings (20).

Example 3.—Find a Saybolt specification that may be used to obtain mineral oils whose absolute viscosities are the same, 0.24 poise ($3.4 \times 10^{-6}$ Reyns), at 120° F. *Procedure:* In Fig. 108, draw line $AB$ through the value 0.24 on the metric scale, intersecting at $L$ the ordinate of 120° F. (lower scale). Reading on the two curved scales, $L$ represents a temperature of 104° F. and a commercial viscosity of 183 sec. S.U.V. From the scales at the bottom of the chart, 183 sec. S.U.V. corresponds to 5.4 Engler degrees and 158 sec. Redwood viscosimeter.

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# ROLLING CONTACT BEARINGS

## BALL BEARINGS

**By J. S. Tawresey**

In rolling contact or anti-friction bearings the load is supported on a series of rolling elements and rolling friction is substituted for the sliding friction of the plain bearing. Ball bearings support the load on a series of balls, and roller bearings carry the load on rollers having various shapes, according to the particular design of bearing. Ball bearings often are referred to as having point contact between balls and raceways; in the same sense, roller bearings are said to have line contact between rollers and races. These are merely descriptive terms, since the elastic deformation occurring under load results in substantial areas of contact in either type.

The comparative utility of ball and roller bearings is summarized on a qualitative basis, and from the point of view of application as follows:

| | *Ball Bearings* | *Roller Bearings* |
|---|---|---|
| Range of size | Small and medium | Medium and large |
| " " load | Normal and heavy | Heavy and extra heavy |
| " " speed | High and medium | Medium and low |
| Type of service | Normal duty<br>Normal shock | Heavy duty<br>Heavy shock |
| Frictional torque | Lowest | Somewhat higher |

Compared with plain bearings, both ball and roller bearings can sustain heavy overloads for short periods without failure. Their lubrication requirements are moderate; this is especially true for high-speed operation. Starting friction of ball and roller bearings is but slightly greater than running friction, an outstanding advantage in machinery that is required to start frequently under load. Anti-friction bearings are manufactured and applied as interchangeable units, thus greatly simplifying the problem of service replacement when occasion arises.

Ball bearings are available for virtually all types of modern machinery. The primary consideration in their use is the factor of reliability in service, and operation not subject to critical limitations which affect the performance of plain bearings. Ordinarily the failure of a ball bearing is characterized by a relatively long period of gradual impairment of smoothness, affording ample opportunity for detection. Contrasting with this, the plain bearing is disposed to develop an accelerated type of failure, resulting in acute conditions of seizure and shut-down.

### 1. BALL BEARING STANDARDS

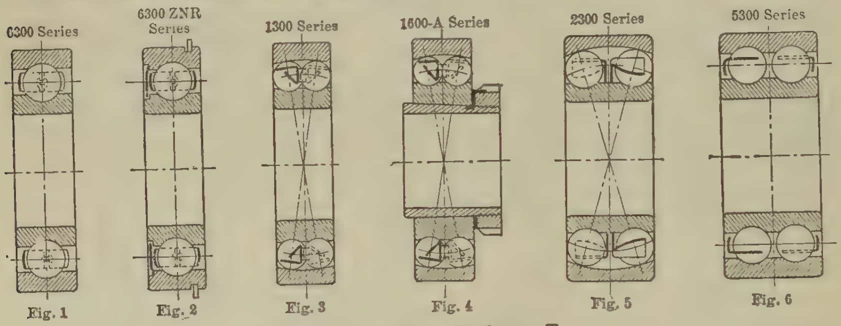
Dimensional and tolerance standards have been adopted for various series and sizes of ball bearings. These are listed in the S.A.E. Handbook of standards and recommended practice. The general acceptance of these standards by both manufacturer and user of bearings gives assurance of their commercial availability. See Tables 1 to 7 for a listing of the most important types and series of single-row and double-row bearings. Dimensions are based on the metric system as a result of bearings first having been manufactured in Europe. Standard tolerances on the bore of the inner race and outside diameter of the outer race are listed in connection with shaft and housing fits, page 13–75.

The manufacture of standard bearings in mass production definitely discourages the use of special and individual designs in which the cost is out of all proportion to that of standard product. In this respect, practice in the use of anti-friction bearings is in marked contrast with that of plain bearings which frequently are designed to meet a specific condition.

### 2. BALL BEARING TYPES

Bearing types are classified readily on the basis of their design and constructional features, leading naturally to a consideration of their functional purpose in terms of load and speed.

**ANNULAR BALL BEARINGS** consist essentially of an inner race, an outer race and a complement of balls spaced by a suitable cage or retainer. They are made in both the single-row, Fig. 1, and double-row, Fig. 6, construction, and may have either a radial contact between balls and raceways, or an inclined angular contact. Radial-contact bearings primarily are intended for radial load, but generally provide for axial stability in both directions where a moderate axial displacement is permissible. Single-row angular-contact bearings, Fig. 7, have a substantial thrust capacity and provide rigid axial support in one direction only. The double-row angular-contact construction has substantial thrust and combined load capacity, together with rigid axial support in both directions. The speed limit of double-row bearings generally is restricted more than that of the single-row, because of features affecting the construction and balance of the cage. The double-row type also is more rigid and sensitive to misalignment than is the single-row.



FIGS. 1–2. Deep-groove Single-row Type
FIGS. 3–5. Self-aligning Type
FIG. 6. Double Row Type
Types of Ball Bearings

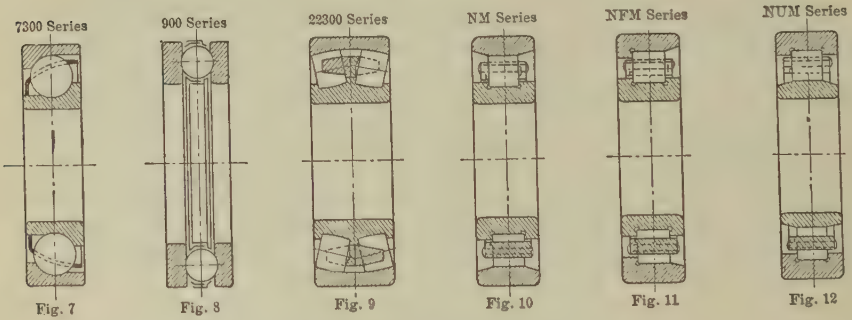


FIG. 7. Angular Contact Type
FIG. 8. Thrust Bearing
FIG. 9. Spherical Roller Bearing
FIGS. 10–12. Cylindrical Roller Bearings
Types of Ball and Roller Bearings

**SELF-ALIGNING RADIAL** bearings, Fig. 3, have a spherical surface to provide for misalignment between inner and outer races. Because of the comparatively small contact angle between balls and raceways, they are primarily radial load carriers and have only sufficient thrust capacity for stabilizing under incidental axial loads. Such bearings made to the wide type standard, Fig. 5, have a substantial thrust capacity due to the greater contact angle. The self-aligning construction has a unique advantage in the case of radial loads which rotate with the shaft, such as those due to mechanical unbalance. Under these circumstances the load is localized on the grooved raceways of the inner ring where the capacity is greater than that provided by the ball complement of the single-row grooved-type bearing. Self-aligning bearings having a single row of balls, and those employing a spherical shell enclosing the outer race manifestly are limited in usefulness and generally inferior to the conventional design.

**THRUST BEARINGS** of the so-called flat type, Fig. 8, have a contact angle of 90 deg., or a load line between ball and raceways that is parallel with the axis of the shaft. The small-bore washer is secured to the shaft, and the large-bore washer is supported by the housing. This type is essentially a one-direction thrust carrier. Thrust in both directions necessitates the use of duplicate units or a bearing of the double direction type. On a horizontal shaft, the weight of the retainer and balls tends to radial displacement of the balls and causes wedging in the grooves. This tendency is opposed by thrust load on the bearing and may be corrected by provision of an adequate pre-load. Thrust bearings are made both with flat rings and spherically-seated rings to allow alignment. In the aligning type, frictional resistance and difficulty of properly centering, makes the spherical seat less attractive in practice than it appears when considered simply as a design feature.

The important characteristics and features of the various ball bearing types can be generally summarized as follows:

**Single Row Radial** (Fig. 1)

Sizes up to 300 mm. bore.
Normal load, combined radial and thrust.
High speed.
Axial displacement permissible under reversing thrust.

**Self-aligning** (Fig. 3)
**Single Row Width**

Predominating radial load, limited thrust capacity.
High speed.
Self-aligning.

**Angular Contact** (Fig. 7)
**Single**

Predominating thrust load, in one direction.
High speed.
Close control of axial displacement.

**Single Direction Thrust** (Fig. 8)

Thrust load only, in one direction.
Limited speed.
Minimum axial displacement.

**Double Row Radial** (Fig. 6)

Sizes up to 110 mm. bore.
Heavy loads, combined radial and thrust.
Medium speed.
Close control of axial position under reversing thrust.

**Self Aligning** (Fig. 5)
**Wide Type**

Predominating radial load, substantial thrust capacity.
High speed.
Self-aligning.

**Angular Contact**
**Duplex**

Predominating thrust load, reversing direction.
High speed.
Close control of axial displacement.

**Double Direction Thrust**

Thrust load only, reversing.
Limited speed.
Minimum axial displacement.

**DIMENSIONAL AND CAPACITY TABLES.**—Tables 1 to 7 show the sizes and load-carrying capacities of the various types of ball bearings. They are compiled from tables issued by S.K.F. Industries, Inc., certain of the larger sizes being omitted. The dimensions, however, agree with the established standards of the S.A.E., and of other manufacturers.

**LOAD CARRYING CAPACITY.**—The ability of a ball bearing to carry shock load and absorb impact depends on the relation between deformation and stress which exists at the load contacts. The capacity of any bearing for sustaining such loads can be judged by the volume of material subjected to stress. It is important to observe that this property is not directly proportional to the normal load rating. For shock loads, the medium and heavy series bearings are far superior to the light series, in sizes having equivalent load ratings.

**MODIFICATION OF BASIC TYPES.**—There are numerous modifications of the basic bearing types, and supplementary provisions have been added to adapt them to certain specialized uses. Among these are the tapered bore adapter sleeve mounting, Fig. 4, the extended inner race set-screw and lock mounting, the plate type shield, the felt washer seal, the outer race snap ring mounting, Fig. 2, and the double seal self-lubricated type. Such specialized types require detailed consideration in order to assure their effective use in any particular design. Aside from an appreciation of elementary principles, experience serves as the best guide in the successful application of bearings to any particular machine. To assist in this respect, the principal bearing manufacturers offer technical advice to prospective users of their product. It is sound policy for engineers and designers to avail themselves of such service.

## 3. DESIGN PRACTICE

**FACTORS GOVERNING SELECTION.**—The type of bearing should be chosen with due regard to the character of load, speed limitation and the need for radial and axial rigidity. The size selected should establish an economic balance between load capacity and service requirements for life of the machine. In many instances, dimensional restrictions affecting shaft size, outside diameter, or width of bearing are controlling factors in

(*Continued on* p. 13-67)

### Table 1.—Dimensions and Load Capacities of Single-row Groove-type Ball Bearings (Fig. 1)

| Bearing No. | d* | | D* | | W* | | R† | Load Reference Rating See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | 6200 Series | 6300 Series | 6400 Series |
| 6200 | 10 | 0.3937 | 30 | 1.1811 | 9 | 0.3543 | 0.024 | 190 | .... | .... |
| 6300 | | | 35 | 1.3780 | 11 | .4331 | .024 | .... | 376 | .... |
| 6201 | 12 | .4724 | 32 | 1.2598 | 10 | .3937 | .024 | 300 | .... | .... |
| 6301 | | | 37 | 1.4567 | 12 | .4724 | .039 | .... | 457 | .... |
| 6202 | 15 | .5906 | 35 | 1.3780 | 11 | .4331 | .024 | 325 | .... | .... |
| 6302 | | | 42 | 1.6535 | 13 | .5118 | .039 | .... | 504 | .... |
| 6203 | 17 | .6693 | 40 | 1.5748 | 12 | .4724 | .039 | 405 | .... | .... |
| 6303 | | | 47 | 1.8504 | 14 | .5512 | .039 | .... | 598 | .... |
| 6403 | | | 62 | 2.4409 | 17 | .6693 | .039 | .... | .... | 1070 |
| 6204 | 20 | .7874 | 47 | 1.8504 | 14 | .5512 | .039 | 545 | .... | .... |
| 6304 | | | 52 | 2.0472 | 15 | .5906 | .039 | .... | 706 | .... |
| 6404 | | | 72 | 2.8346 | 19 | .7480 | .039 | .... | .... | 1440 |
| 6205 | 25 | .9843 | 52 | 2.0472 | 15 | .5906 | .039 | 605 | .... | .... |
| 6305 | | | 62 | 2.4409 | 17 | .6693 | .039 | .... | 925 | .... |
| 6405 | | | 80 | 3.1496 | 21 | .8268 | .059 | .... | .... | 1630 |
| 6206 | 30 | 1.1811 | 62 | 2.4409 | 16 | .6299 | .039 | 815 | .... | .... |
| 6306 | | | 72 | 2.8346 | 19 | .7480 | .039 | .... | 1,222 | .... |
| 6406 | | | 90 | 3.5433 | 23 | .9055 | .059 | .... | .... | 1820 |
| 6207 | 35 | 1.3780 | 72 | 2.8346 | 17 | .6693 | .039 | 1110 | .... | .... |
| 6307 | | | 80 | 3.1496 | 21 | .8268 | .059 | .... | 1,470 | .... |
| 6407 | | | 100 | 3.9370 | 25 | .9843 | .059 | .... | .... | 2360 |
| 6208 | 40 | 1.5748 | 80 | 3.1496 | 18 | .7087 | .039 | 1250 | .... | .... |
| 6308 | | | 90 | 3.5433 | 23 | .9055 | .059 | .... | 1,765 | .... |
| 6408 | | | 110 | 4.3307 | 27 | 1.0630 | .079 | .... | .... | 2770 |
| 6209 | 45 | 1.7717 | 85 | 3.3465 | 19 | 0.7480 | .039 | 1410 | .... | .... |
| 6309 | | | 100 | 3.9370 | 25 | 0.9843 | .059 | .... | 2,280 | .... |
| 6409 | | | 120 | 4.7244 | 29 | 1.1417 | .079 | .... | .... | 3370 |
| 6210 | 50 | 1.9685 | 90 | 3.5433 | 20 | 0.7874 | .039 | 1520 | .... | .... |
| 6310 | | | 110 | 4.3307 | 27 | 1.0630 | .079 | .... | 2,660 | .... |
| 6410 | | | 130 | 5.1181 | 31 | 1.2205 | .079 | .... | .... | 3740 |
| 6211 | 55 | 2.1654 | 100 | 3.9370 | 21 | 0.8268 | .059 | 1850 | .... | .... |
| 6311 | | | 120 | 4.7244 | 29 | 1.1417 | .079 | .... | 3,040 | .... |
| 6411 | | | 140 | 5.5118 | 33 | 1.2992 | .079 | .... | .... | 4350 |
| 6212 | 60 | 2.3622 | 110 | 4.3307 | 22 | 0.8661 | .059 | 2260 | .... | .... |
| 6312 | | | 130 | 5.1181 | 31 | 1.2205 | .079 | .... | 3,480 | .... |
| 6412 | | | 150 | 5.9055 | 35 | 1.3780 | .079 | .... | .... | 4850 |
| 6213 | 65 | 2.5591 | 120 | 4.7244 | 23 | 0.9055 | .059 | 2470 | .... | .... |
| 6313 | | | 140 | 5.5118 | 33 | 1.2992 | .079 | .... | 3,860 | .... |
| 6413 | | | 160 | 6.2992 | 37 | 1.4567 | .079 | .... | .... | 5260 |
| 6214 | 70 | 2.7559 | 125 | 4.9213 | 24 | 0.9449 | .059 | 2670 | .... | .... |
| 6314 | | | 150 | 5.9055 | 35 | 1.3780 | .079 | .... | 4,350 | .... |
| 6414 | | | 180 | 7.0866 | 42 | 1.6535 | .098 | .... | .... | 6510 |
| 6215 | 75 | 2.9528 | 130 | 5.1181 | 25 | 0.9843 | .059 | 2830 | .... | .... |
| 6315 | | | 160 | 6.2992 | 37 | 1.4567 | .079 | .... | 4,780 | .... |
| 6415 | | | 190 | 7.4803 | 45 | 1.7717 | .098 | .... | .... | 7050 |
| 6216 | 80 | 3.1496 | 140 | 5.5118 | 26 | 1.0236 | .079 | 3100 | .... | .... |
| 6316 | | | 170 | 6.6929 | 39 | 1.5354 | .079 | .... | 5,260 | .... |
| 6416 | | | 200 | 7.8740 | 48 | 1.8898 | .098 | .... | .... | 7650 |
| 6217 | 85 | 3.3465 | 150 | 5.9055 | 28 | 1.1024 | .079 | 3535 | .... | .... |
| 6317 | | | 180 | 7.0866 | 41 | 1.6142 | .098 | .... | 5,700 | .... |
| 6417 | | | 210 | 8.2677 | 52 | 2.0472 | .118 | .... | .... | 8150 |
| 6218 | 90 | 3.5433 | 160 | 6.2992 | 30 | 1.1811 | .079 | 4070 | .... | .... |
| 6318 | | | 190 | 7.4803 | 43 | 1.6929 | .098 | .... | 6,250 | .... |
| 6418 | | | 225 | 8.8583 | 54 | 2.1260 | .118 | .... | .... | 8600 |
| 6219 | 95 | 3.7402 | 170 | 6.6929 | 32 | 1.2598 | .079 | 4520 | .... | .... |
| 6319 | | | 200 | 7.8740 | 45 | 1.7717 | .098 | .... | 6,790 | .... |
| 6220 | 100 | 3.9370 | 180 | 7.0866 | 34 | 1.3386 | .079 | 5000 | .... | .... |
| 6320 | | | 215 | 8.4646 | 47 | 1.8504 | .098 | .... | 7,860 | .... |
| 6221 | 105 | 4.1339 | 190 | 7.4803 | 36 | 1.4173 | .079 | 5440 | .... | .... |
| 6321 | | | 225 | 8.8583 | 49 | 1.9291 | .098 | .... | 8,410 | .... |

*See footnotes at end of table.*

*Table continued on following page.*

**Table 1.—Dimensions and Load Capacities of Single-row Groove-type Ball Bearings.—*Continued***

| Bearing No. | d* | | D* | | W* | | R† | Load Reference Rating See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | 6200 Series | 6300 Series | 6400 Series |
| 6222 | 110 | 4.3307 | 200 | 7.8740 | 38 | 1.4961 | 0.079 | 5,970 | .... | .... |
| 6322 | | | 240 | 9.4488 | 50 | 1.9685 | .098 | .... | 9,500 | .... |
| 6224 | 120 | 4.7244 | 215 | 8.4646 | 40 | 1.5748 | .079 | 6,250 | .... | .... |
| 6324 | | | 260 | 10.2362 | 55 | 2.1654 | .098 | .... | 9,500 | .... |
| 6226 | 130 | 5.1181 | 230 | 9.0551 | 40 | 1.5748 | .098 | 6,700 | .... | .... |
| 6326 | | | 280 | 11.0236 | 58 | 2.2835 | .118 | .... | 10,580 | .... |
| 6228 | 140 | 5.5118 | 250 | 9.8425 | 42 | 1.6535 | .098 | 7,240 | .... | .... |
| 6328 | | | 300 | 11.8110 | 62 | 2.4409 | .118 | .... | 11,970 | .... |
| 6230 | 150 | 5.9055 | 270 | 10.6299 | 45 | 1.7717 | .098 | 7,650 | .... | .... |
| 6330 | | | 320 | 12.5984 | 65 | 2.5591 | .118 | .... | 12,800 | .... |
| 6232 | 160 | 6.2992 | 290 | 11.4173 | 48 | 1.8897 | .098 | 8,200 | .... | .... |
| 6332 | | | 340 | 13.3858 | 68 | 2.6772 | .118 | .... | 13,000 | .... |
| 6234 | 170 | 6.6929 | 310 | 12.2047 | 52 | 2.0472 | .118 | 9,500 | .... | |
| 6334 | | | 360 | 14.1732 | 72 | 2.8346 | .118 | .... | 14,600 | .... |
| 6236 | 180 | 7.0866 | 320 | 12.5984 | 52 | 2.0472 | .118 | 10,300 | .... | .... |
| 6336 | | | 380 | 14.9606 | 75 | 2.9528 | .118 | .... | 16,850 | .... |
| 6238 | 190 | 7.4803 | 340 | 13.3858 | 55 | 2.1654 | .118 | 11,700 | .... | .... |
| 6338 | | | 400 | 15.7480 | 78 | 3.0709 | .157 | .... | 17,400 | .... |
| 6240 | 200 | 7.8740 | 360 | 14.1732 | 58 | 2.2834 | .118 | 12,200 | .... | .... |
| 6340 | | | 420 | 16.5354 | 80 | 3.1496 | .157 | .... | 17,950 | .... |
| 6344 | 220 | 8.6614 | 460 | 18.1102 | 88 | 3.4646 | .157 | .... | 19,600 | .... |
| 6348 | 240 | 9.4488 | 500 | 19.6850 | 95 | 3.7402 | .157 | .... | 21,200 | .... |
| 6352 | 260 | 10.2362 | 540 | 21.2598 | 102 | 4.0157 | .197 | .... | 23,900 | .... |

* See Fig. 12.

† $R$ = radius of maximum fillet on shaft or housing that will be cleared by bearing.

**Table 2.—Dimensions and Load Capacities of Self-aligning Ball Bearings (Figs. 3 and 5)**

| Bearing No. | d* | | D* | | W* | | R† | Load Reference Rating See Fig. 15 and p. 13–72 | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in | in. | 1200 Series | 1300 Series | 2200 Series | 2300 Series |
| 1200 | 10 | 0.3937 | 30 | 1.1811 | 9 | 0.3543 | 0.024 | 220 | .... | .... | .... |
| 1300 | | | 35 | 1.3780 | 11 | .4331 | .024 | .... | 295 | .... | .... |
| 2200 | | | 30 | 1.1811 | 14 | .5512 | .024 | .... | .... | 270 | .... |
| 2300 | | | 35 | 1.3780 | 17 | .6693 | .024 | .... | .... | .... | 420 |
| 1201 | 12 | .4724 | 32 | 1.2598 | 10 | .3937 | .024 | 225 | .... | .... | .... |
| 1301 | | | 37 | 1.4567 | 12 | .4724 | .039 | .... | 380 | .... | .... |
| 2201 | | | 32 | 1.2598 | 14 | .5512 | .024 | .... | .... | 300 | .... |
| 2301 | | | 37 | 1.4567 | 17 | .6693 | .039 | .... | .... | .... | 460 |
| 1202 | 15 | .5906 | 35 | 1.3780 | 11 | .4331 | .024 | 315 | .... | .... | .... |
| 1302 | | | 42 | 1.6535 | 13 | .5118 | .039 | .... | 405 | .... | .... |
| 2202 | | | 35 | 1.3780 | 14 | .5512 | .024 | .... | .... | 320 | .... |
| 2302 | | | 42 | 1.6535 | 17 | .6693 | .039 | .... | .... | .... | 500 |
| 1203 | 17 | .6693 | 40 | 1.5748 | 12 | .4724 | .039 | 350 | .... | .... | .... |
| 1303 | | | 47 | 1.8504 | 14 | .5512 | .039 | .... | 545 | .... | .... |
| 2203 | | | 40 | 1.5748 | 16 | .6299 | .039 | .... | .... | 410 | .... |
| 2303 | | | 47 | 1.8504 | 19 | .7480 | .039 | .... | .... | .... | 595 |
| 1204 | 20 | .7874 | 47 | 1.8504 | 14 | .5512 | .039 | 460 | .... | .... | .... |
| 1304 | | | 52 | 2.0472 | 15 | .5906 | .039 | .... | 570 | .... | .... |
| 2204 | | | 47 | 1.8504 | 18 | .7087 | .039 | .... | .... | 545 | .... |
| 2304 | | | 52 | 2.0472 | 21 | .8268 | .039 | .... | .... | .... | 785 |
| 1205 | 25 | .9843 | 52 | 2.0472 | 15 | .5906 | .039 | 580 | .... | .... | .... |
| 1305 | | | 62 | 2.4409 | 17 | .6693 | .039 | .... | 840 | .... | .... |
| 2205 | | | 52 | 2.0472 | 18 | .7087 | .039 | .... | .... | 595 | .... |
| 2305 | | | 62 | 2.4409 | 24 | .9449 | .039 | .... | .... | .... | 1030 |
| 1206 | 30 | 1.1811 | 62 | 2.4409 | 16 | .6299 | .039 | 765 | .... | .... | .... |
| 1306 | | | 72 | 2.8346 | 19 | .7480 | .039 | .... | 1030 | .... | .... |
| 2206 | | | 62 | 2.4409 | 20 | .7874 | .039 | .... | .... | 760 | .... |
| 2306 | | | 72 | 2.8346 | 27 | 1.0630 | .039 | .... | .... | .... | 1360 |

*See footnotes at end of table.*

*Table continued on following page*

**Table 2.—Dimensions and Load Capacities of Self-aligning Ball Bearings.—*Continued***

| Bearing No. | *d** | | *D** | | *W** | | *R*† | Load Reference Rating See Fig. 15 and p. 13–72 | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | 1200 Series | 1300 Series | 2200 Series | 2300 Series |
| 1207 | 35 | 1.3780 | 72 | 2.8346 | 17 | 0.6693 | 0.039 | 850 | .... | .... | .... |
| 1307 | | | 80 | 3.1496 | 21 | 0.8268 | .059 | .... | 1,250 | .... | .... |
| 2207 | | | 72 | 2.8346 | 23 | 0.9055 | .039 | .... | .... | 1085 | .... |
| 2307 | | | 80 | 3.1496 | 31 | 1.2205 | .059 | .... | .... | .... | 1,710 |
| 1208 | 40 | 1.5748 | 80 | 3.1496 | 18 | 0.7087 | .039 | 1075 | .... | .... | .... |
| 1308 | | | 90 | 3.5433 | 23 | 0.9055 | .059 | .... | 1,520 | .... | .... |
| 2208 | | | 80 | 3.1496 | 23 | 0.9055 | .039 | .... | .... | 1170 | .... |
| 2308 | | | 90 | 3.5433 | 33 | 1.2992 | .059 | .... | .... | .... | 2,150 |
| 1209 | 45 | 1.7717 | 85 | 3.3465 | 19 | 0.7480 | .039 | 1220 | .... | .... | .... |
| 1309 | | | 100 | 3.9370 | 25 | 0.9843 | .059 | .... | 1,950 | .... | .... |
| 2209 | | | 85 | 3.3465 | 23 | 0.9055 | .039 | .... | .... | 1280 | .... |
| 2309 | | | 100 | 3.9370 | 36 | 1.4173 | .059 | .... | .... | .... | 2,440 |
| 1210 | 50 | 1.9685 | 90 | 3.5433 | 20 | 0.7874 | .039 | 1330 | .... | .... | .... |
| 1310 | | | 110 | 4.3307 | 27 | 1.0630 | .079 | .... | 2,170 | .... | .... |
| 2210 | | | 90 | 3.5433 | 23 | 0.9055 | .039 | .... | .... | 1330 | .... |
| 2310 | | | 110 | 4.3307 | 40 | 1.5748 | .079 | .... | .... | .... | 2,880 |
| 1211 | 55 | 2.1654 | 100 | 3.9370 | 21 | 0.8268 | .059 | 1530 | .... | .... | .... |
| 1311 | | | 120 | 4.7244 | 29 | 1.1417 | .079 | .... | 2,600 | .... | .... |
| 2211 | | | 100 | 3.9370 | 25 | 0.9843 | .059 | .... | .... | 1550 | .... |
| 2311 | | | 120 | 4.7244 | 43 | 1.6929 | .079 | .... | .... | .... | 3,360 |
| 1212 | 60 | 2.3622 | 110 | 4.3307 | 22 | 0.8661 | .059 | 1820 | .... | .... | .... |
| 1312 | | | 130 | 5.1181 | 31 | 1.2205 | .079 | .... | 2,980 | .... | .... |
| 2212 | | | 110 | 4.3307 | 28 | 1.1024 | .059 | .... | .... | 1930 | .... |
| 2312 | | | 130 | 5.1181 | 46 | 1.8110 | .079 | .... | .... | .... | 3,850 |
| 1213 | 65 | 2.5591 | 120 | 4.7244 | 23 | 0.9055 | .059 | 1880 | .... | .... | .... |
| 1313 | | | 140 | 5.5118 | 33 | 1.2992 | .079 | .... | 3,270 | .... | .... |
| 2213 | | | 120 | 4.7244 | 31 | 1.2205 | .059 | .... | .... | 2420 | .... |
| 2313 | | | 140 | 5.5118 | 48 | 1.8898 | .079 | .... | .... | .... | 4,400 |
| 1214 | 70 | 2.7559 | 125 | 4.9213 | 24 | 0.9449 | .059 | 2090 | .... | .... | .... |
| 1314 | | | 150 | 5.9055 | 35 | 1.3780 | .079 | .... | 3,840 | .... | .... |
| 2214 | | | 125 | 4.9213 | 31 | 1.2205 | .059 | .... | .... | 2500 | .... |
| 2314 | | | 150 | 5.9055 | 51 | 2.0079 | .079 | .... | .... | .... | 4,930 |
| 1215 | 75 | 2.9528 | 130 | 5.1181 | 25 | 0.9843 | .059 | 2370 | .... | .... | .... |
| 1315 | | | 160 | 6.2992 | 37 | 1.4567 | .079 | .... | 4,120 | .... | .... |
| 2215 | | | 130 | 5.1181 | 31 | 1.2205 | .059 | .... | .... | 2600 | .... |
| 2315 | | | 160 | 6.2992 | 55 | 2.1654 | .079 | .... | .... | .... | 5,440 |
| 1216 | 80 | 3.1496 | 140 | 5.5118 | 26 | 1.0236 | .079 | 2500 | .... | .... | .... |
| 1316 | | | 170 | 6.6929 | 39 | 1.5354 | .079 | .... | 4,610 | .... | .... |
| 2216 | | | 140 | 5.5118 | 33 | 1.2992 | .079 | .... | .... | 2820 | .... |
| 2316 | | | 170 | 6.6929 | 58 | 2.2835 | .079 | .... | .... | .... | 5,970 |
| 1217 | 85 | 3.3465 | 150 | 5.9055 | 28 | 1.1024 | .079 | 3000 | .... | .... | .... |
| 1317 | | | 180 | 7.0866 | 41 | 1.6142 | .098 | .... | 5,040 | .... | .... |
| 2217 | | | 150 | 5.9055 | 36 | 1.4173 | .079 | .... | .... | 3360 | .... |
| 2317 | | | 180 | 7.0866 | 60 | 2.3622 | .098 | .... | .... | .... | 6,530 |
| 1218 | 90 | 3.5433 | 160 | 6.2992 | 30 | 1.1811 | .079 | 3420 | .... | .... | .... |
| 1318 | | | 190 | 7.4803 | 43 | 1.6929 | .098 | .... | 5,700 | .... | .... |
| 2218 | | | 160 | 6.2992 | 40 | 1.5748 | .079 | .... | .... | 3950 | .... |
| 2318 | | | 190 | 7.4803 | 64 | 2.5197 | .098 | .... | .... | .... | 7,060 |
| 1219 | 95 | 3.7402 | 170 | 6.6929 | 32 | 1.2598 | .079 | 3840 | .... | .... | .... |
| 1319 | | | 200 | 7.8740 | 45 | 1.7717 | .098 | .... | 6,500 | .... | .... |
| 2219 | | | 170 | 6.6929 | 43 | 1.6929 | .079 | .... | .... | 4620 | .... |
| 2319 | | | 200 | 7.8740 | 67 | 2.6378 | .098 | .... | .... | .... | 7,600 |
| 1220 | 100 | 3.9370 | 180 | 7.0866 | 34 | 1.3386 | .079 | 4200 | .... | .... | .... |
| 1320 | | | 215 | 8.4646 | 47 | 1.8504 | .098 | .... | 7,050 | .... | .... |
| 2220 | | | 180 | 7.0866 | 46 | 1.8110 | .079 | .... | .... | 5260 | .... |
| 2320 | | | 215 | 8.4646 | 73 | 2.8740 | .098 | .... | .... | .... | 8,960 |
| 1221 | 105 | 4.1339 | 190 | 7.4803 | 36 | 1.4173 | .079 | 4520 | .... | .... | .... |
| 1321 | | | 225 | 8.8583 | 49 | 1.9291 | .098 | .... | 7,860 | .... | .... |
| 2221 | | | 190 | 7.4803 | 50 | 1.9685 | .079 | .... | .... | 5700 | .... |
| 2321 | | | 225 | 8.8583 | 77 | 3.0315 | .098 | .... | .... | .... | 9,780 |
| 1222 | 110 | 4.3307 | 200 | 7.8740 | 38 | 1.4961 | .079 | 5150 | .... | .... | .... |
| 1322 | | | 240 | 9.4488 | 50 | 1.9685 | .098 | .... | 8,460 | .... | .... |
| 2222 | | | 200 | 7.8740 | 53 | 2.0866 | .079 | .... | .... | 6520 | .... |
| 2322 | | | 240 | 9.4488 | 80 | 3.1496 | .098 | .... | .... | .... | 10,600 |
| 1324 | 120 | 4.7244 | 260 | 10.2362 | 55 | 2.1654 | .098 | .... | 10,030 | .... | .... |

* See Fig. 13. † *R* = radius of maximum fillet on shaft or housing that will be cleared by bearing.

Table 3.—Dimensions and Load Capacities of Double-row Groove-type Ball Bearings (Fig. 6)

| Bearing No. | d* | | D* | | W* | R† | Load Reference Rating See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | in. | in. | 5200 Series | 5300 Series | 5400 Series |
| 5200 | 10 | 0.3937 | 30 | 0.1811 | 0.5625 | 0.024 | .... | .... | .... |
| 5201 | 12 | .4724 | 32 | 1.2598 | .6250 | .024 | .... | .... | .... |
| 5301 | | | 37 | 1.4567 | .7500 | .039 | .... | 735 | .... |
| 5202 | 15 | .5906 | 35 | 1.3780 | .6250 | .024 | 527 | .... | .... |
| 5302 | | | 42 | 1.6535 | .7500 | .039 | .... | 815 | .... |
| 5203 | 17 | .6693 | 40 | 1.5748 | .6875 | .039 | 651 | .... | .... |
| 5303 | | | 47 | 1.8504 | .8750 | .039 | .... | 980 | .... |
| 5204 | 20 | .7874 | 47 | 1.8504 | .8125 | .039 | 895 | .... | .... |
| 5304 | | | 52 | 2.0472 | .8750 | .039 | .... | 1,140 | .... |
| 5205 | 25 | .9843 | 52 | 2.0472 | .8125 | .039 | 950 | .... | .... |
| 5305 | | | 62 | 2.4409 | 1.0000 | .039 | .... | 1,520 | .... |
| 5405 | | | 80 | 3.1496 | 1.3750 | .059 | .... | .... | 2,610 |
| 5206 | 30 | 1.1811 | 62 | 2.4409 | 0.9375 | .039 | 1360 | .... | .... |
| 5306 | | | 72 | 2.8346 | 1.1875 | .039 | .... | 1,980 | .... |
| 5406 | | | 90 | 3.5433 | 1.5625 | .059 | .... | .... | 3,430 |
| 5207 | 35 | 1.3780 | 72 | 2.8346 | 1.0625 | .039 | 1790 | .... | .... |
| 5307 | | | 80 | 3.1496 | 1.3750 | .059 | .... | 2,340 | .... |
| 5407 | | | 100 | 3.9370 | 1.7500 | .059 | .... | .... | 3,910 |
| 5208 | 40 | 1.5748 | 80 | 3.1496 | 1.1875 | .039 | 2000 | .... | .... |
| 5308 | | | 90 | 3.5433 | 1.4375 | .059 | .... | 2,870 | .... |
| 5408 | | | 110 | 4.3307 | 1.9375 | .079 | .... | .... | 4,450 |
| 5209 | 45 | 1.7717 | 85 | 3.3465 | 1.1875 | .039 | 2275 | .... | .... |
| 5309 | | | 100 | 3.9370 | 1.5625 | .059 | .... | 3,690 | .... |
| 5409 | | | 120 | 4.7244 | 2.1250 | .079 | .... | .... | 5,625 |
| 5210 | 50 | 1.9685 | 90 | 3.5433 | 1.1875 | .039 | 2440 | .... | .... |
| 5310 | | | 110 | 4.3307 | 1.7500 | .079 | .... | 4,280 | .... |
| 5410 | | | 130 | 5.1181 | 2.3125 | .079 | .... | .... | 6,950 |
| 5211 | 55 | 2.1654 | 100 | 3.9370 | 1.3125 | .059 | 2980 | .... | .... |
| 5311 | | | 120 | 4.7244 | 1.9375 | .079 | .... | 1,960 | .... |
| 5411 | | | 140 | 5.5118 | 2.5000 | .079 | .... | .... | 6,950 |
| 5212 | 60 | 2.3622 | 110 | 4.3307 | 1.4375 | .059 | 3640 | .... | .... |
| 5312 | | | 130 | 5.1181 | 2.1250 | .079 | .... | 5,440 | .... |
| 5412 | | | 150 | 5.9055 | 2.6250 | .079 | .... | .... | 7,600 |
| 5213 | 65 | 2.5591 | 120 | 4.7244 | 1.5000 | .059 | 3960 | .... | .... |
| 5313 | | | 140 | 5.5118 | 2.3125 | .079 | .... | 6,250 | .... |
| 5413 | | | 160 | 6.2992 | 2.8125 | .079 | .... | .... | 10,150 |
| 5214 | 70 | 2.7559 | 125 | 4.9213 | 1.5625 | .059 | 4290 | .... | .... |
| 5314 | | | 150 | 5.9055 | 2.5000 | .079 | .... | 6,790 | .... |
| 5414 | | | 180 | 7.0866 | 3.1250 | .098 | .... | .... | 10,150 |
| 5215 | 75 | 2.9528 | 130 | 5.1181 | 1.6250 | .059 | 4550 | .... | .... |
| 5315 | | | 160 | 6.2992 | 2.6875 | .079 | .... | 7,600 | .... |
| 5415 | | | 190 | 7.4803 | 3.2500 | .098 | .... | .... | 11,400 |
| 5216 | 80 | 3.1496 | 140 | 5.5118 | 1.7500 | .079 | 5050 | .... | .... |
| 5316 | | | 170 | 6.6929 | 2.6875 | .079 | .... | 8,410 | .... |
| 5416 | | | 200 | 7.8740 | 3.4375 | .098 | .... | .... | 12,100 |
| 5217 | 85 | 3.3465 | 150 | 5.9055 | 1.9375 | .079 | 5750 | .... | .... |
| 5317 | | | 180 | 7.0866 | 2.8750 | .098 | .... | 9,240 | .... |
| 5417 | | | 210 | 8.2677 | 3.6250 | .118 | .... | .... | 13,000 |
| 5218 | 90 | 3.5433 | 160 | 6.2992 | 2.0625 | .079 | 6520 | .... | .... |
| 5318 | | | 190 | 7.4803 | 2.8750 | .098 | .... | 10,060 | .... |
| 5219 | 95 | 3.7402 | 170 | 6.6929 | 2.1875 | .079 | 7350 | .... | .... |
| 5319 | | | 200 | 7.8740 | 3.0625 | .098 | .... | 10,890 | .... |
| 5220 | 100 | 3.9370 | 180 | 7.0866 | 2.3750 | .079 | 8150 | .... | .... |
| 5320 | | | 215 | 8.4646 | 3.2500 | .098 | .... | 12,500 | .... |
| 5221 | 105 | 4.1339 | 190 | 7.4803 | 2.5625 | .079 | 8960 | .... | .... |
| 5321 | | | 225 | 8.8583 | 3.4375 | .098 | .... | 13,600 | .... |
| 5222 | 110 | 4.3307 | 200 | 7.8740 | 2.7500 | .079 | 9790 | .... | .... |
| 5322 | | | 240 | 9.4488 | 3.6250 | .098 | .... | 15,200 | |

* See Fig. 13. † $R$ = radius of maximum fillet on shaft or housing that will be cleared by bearing.

### Table 4.—Dimensions and Load Capacities of Angular-contact Type Ball Bearings (Fig. 7)

| Bearing No. | d* | | D* | | W* | | R† | Load Reference Rating in Thrust Load. See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | 7200 Series | 7300 Series | 7400 Series |
| 7304 | 20 | 0.7874 | 52 | 2.0472 | 15 | 0.5906 | 0.04 | .... | 1,440 | .... |
| 7205 | 25 | .9843 | 52 | 2.0472 | 15 | .5906 | .04 | 1,200 | .... | .... |
| 7305 | | | 62 | 2.4409 | 17 | .6693 | .04 | .... | 2,020 | .... |
| 7405 | | | 80 | 3.1496 | 21 | .8268 | .06 | .... | .... | 3,400 |
| 7206 | 30 | 1.1811 | 62 | 2.4409 | 16 | .6299 | .04 | 1,740 | .... | .... |
| 7306 | | | 72 | 2.8346 | 19 | .7480 | .06 | .... | 2,570 | .... |
| 7406 | | | 90 | 3.5433 | 23 | .9055 | .08 | .... | .... | 3,950 |
| 7207 | 35 | 1.3780 | 72 | 2.8346 | 17 | .6693 | .04 | 2,270 | .... | .... |
| 7307 | | | 80 | 3.1496 | 21 | .8268 | .06 | .... | 3,070 | ... |
| 7407 | | | 100 | 3.9370 | 25 | .9843 | .08 | .... | .... | 5,050 |
| 7208 | 40 | 1.5748 | 80 | 3.1496 | 18 | .7087 | .06 | 2,740 | .... | .... |
| 7308 | | | 90 | 3.5433 | 23 | .9055 | .08 | .... | 3,720 | .... |
| 7408 | | | 110 | 4.3307 | 27 | 1.0630 | .08 | .... | .... | 5,900 |
| 7209 | 45 | 1.7717 | 85 | 3.3465 | 19 | 0.7480 | .06 | 3,060 | .... | .... |
| 7309 | | | 100 | 3.9370 | 25 | 0.9843 | .08 | .... | 4,860 | .... |
| 7409 | | | 120 | 4.7244 | 29 | 1.1417 | .08 | .... | .... | 7,000 |
| 7210 | 50 | 1.9685 | 90 | 3.5433 | 20 | 0.7874 | .06 | 3,340 | .... | .... |
| 7310 | | | 110 | 4.3307 | 27 | 1.0630 | .08 | .... | 5,650 | .... |
| 7410 | | | 130 | 5.1181 | 31 | 1.2205 | .12 | .... | .... | 7,900 |
| 7211 | 55 | 2.1654 | 100 | 3.9370 | 21 | 0.8268 | .06 | 3,940 | .... | ... |
| 7311 | | | 120 | 4.7244 | 29 | 1.1417 | .08 | .... | 6,580 | .... |
| 7411 | | | 140 | 5.5118 | 33 | 1.2992 | .12 | .... | .... | 9,050 |
| 7212 | 60 | 2.3622 | 110 | 4.3307 | 22 | 0.8661 | .08 | 4,700 | .... | .... |
| 7312 | | | 130 | 5.1181 | 31 | 1.2205 | .12 | .... | 7,500 | .... |
| 7412 | | | 150 | 5.9055 | 35 | 1.3780 | .12 | .... | .... | 9,900 |
| 7213 | 65 | 2.5591 | 120 | 4.7244 | 23 | 0.9055 | .08 | 5,350 | .... | .... |
| 7313 | | | 140 | 5.5118 | 33 | 1.2992 | .12 | .... | 8,320 | .... |
| 7413 | | | 160 | 6.2992 | 37 | 1.4567 | .12 | .... | .... | 10,700 |
| 7214 | 70 | 2.7559 | 125 | 4.9213 | 24 | 0.9449 | .08 | 5,860 | .... | .... |
| 7314 | | | 150 | 5.9055 | 35 | 1.3780 | .12 | .... | 9,100 | .... |
| 7414 | | | 180 | 7.0866 | 42 | 1.6535 | .12 | .... | .... | 14,300 |
| 7215 | 75 | 2.9528 | 130 | 5.1181 | 25 | 0.9843 | .08 | 6,050 | .... | .... |
| 7315 | | | 160 | 6.2992 | 37 | 1.4567 | .12 | .... | 10,000 | .... |
| 7415 | | | 190 | 7.4803 | 45 | 1.7717 | .14 | .... | .... | 14,800 |
| 7216 | 80 | 3.1496 | 140 | 5.5118 | 26 | 1.0236 | .08 | 6,900 | .... | .... |
| 7316 | | | 170 | 6.6929 | 39 | 1.5354 | .12 | .... | 11,200 | .... |
| 7416 | | | 200 | 7.8740 | 48 | 1.8898 | .14 | .... | .... | 15,800 |
| 7217 | 85 | 3.3465 | 150 | 5.9055 | 28 | 1.1024 | .08 | 7,600 | .... | .... |
| 7317 | | | 180 | 7.0866 | 41 | 1.6142 | .12 | .... | 12,000 | .... |
| 7417 | | | 210 | 8.2677 | 52 | 2.0472 | .16 | .... | .... | 17,000 |
| 7218 | 90 | 3.5433 | 160 | 6.2992 | 30 | 1.1811 | .08 | 8,900 | .... | .... |
| 7318 | | | 190 | 7.4803 | 43 | 1.6929 | .14 | .... | 13,300 | .... |
| 7418 | | | 225 | 8.8583 | 54 | 2.1260 | .16 | .... | .... | 20,200 |
| 7219 | 95 | 3.7402 | 170 | 6.6929 | 32 | 1.2598 | .12 | 9,800 | .... | .... |
| 7319 | | | 200 | 7.8740 | 45 | 1.7717 | .14 | .... | 14,600 | .... |
| 7419 | | | 240 | 9.4488 | 55 | 2.1654 | .16 | .... | .... | 22,600 |
| 7220 | 100 | 3.9370 | 180 | 7.0866 | 34 | 1.3386 | .12 | 10,700 | .... | .... |
| 7320 | | | 215 | 8.4646 | 47 | 1.8504 | .14 | .... | 16,500 | .... |
| 7420 | | | 250 | 9.8425 | 58 | 2.2835 | .16 | .... | .... | 23,800 |
| 7221 | 105 | 4.1339 | 190 | 7.4803 | 36 | 1.4173 | .12 | 11,600 | .... | .... |
| 7321 | | | 225 | 8.8583 | 49 | 1.9291 | .14 | .... | 17,900 | .... |
| 7222 | 110 | 4.3307 | 200 | 7.8740 | 38 | 1.4961 | .12 | 12,900 | .... | .... |
| 7322 | | | 240 | 9.4488 | 50 | 1.9685 | .14 | .... | 19,800 | .... |
| 7224 | 120 | 4.7244 | 215 | 8.4646 | 40 | 1.5748 | .08 | 15,200 | .... | .... |
| 7324 | | | 260 | 10.2362 | 55 | 2.1654 | .10 | .... | 24,000 | .... |
| 7226 | 130 | 5.1181 | 230 | 9.0551 | 40 | 1.5748 | .10 | 17,200 | .... | .... |
| 7326 | | | 280 | 11.0236 | 58 | 2.2835 | .12 | .... | 27,000 | .... |
| 7228 | 140 | 5.5118 | 250 | 9.8425 | 42 | 1.6535 | .10 | 18,700 | .... | .... |
| 7328 | | | 300 | 11.8110 | 62 | 2.4409 | .12 | .... | 29,600 | .... |

* See Fig. 13. † $R$ = radius of maximum fillet on shaft or housing that will be cleared by bearing.

**Table 4.—Dimensions and Load Capacities of Angular-contact Type Ball Bearings.**—*Continued*

| Bearing No. | d* | | D* | | W* | | R† | Load Reference Rating in Thrust Load. See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | 7200 Series | 7300 Series | 7400 Series |
| 7230 | 150 | 5.9055 | 270 | 10.6299 | 45 | 1.7716 | 0.10 | 21,600 | .... | .... |
| 7330 | | | 320 | 12.5984 | 65 | 2.5591 | .12 | .... | 32,400 | .... |
| 7232 | 160 | 6.2992 | 290 | 11.4173 | 48 | 1.8897 | .10 | 23,000 | .... | .... |
| 7332 | | | 340 | 13.3858 | 68 | 2.6772 | .12 | .... | 33,600 | .... |
| 7234 | 170 | 6.6929 | 310 | 12.2047 | 52 | 2.0472 | .12 | 26,000 | .... | .... |
| 7334 | | | 360 | 14.1732 | 72 | 2.8346 | .12 | .... | 38,200 | .... |
| 7236 | 180 | 7.0866 | 320 | 12.5984 | 52 | 2.0472 | .12 | 27,000 | .... | .... |
| 7336 | | | 380 | 14.9606 | 75 | 2.9528 | .12 | .... | 41,500 | .... |
| 7238 | 190 | 7.4803 | 340 | 13.3858 | 55 | 2.1654 | .12 | 29,200 | .... | .... |
| 7338 | | | 400 | 15.7480 | 78 | 3.0709 | .16 | .... | 44,500 | .... |
| 7240 | 200 | 7.8740 | 360 | 14.1732 | 58 | 2.2834 | .12 | 32,600 | .... | .... |
| 7340 | | | 420 | 16.5354 | 80 | 3.1496 | .16 | .... | 46,000 | .... |

* See Fig. 13. † $R$ = radius of maximum fillet on housing or shaft that will be cleared by bearing.

(*Continued from* p. 13–61)

deciding the bearing size. Ordinarily a rather wide range of bearing sizes is available to meet given conditions of load and speed; in such cases it is wise to determine which types and sizes show an advantage in the matter of cost.

Primary importance attaches to the proportioning of the bearing to the average load. Unlike practice with plain bearings, it is not desirable to base design on maximum or extreme conditions. The service life of an anti-friction bearing ultimately is determined by the *mean effective*, rather than the maximum, load, and a large reserve capacity is available for overload of short duration. The attention of the designer should be directed to making a good estimate of the average load conditions, and from such data bearing life may be estimated with a comparable degree of accuracy.

**LOAD-LIFE RATING.**—The load-life rating of ball bearings has been developed empirically from extensive series of tests, in which the lives of very many bearings were observed when operated under known and controlled conditions. This investigation disclosed the fact that ball bearings ultimately show a typical fatigue deterioration of the raceway surfaces. The repetition of contact stresses between raceways and the loaded balls induces a progressive system of surface cracks, and continued operation results in flaking or spalling of the surface. The rough and eroded appearance of the surfaces is characteristic of this action. Fatigue constitutes an inherent limitation to normal useful life of a bearing and an understanding of the factors involved affords the basis on which bearing life is predetermined.

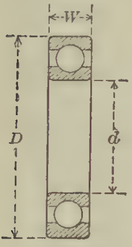


Fig. 13

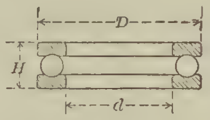


Fig. 14

In considering the life of a bearing it should be noted that faulty design practice may lead to premature failure as the result of intrusion of dirt, water or other destructive foreign material. Neglect of lubricant supply, and other forms of abuse, may unnecessarily impair bearing life. These conditions may be avoided or corrected by the exercise of proper judgment in design and suitable control in maintenance. When a bearing is selected, of type and size adapted to the operating conditions, and due consideration is given to the details of mounting design and installation, the service performance can be accurately forecast by means of the life-load rating.

A group of bearings identical in design and manufacture and tested under the same conditions are found to show a characteristic deviation among the individual lives. This dispersion is common to many types of endurance phenomena, including such diverse entities as human mortality, and lives of electric lamps, gear teeth and valve springs. The average life of the whole group tested serves as the best description of the endurance quality of the group, but in addition, it is important to define the sequence of failure, especially that of the shortest lives in relation to the group average. Extensive tests have shown that 90% of the bearings attain a life exceeding one-fifth of the average life, or that 10% of the bearings may develop fatigue at a life less than one-fifth of the average. Experience in the engineering and practical application of ball bearings has shown that the life exceeded by 90% of the bearings is a useful criterion of reliability, while the average life of the group serves as a direct measure of productive performance. This conception of bearing life employs a statistical viewpoint, wherein the individual bearing loses its identity and estimates of life refer to a group as a whole.

**Endurance-Load Relations.**—The endurance properties of materials are conventionally investigated on the basis of the *S–N* diagram wherein the intensity of stress *S* is related to the number of stress cycles *N*. Sufficient data will serve to indicate the existence

**Table 5.—Dimensions and Load Capacities of Ball Thrust Bearings*** (Fig. 8)

| Bearing No. | *d*† | | *D*† | | *H*† | | *R*‡ | Load Reference Rating in Thrust Load. See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | 2900 Series | 900 Series | 3900 Series |
| 2900 | 10 | 0.3937 | 26 | 1.0236 | 12 | 0.472 | 0.04 | 400 | .... | .... |
| 900 | | | 25 | 0.9843 | 12 | .472 | .02 | .... | 410 | .... |
| 2901 | 12 | .4724 | 28 | 1.1024 | 12 | .472 | .04 | 425 | .... | .... |
| 2902 | 15 | .5906 | 31 | 1.2205 | 12 | .472 | .04 | 470 | .... | .... |
| 902 | | | 35 | 1.3780 | 14 | .551 | .02 | .... | 700 | .... |
| 2903 | 18 | .7087 | 35 | 1.3780 | 12 | .472 | .04 | 520 | .... | .... |
| 2904 | 20 | .7874 | 37 | 1.4567 | 12 | .472 | .04 | 540 | .... | .... |
| 2904 1/2 | 22 | .8661 | 42 | 1.6535 | 14 | .551 | .04 | 810 | .... | .... |
| 904 | 20 | .7874 | 40 | 1.5748 | 14 | .551 | .02 | .... | 780 | ... |
| 2905 | 25 | .9843 | 45 | 1.7717 | 14 | .551 | .04 | 855 | .... | .... |
| 905 | | | 48 | 1.8898 | 15.5 | .610 | .02 | .... | 1020 | .... |
| 3905 | | | 52 | 2.0472 | 16 | .630 | .06 | .... | .... | 1010 |
| 2906 | 30 | 1.1811 | 50 | 1.9685 | 14 | .551 | .04 | 950 | .... | .... |
| 906 | | | 53 | 2.0866 | 15.5 | .610 | .02 | .... | 1110 | .... |
| 3906 | | | 60 | 2.3622 | 19 | .748 | .06 | .... | .... | 1390 |
| 2907 | 35 | 1.3780 | 55 | 2.1654 | 16 | .630 | .04 | 1160 | .... | .... |
| 907 | | | 62 | 2.4409 | 18 | .709 | .02 | .... | 1530 | .... |
| 3907 | | | 68 | 2.6772 | 22 | .866 | .06 | .... | .... | 1880 |
| 2908 | 40 | 1.5748 | 60 | 2.3622 | 16 | .630 | .04 | 1250 | .... | .... |
| 908 | | | 64 | 2.5197 | 18 | .709 | .02 | .... | 1400 | .... |
| 3908 | | | 76 | 2.9921 | 25 | .984 | .06 | .... | .... | 2360 |
| 2909 | 45 | 1.7717 | 68 | 2.6772 | 16 | .630 | .04 | 1300 | .... | .... |
| 909 | | | 73 | 2.8740 | 22 | .866 | .02 | .... | 1950 | .... |
| 3909 | | | 85 | 3.3465 | 28 | 1.102 | .06 | .... | .... | 2920 |
| 2910 | 50 | 1.9685 | 74 | 2.9134 | 18 | 0.709 | .04 | 1680 | .... | .... |
| 910 | | | 78 | 3.0709 | 22 | 0.866 | .02 | .... | 2220 | .... |
| 3910 | | | 92 | 3.6220 | 31 | 1.221 | .06 | .... | .... | 3550 |
| 2911 | 55 | 2.1654 | 78 | 3.0709 | 18 | 0.709 | .04 | 1780 | .... | .... |
| 911 | | | 88 | 3.4646 | 24.5 | 0.965 | .02 | .... | 2700 | .... |
| 3911 | | | 100 | 3.9370 | 33 | 1.299 | .06 | .... | .... | 4040 |
| 2912 | 60 | 2.3622 | 82 | 3.2283 | 18 | 0.709 | .04 | 1820 | .... | .... |
| 912 | | | 90 | 3.5433 | 24.5 | 0.965 | .02 | .... | 2500 | .... |
| 3912 | | | 106 | 4.1732 | 35 | 1.378 | .08 | .... | .... | 4380 |
| 2913 | 65 | 2.5591 | 90 | 3.5433 | 20 | 0.787 | .04 | 2050 | .... | .... |
| 913 | | | 100 | 3.9370 | 27 | 1.063 | .04 | .... | 3060 | .... |
| 3913 | | | 112 | 4.4094 | 36 | 1.417 | .08 | .... | .... | 4800 |
| 2914 | 70 | 2.7559 | 95 | 3.7902 | 20 | 0.787 | .04 | 2120 | .... | .... |
| 914 | | | 103 | 4.0551 | 27 | 1.063 | .04 | .... | 3200 | .... |
| 3914 | | | 120 | 4.7244 | 38 | 1.496 | .08 | .... | .... | 5400 |
| 2915 | 75 | 2.9528 | 100 | 3.9370 | 20 | 0.787 | .04 | 2230 | .... | .... |
| 915 | | | 110 | 4.3307 | 27 | 1.063 | .04 | .... | 3330 | .... |
| 3915 | | | 128 | 5.0394 | 41 | 1.614 | .08 | .... | .... | 6000 |
| 2916 | 80 | 3.1496 | 110 | 4.3307 | 22 | 0.866 | .04 | 2500 | .... | .... |
| 916 | | | 115 | 4.5276 | 29 | 1.142 | .04 | .... | 3600 | .... |
| 3916 | | | 136 | 5.3543 | 44 | 1.732 | .08 | .... | .... | 6680 |
| 2917 | 85 | 3.3465 | 115 | 4.5276 | 22 | 0.866 | .04 | 2600 | .... | .... |
| 917 | | | 125 | 4.9213 | 30.5 | 1.201 | .04 | .... | 4600 | .... |
| 3917 | | | 145 | 5.7087 | 47 | 1.850 | .08 | .... | .... | 7500 |
| 2918 | 90 | 3.5433 | 120 | 4.7244 | 22 | 0.866 | .04 | 2780 | .... | .... |
| 918 | | | 135 | 5.3150 | 30.5 | 1.201 | .04 | .... | 5400 | .... |
| 3918 | | | 155 | 6.1024 | 50 | 1.969 | .08 | .... | .... | 8350 |

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* Thrust bearings have not been conclusively standardized (1937). This table gives the standards of S.K.F. Industries, Inc. † See Fig. 14.

‡ *R* = radius of maximum fillet on shaft or housing that will be cleared by bearing.

of any fatigue limit, or that stress below which progressive failure will not occur. Endurance properties of bearings have been investigated in the same manner, but it is found more convenient to deal with the external load applied to the bearing, and the total number of revolutions attained prior to the appearance of fatigue, as the related variables. A given point on the raceway is subjected to a cycle of stress each time that it comes into contact with a ball in the loaded zone. The fundamental relation between life and load is that the total number of revolutions, before the inception of fatigue, will vary inversely as the cube of the applied load. This basic relation is independent of the speed at which

**Table 6.—Dimensions and Load Capacities of Spherical Roller Bearings** (Fig. 9)

| Bearing No. | *d** |  | *D** |  | *W** |  | *R*† | Load Reference Rating. See Fig. 15 and p. 13–72 |  |
|---|---|---|---|---|---|---|---|---|---|
|  | mm. | in. | mm. | in. | mm. | in. | in. | 22200 Series | 22300 Series |
| 22308 | 40 | 1.5748 | 90 | 3.5433 | 33 | 1.299 | 0.059 | ..... | 3,580 |
| 22309 | 45 | 1.7717 | 100 | 3.9370 | 36 | 1.417 | .059 | ..... | 4,500 |
| 22310 | 50 | 1.9685 | 110 | 4.3307 | 40 | 1.575 | .079 | ..... | 6,250 |
| 22311 | 55 | 2.1654 | 120 | 4.7244 | 43 | 1.692 | .079 | ..... | 7,340 |
| 22312 | 60 | 2.3622 | 130 | 5.1181 | 46 | 1.811 | .079 | ..... | 8,950 |
| 22313 | 65 | 2.5591 | 140 | 5.5118 | 48 | 1.890 | .079 | ..... | 9,500 |
| 22314 | 70 | 2.7559 | 150 | 5.9055 | 51 | 2.008 | .079 | ..... | 12,500 |
| 22315 | 75 | 2.9528 | 160 | 6.2992 | 55 | 2.165 | .079 | ..... | 13,020 |
| 22216 | 80 | 3.1496 | 140 | 5.5118 | 33 | 1.299 | .079 | 5,300 | ..... |
| 22316 |  |  | 170 | 6.6929 | 58 | 2.283 | .079 | ..... | 15,200 |
| 22217 | 85 | 3.3465 | 150 | 5.9055 | 36 | 1.417 | .079 | 6,820 | ..... |
| 22317 |  |  | 180 | 7.0866 | 60 | 2.362 | .098 | ..... | 16,810 |
| 22218 | 90 | 3.5433 | 160 | 6.2992 | 40 | 1.575 | .079 | 8,640 | ..... |
| 22318 |  |  | 190 | 7.4803 | 64 | 2.520 | .098 | ..... | 20,100 |
| 22219 | 95 | 3.7402 | 170 | 6.6929 | 43 | 1.693 | .079 | 10,300 | ..... |
| 22319 |  |  | 200 | 7.8740 | 67 | 2.637 | .098 | ..... | 21,400 |
| 22220 | 100 | 3.9370 | 180 | 7.0866 | 46 | 1.811 | .079 | 12,000 | ..... |
| 22320 |  |  | 215 | 8.4646 | 73 | 2.874 | .098 | ..... | 26,300 |
| 22222 | 110 | 4.3307 | 200 | 7.8740 | 53 | 2.087 | .079 | 15,300 | ..... |
| 22322 |  |  | 240 | 9.4488 | 80 | 3.150 | .098 | ..... | 31,500 |
| 22224 | 120 | 4.7244 | 215 | 8.4646 | 58 | 2.283 | .079 | 18,800 | ..... |
| 22324 |  |  | 260 | 10.2362 | 86 | 3.386 | .098 | ..... | 38,000 |
| 22226 | 130 | 5.1181 | 230 | 9.0551 | 64 | 2.520 | .098 | 23,600 | ..... |
| 22326 |  |  | 280 | 11.0236 | 93 | 3.661 | .118 | ..... | 44,000 |
| 22228 | 140 | 5.5118 | 250 | 9.8425 | 68 | 2.677 | .098 | 27,200 | ..... |
| 22328 |  |  | 300 | 11.8110 | 102 | 4.016 | .118 | ..... | 48,300 |
| 22230 | 150 | 5.9055 | 270 | 10.6299 | 73 | 2.874 | .098 | 30,400 | ..... |
| 22330 |  |  | 320 | 12.5984 | 108 | 4.252 | .118 | ..... | 54,400 |
| 22232 | 160 | 6.2992 | 290 | 11.4173 | 80 | 3.150 | .098 | 36,900 | ..... |
| 22332 |  |  | 340 | 13.3858 | 114 | 4.488 | .118 | ..... | 59,800 |
| 22234 | 170 | 6.6929 | 310 | 12.2047 | 86 | 3.386 | .118 | 40,700 | ..... |
| 22334 |  |  | 360 | 14.1732 | 120 | 4.724 | .118 | ..... | 68,000 |
| 22236 | 180 | 7.0866 | 320 | 12.5984 | 86 | 3.386 | .118 | 41,800 | ..... |
| 22336 |  |  | 380 | 14.9606 | 126 | 4.961 | .118 | ..... | 73,400 |
| 22238 | 190 | 7.4803 | 340 | 13.3858 | 92 | 3.622 | .118 | 46,700 | ..... |
| 22338 |  |  | 400 | 15.7480 | 132 | 5.197 | .157 | ..... | 81,500 |
| 22240 | 200 | 7.8740 | 360 | 14.1732 | 98 | 3.858 | .118 | 52,000 | ..... |
| 22340 |  |  | 420 | 16.5354 | 138 | 5.433 | .157 | ..... | 87,000 |
| 22244 | 220 | 8.6614 | 400 | 15.7480 | 108 | 4.252 | .118 | 64,600 | ..... |
| 22344 |  |  | 460 | 18.1102 | 145 | 5.709 | .157 | ..... | 103,000 |
| 22248 | 240 | 9.4488 | 440 | 17.3228 | 120 | 4.724 | .118 | 80,000 | ..... |
| 22348 |  |  | 500 | 19.6850 | 155 | 6.102 | .157 | ..... | 119,400 |
| 22252 | 260 | 10.2362 | 480 | 18.8976 | 130 | 5.118 | .157 | 93,500 | ..... |
| 22352 |  |  | 540 | 21.2598 | 165 | 6.496 | .197 | ..... | 138,900 |
| 22256 | 280 | 11.0236 | 500 | 19.6850 | 130 | 5.118 | .157 | 98,700 | ..... |
| 22356 |  |  | 580 | 22.8346 | 175 | 6.890 | .197 | ..... | 157,800 |
| 22260 | 320 | 11.8110 | 540 | 21.2598 | 140 | 5.512 | .157 | 117,000 | ..... |
| 22264 | 300 | 12.5984 | 580 | 22.8346 | 150 | 5.906 | .157 | 133,000 | ..... |

* See Fig. 13.

† *R* = radius of maximum fillet on shaft or housing that will be cleared by bearing.

the bearing is run, so that life, measured in hours, will vary inversely as the speed in rev. per min. Endurance testing of bearings has failed to reveal any counterpart of the fatigue limit normally disclosed by a series of conventional material tests. Loads of very small

**Table 7.—Dimensions and Load Capacities of Cylindrical Roller Bearings** (Figs. 10–12)

| Bearing No. | d* | | D* | | W* | | R* | Load Reference Rating. See Fig. 15 and p. 13–72 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | mm. | in. | mm. | in. | mm. | in. | in. | NL Series | NM Series | NS Series |
| NL 20 | 20 | 0.7874 | 47 | 1.8504 | 14 | 0.5512 | 0.039 | 545 | .... | .... |
| NM 20 | | | 52 | 2.0472 | 15 | .5906 | | .... | 815 | .... |
| NL 25 | 25 | .9843 | 52 | 2.0472 | 15 | .5906 | .039 | 620 | .... | .... |
| NM 25 | | | 62 | 2.4409 | 17 | .6693 | .039 | .... | 1,035 | .... |
| NS 25 | | | 80 | 3.1496 | 21 | .8268 | .059 | .... | .... | 1,740 |
| NL 30 | 30 | 1.1811 | 62 | 2.4409 | 16 | .6299 | .039 | 815 | .... | .... |
| NM 30 | | | 72 | 2.8346 | 19 | .7480 | .039 | .... | 1,370 | .... |
| NS 30 | | | 90 | 3.5433 | 23 | .9055 | .059 | .... | .... | 2,370 |
| NL 35 | 35 | 1.3780 | 72 | 2.8346 | 17 | .6693 | .039 | 1180 | .... | .... |
| NM 35 | | | 80 | 3.1496 | 21 | .8268 | .059 | .... | 1,780 | .... |
| NS 35 | | | 100 | 3.9370 | 25 | .9843 | .059 | .... | .... | 2,920 |
| NL 40 | 40 | 1.5748 | 80 | 3.1496 | 18 | .7087 | .039 | 1550 | .... | .... |
| NM 40 | | | 90 | 3.5433 | 23 | .9055 | .059 | .... | 2,100 | .... |
| NS 40 | | | 110 | 4.3307 | 27 | 1.0630 | .079 | .... | .... | 3,780 |
| NL 45 | 45 | 1.7717 | 85 | 3.3465 | 19 | 0.7480 | .039 | 1625 | .... | .... |
| NM 45 | | | 100 | 3.9370 | 25 | .9843 | .059 | .... | 2,880 | .... |
| NS 45 | | | 120 | 4.7244 | 29 | 1.1417 | .079 | .... | .... | 4,500 |
| NL 50 | 50 | 1.9685 | 90 | 3.5433 | 20 | 0.7874 | .039 | 1700 | .... | .... |
| NM 50 | | | 110 | 4.3307 | 27 | 1.0630 | .079 | .... | 3,500 | .... |
| NS 50 | | | 130 | 5.1181 | 31 | 1.2205 | .079 | .... | .... | 5,550 |
| NL 55 | 55 | 2.1654 | 100 | 3.9370 | 21 | 0.8268 | .059 | 2070 | .... | .... |
| NM 55 | | | 120 | 4.7244 | 29 | 1.1417 | .079 | .... | 4,220 | .... |
| NS 55 | | | 140 | 5.5118 | 33 | 1.2992 | .079 | .... | .... | 5,550 |
| NL 60 | 60 | 2.3622 | 110 | 4.3307 | 22 | 0.8661 | .059 | 2590 | .... | .... |
| NM 60 | | | 130 | 5.1181 | 31 | 1.2205 | .079 | .... | 4,800 | .... |
| NS 60 | | | 150 | 5.9055 | 35 | 1.3780 | .079 | .... | .... | 6,750 |
| NL 65 | 65 | 2.5591 | 120 | 4.7244 | 23 | 0.9055 | .059 | 3030 | .... | .... |
| NM 65 | | | 140 | 5.5118 | 33 | 1.2992 | .079 | .... | 5,340 | .... |
| NS 65 | | | 160 | 6.2992 | 37 | 1.4567 | .079 | .... | .... | 7,760 |
| NL 70 | 70 | 2.7559 | 125 | 4.9213 | 24 | 0.9449 | .059 | 3030 | .... | .... |
| NM 70 | | | 150 | 5.9055 | 35 | 1.3780 | .079 | .... | 6,200 | .... |
| NS 70 | | | 180 | 7.0866 | 42 | 1.6535 | .098 | .... | .... | 10,000 |
| NL 75 | 75 | 2.9528 | 130 | 5.1181 | 25 | 0.9843 | .059 | 3600 | .... | .... |
| NM 75 | | | 160 | 6.2992 | 37 | 1.4567 | .079 | .... | 7,400 | .... |
| NS 75 | | | 190 | 7.4803 | 45 | 1.7717 | .098 | .... | .... | 10,700 |
| NL 80 | 80 | 3.1496 | 140 | 5.5118 | 26 | 1.0236 | .079 | 4000 | .... | .... |
| NM 80 | | | 170 | 6.6929 | 39 | 1.5354 | .079 | .... | 7,400 | .... |
| NS 80 | | | 200 | 7.8740 | 48 | 1.8898 | .098 | .... | .... | 12,600 |
| NL 85 | 85 | 3.3465 | 150 | 5.9055 | 28 | 1.1024 | .079 | 4600 | .... | .... |
| NM 85 | | | 180 | 7.0866 | 41 | 1.6142 | .098 | .... | 8,880 | .... |
| NS 85 | | | 210 | 8.2677 | 52 | 2.0472 | .118 | .... | .... | 14,100 |
| NL 90 | 90 | 3.5433 | 160 | 6.2992 | 30 | 1.1811 | .079 | 5750 | .... | .... |
| NM 90 | | | 190 | 7.4803 | 43 | 1.6929 | .098 | .... | 9,600 | .... |
| NS 90 | | | 225 | 8.8583 | 54 | 2.1260 | .118 | .... | .... | 15,900 |
| NL 95 | 95 | 3.7402 | 170 | 6.6929 | 32 | 1.2598 | .079 | 6500 | .... | .... |
| NM 95 | | | 200 | 7.8740 | 45 | 1.7717 | .098 | .... | 11,100 | .... |
| NL 100 | 100 | 3.9370 | 180 | 7.0866 | 34 | 1.3386 | .079 | 7180 | .... | .... |
| NM 100 | | | 215 | 8.4646 | 47 | 1.8504 | .098 | .... | 12,600 | .... |
| NL 105 | 105 | 4.1339 | 190 | 7.4803 | 36 | 1.4173 | .079 | 7750 | .... | .... |
| NM 105 | | | 225 | 8.8583 | 49 | 1.9291 | .098 | .... | 14,800 | .... |
| NL 110 | 110 | 4.3307 | 200 | 7.8740 | 38 | 1.4961 | .079 | 9250 | .... | .... |
| NM 110 | | | 240 | 9.4488 | 50 | 1.9685 | .098 | .... | 16,650 | .... |
| NM 120 | 120 | 4.7244 | 260 | 10.2362 | 55 | 2.1654 | .098 | .... | 20,000 | .... |
| NM 130 | 130 | 5.1181 | 280 | 11.0236 | 58 | 2.2835 | .118 | .... | 23,700 | .... |
| NM 140 | 140 | 5.5118 | 300 | 11.8110 | 62 | 2.4409 | .118 | .... | 26,200 | .... |
| NM 150 | 150 | 5.9055 | 320 | 12.5984 | 65 | 2.5591 | .118 | .... | 28,800 | .... |

* See Fig. 13.

† *R* = radius of maximum fillet on shaft or housing that will be cleared by bearing.

intensity will result in extremely long life, but fatigue ultimately will develop and progressively become more apparent.

The ball bearing industry has made an important contribution to the art of machine design in that bearing life is recognized as a statistically determinant quantity, and fatigue is reckoned with as a predetermined factor subject to definite laws.

## 4. SELECTION OF BEARING SIZE

Selection of the proper bearing size is effected by equating a measure of the operating conditions to a corresponding measure of the bearing performance. Operating conditions are defined in terms of the bearing *load* and shaft *speed.* Performance is expressed in terms of the *reference rating* and the *design life.* The equation between the two requirements is:

$$P \times \sqrt[3]{N} = R \times (1/\sqrt[3]{H}) \times 100 \quad . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

where $P$ = load, lb.; $N$ = speed, r.p.m.; $R$ = reference rating, lb.; $H$ = design life, hours. It will be observed that this affords a direct solution of the problem of selecting bearing size, expressed in terms of the quantities that are naturally involved. This design practice supersedes former methods which employed ambiguous factors in an effort to define standards of performance.

Reference rating values for various types of bearings are given in the Tables 1 to 7, arranged according to series and size. The basis for the reference rating is explained in detail on p. 13–72.

When bearing load $P$ and speed $N$ are given, corresponding values of reference rating and design life are found as follows:

**Reference Rating** $R$ required for a design life $H$ is

$$R = P \times \sqrt[3]{N} \times (\sqrt[3]{H}/100) \quad . \; . \; . \; . \; . \; . \; . \quad [2]$$

**Design Life** $H$ obtained with a given reference rating $R$ is

$$H = \{1/(P^3 \times N)\} \times R^3 \times 1{,}000{,}000 \quad . \; . \; . \; . \; . \; . \; . \quad [3]$$

With given values of reference rating $R$ and design life $H$, permissible load or speed is found.

**Load** $P$ which can be carried at a given speed $N$ is

$$P = (R/\sqrt[3]{H}) \times (100/\sqrt[3]{N}) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [4]$$

**Speed** $N$ permissible for a given load $P$ (speed not to exceed limit of r.p.m. for given bearing) is

$$N = (R^3/H) \times (1{,}000{,}000/P^3) \quad . \; . \; . \; . \; . \; . \; . \; . \quad [5]$$

**BEARING SELECTION CHART.**—A graphical solution of these relations is given by the chart, Fig. 15, which will be found sufficiently precise for preliminary design and checking purposes. In using the chart it is important to note that corresponding values of *load* and *speed*, and corresponding values of *reference rating* and *design life*, must be taken together in entering or reading from the diagram.

Design life and its relation to average service life for a group of bearings is discussed on p. 13–74.

**Use of the Chart.**—The chart can be used for the selection of bearing size. For this purpose, corresponding values of load and speed are located on the $P$ and $N$ scales, from which the chart is entered to locate the diagonal guide line through their intersection. This guide line then defines corresponding values of reference rating and design life under the given conditions of load and speed. The reference rating is the key to the bearing size in using the diagram.

When the chart is employed to determine permissible load or speed, it is entered from corresponding values of reference rating and design life located on the $R$ and $H$ scales. The intersection of these given values locates the diagonal guide line which will define the permissible combinations of load and speed, as read from the $P$ and $N$ scales.

### Load

The load used for design purposes is based on an analysis of the reactions affecting the bearing, and is reduced to an estimate of the average radial component and thrust components. Factors of safety should not be included in such calculations, as this defeats the purpose of determining the design life and corresponding average service life.

Combined thrust and radial load is converted to an equivalent radial load by:

$$P = P_R + Y \times A \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [6]$$

where $P$ = equivalent radial load, lb.; $P_R$ = radial load component, lb.; $A$ = thrust or axial load component, lb.; $Y$ = factor, converting thrust to radial load, and depending on type and size of bearing. Values of the $Y$ factor are as follows for various types of bearings:

**Single-row Deep-groove Bearings.**—For high speed, $Y = 1.5$. For low speed, $Y = 1.0$.

**Double-row Angular-contact Bearings.**— $Y = 1.1$.

**Self-aligning Ball Bearings.**—Medium series 1300 to 1303, $Y = 2.5$; 1304 to 1306, $Y = 3.0$; 1307 to 1324, $Y = 3.5$. Light series, wide type, 2200 to 2206, $Y = 2.0$; 2207 to 2209, $Y = 2.5$; 2210 to 2222, $Y = 3.0$. Medium series, wide type 2300 to 2304, $Y = 1.5$; 2305 to 2372, $Y = 2.0$.

**Spherical Roller Bearings.**—22216 to 22264, $Y = 3.0$; 22308 to 22356, $Y = 2.0$.

**AVERAGE LOAD.**—When a bearing is subjected to a varying load, this can be reduced to an equivalent *mean effective load* by an averaging procedure. For this purpose the varying operating conditions should be expressed in terms of load and speed, and the corresponding percentage of time, or of the total life, during which they apply. Using subscripts to designate the several operating conditions, the mean effective load and corresponding average speed is determined as follows:

$$N_\epsilon = \Sigma\, N_1\phi_1 + N_2\phi_2 + N_3\phi_3 + \ldots N_n\phi_n \quad [7]$$

$$P_\epsilon = \sqrt[3]{\frac{1}{N_\epsilon}\Sigma\, P_1{}^3N_1\phi_1 + P_2{}^3N_2\phi_2 + P_3{}^3N_3\phi_3 + \ldots P_n{}^3N_n\phi_n} \quad [8]$$

where $N_\epsilon$ = effective or average speed, r.p.m.; $P_\epsilon$ = effective or average load, lb.; $P$ = load, lb.; $N$ = speed, r.p.m.; $\phi$ = percent of time expressed as a decimal; subscripts 1, 2, 3 . . . $n$ indicate the several given operating conditions.

Values of $P_\epsilon$ and $N_\epsilon$, determined as above, can be used in the selection of bearing size with the assurance that the design life and average service life, so estimated, will be exactly the same as that due to the varying load conditions.

The chart, Fig. 15, can be used to obtain a graphical solution of the selection of bearing size for a varying load. In this case the auxiliary scales, at the bottom and to the left, marked "scale of $(R)^3$" and "percentage of time" are used. The chart is entered with corresponding values of load and speed for one of the operating conditions, which locates a diagonal guide line. The chart then is entered with the given percentage of time and from the intersection with the same diagonal, the corresponding value of $(R)^3$ is determined. The value of $(R)^3$ is found in this way for each of the operating conditions and the sum of these values determined. The chart is then entered with this sum on the $(R)^3$ scale and the 100% ordinate from the "percent of time" scale, fixing the location of a final diagonal guide line. This diagonal defines corresponding values of reference rating and design life, corresponding exactly with the varying load condition as originally defined.

Care should be taken in noting the position of the decimal point in reading from the $(R)^3$ scale, and in checking the individual values for percent of time, to add up to 100%.

## Speed

The life-load relation does not apply for speeds under 15 r.p.m. The maximum speed at which a bearing may be operated depends on the type and size, and especially on the material and construction of the cage or retainer. It will be considerably influenced by the provision made for lubrication. A speed limit line is shown on the chart, Fig. 15. This may be taken as applying directly to single-row ball bearings. For other bearing types, it is advisable to consult with the bearing manufacturer when speeds are high, as suggested by proximity to the limit line of the chart.

When the speed varies, an average value as determined by equation [7] should be employed for design purposes.

The maximum static load which may be applied to a bearing can be approximately determined by taking the speed $N$ as 15 r.p.m. and the design life $H$ as 500 hr.

## 5. REFERENCE RATING

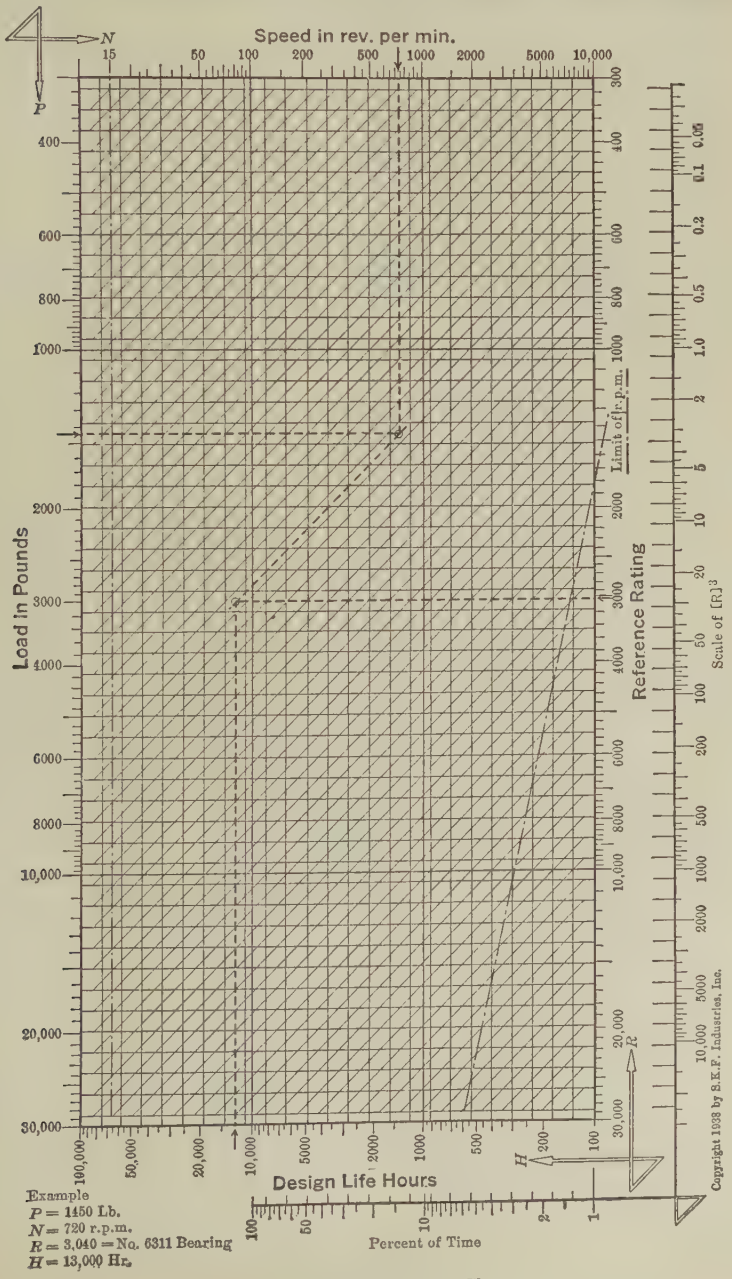
**REFERENCE RATING** is the load in pounds, which applied to the bearing at a speed of 1000, r.p.m., will result in a design life of 1000 hr., or an average service life of 5000 hr.

The actual load carrying capacity of various types of anti-friction bearings cannot be inferred by the indiscriminate comparison of catalog ratings. There is a conspicuous lack of uniformity in the rating practice followed by different manufacturers in published references to their product. It is essential that a uniform basis be established to enable a true comparison to be made of the load-carrying ability of various types, series and sizes of bearings. The reference rating employed herein has been developed for that purpose.

The reference rating serves as a measure of the specific load carrying capacity of a given bearing. It can be calculated from a knowledge of the design features and an empirical constant determined for each particular type of bearing. The general equation for ball bearings is:

$$R = C \times Z^{2/3} \times \{D^2/(1 + 0.508D)\} \cos \alpha \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

where $R$ = reference rating, lb.; $C$ = an empirical constant dependent on design and



Example
$P$ = 1450 Lb.
$N$ = 720 r.p.m.
$R$ = 3,040 = No. 6311 Bearing
$H$ = 13,000 Hr.

FIG. 15. Bearing Selection Chart

material, and established by endurance tests; $Z$ = number of balls per row; $D$ = diameter of ball, in.; $\alpha$ = contact angle, measured from a plane normal to the shaft axis.

For single-row deep-groove bearings, Table 1, and self-aligning ball bearings, Table 2, this equation becomes:

$$R = 1630\,Z^{2/3}\{D^2/(1 + 0.508\,D)\}\cos\alpha \quad [10]$$

For double-row angular-contact bearings, Table 3,

$$R = 2900\,Z^{2/3}\{D^2/(1 + 0.508\,D)\}\cos\alpha \quad [11]$$

For single-row angular-contact bearings, rated in thrust load, Table 4,

$$R = 4200\,Z^{2/3}\{D^2/(1 + 0.508\,D)\}\sin\alpha \quad [12]$$

Thrust type bearings, Table 5, are rated in thrust load, and

$$R = 2175\,Z^{2/3}\{D^2/(1 + 0.508\,D)\} \quad [13]$$

The reference rating is stated in terms of radial load for radial bearings, and thrust load for single-row angular-contact and thrust-type bearings. The rating applies to the normal condition of stationary housing and rotating shaft, in which the radial load is distributed over the inner race and localized on the outer race. Under the reverse condition, where the load is distributed around the outer race and localized on the inner race, the reference rating must be multiplied by a correction factor as follows: Self-aligning ball-bearing, correction factor = 0.90; other types of radial bearing, correction factor = 0.75.

The reference rating is the key value serving to relate a particular bearing size to the problem of load, speed and design life.

## 6. DESIGN LIFE AND SERVICE LIFE

**DESIGN LIFE** is a measure of bearing performance that is particularly useful in design practice. It is the number of hours for which at least 90% of a group of bearings will operate before the inception of fatigue. Practical experience has proven this proportion to be a valuable guide in estimating the reliability of a given design.

**SERVICE LIFE** is the average number of hours for which the entire group of bearings will operate prior to the inception of fatigue. It is equal to five times the number of hours comprising the design life. The service life is an average value representing the economical performance of a group of bearings and serving as a criterion of bearing replacement.

### Examples of Bearing Selection

Example 1.—Blower equipment is to be installed in a power substation where continuous operation is contemplated and a high degree of reliability is essential because of cost and inconvenience of a shut-down. Past experience has shown an average service life of 8 years to be a satisfactory basis for proportioning the bearings in such equipment.

*Operating conditions* are: $P_R$ = radial load on bearing = 950 lb.; $A$ = thrust load on bearing = 400 lb.

The equivalent radial load is determined by applying the appropriate $Y$ factor (see p. 13–71). Because of the substantial thrust component it is decided to use the deep-groove type of bearing. The $Y$ factor is taken as 1.25, intermediate between the values for low and high speed. Then:

$P$ = equivalent radial load = 950 + (1.25 × 400) = 1450 lb.; $N$ = given speed = 720 r.p.m.

*Performance Requirements.*—The designated average service life of 8 years, for continuous operation, is expressed in hours = 8 × 365 × 24 = 72,080 hr.

Design life $H$, to be used for the bearing selection, is 1/5 of the average service life, whence $H$ = 72,080/5 = 14,420 hr. The reference rating $R$, required under these conditions, is then found from equation [2], from which

$$R = 1450 \times \sqrt[3]{720} \times (\sqrt[3]{14{,}420}/100) = 3171 \text{ lb.}$$

Referring to the ratings for single row groove type bearings, Table 1, it is seen that the conditions are met by the following bearings:

| Bearing No. | Bore, in. | Outside Diam., in. | Width, in. | Reference Rating |
|---|---|---|---|---|
| 6216 | 3.1496 | 5.5118 | 1.0236 | 3100 |
| 6311 | 2.1654 | 4.7244 | 1.1417 | 3040 |
| 6409 | 1.7717 | 4.7244 | 1.1417 | 3370 |

The final selection of bearing size then depends on shaft size and outside diameter requirements, imposed by the design.

Example 2.—A double-row groove-type of bearing with a bore of about 2 in. is to support a radial load, rotating with the shaft. Speed $N$ is 720 r.p.m. Design life $H$ is established at 14,000 hr. in consideration of the particular service requirements of the equipment. Required the permissible radial load for the bearing size selected.

A No. 5310 bearing (see Table 3) has a bore of 1.9685 in. and a reference rating of 4280 lb. The correction factor to account for the rotating character of load (see above) is 0.75. Therefore, the rating to be used is $R$ = 4280 × 0.75 = 3210.

The permissible load is found by equation [4], whence

$$P = (3210/\sqrt[3]{14{,}000}) \times (100/\sqrt[3]{720}) = 1486 \text{ lb.}$$

Example 3.—A No. 6312 single-row groove-type bearing is subjected to a variable load cycle, as follows, expressed in load, speed, and percent of time:

| | | | |
|---|---|---|---|
| $P_1 = 2000$ lb. | $P_2 = 1000$ lb. | $P_3 = 500$ lb. | $P_4 = 0$ |
| $N_1 = 1000$ r.p.m. | $N_2 = 2000$ r.p.m. | $N_3 = 3000$ r.p.m. | $N_4 = 0$ |
| $\phi = 25\%$ | $\phi_2 = 50\%$ | $\phi_3 = 10\%$ | $\phi_4 = 15\%$ |

The average load conditions equivalent to this cycle are determined by equations [7] and [8]. From equation [7],

$$N_\epsilon = (1000 \times 0.25) + (2000 \times 0.50) + (3000 \times 0.10) = 1550 \text{ r.p.m.}$$

From equation [8],

$$P_\epsilon = \sqrt[3]{(1/1550)\{2000^3 \times 1000 \times 0.25) + (1000^3 \times 2000 \times 0.50) + (500^3 \times 3000 \times 0.10)} = 1251 \text{ lb.}$$

The design life of the No. 6312 bearing under this variable load cycle then is determined by substituting in equation [3], taking the values of $P_\epsilon$ and $N_\epsilon$ as the values of $P$ and $N$ in the equation. From Table [1], $R = 3480$. Then

$$H = \{1/(1251^3 \times 1550)\} \times 3480^3 \times 1{,}000{,}000 = 13{,}900 \text{ hr.}$$

$$13{,}900 \times 5 = 69{,}500 \text{ hr.} = \text{average service life.}$$

This same problem is solved by using the chart, Fig. 15. Entering the chart with load 2,000 and speed 1,000, the $R^3$ value for 25% of the time is noted as 20. Corresponding values for loads and speeds at 50% and 10% of the time are 9.5 and 0.35, respectively. The sum of these $R^3$ values is 29.85.

Entering the chart from the $R^3$ scale at 29.85 and 100% on the percent of time scale determines the final guide diagonal giving corresponding values of rating and life. For a reference rating of 3480 lb. (No. 6312 bearing) the design life is read as 14,000 hr.

Example 4.—A spherical roller bearing, size 22332, is to be used to support a shaft rotating at 500 r.p.m. It is desired to determine the load that may be applied for an average service life of five years, operating 4,000 hr. per year.

The average service life in hours is equal to 5 × 4,000 = 20,000 hr. Design life $H = 20{,}000/5 = 4{,}000$ hr.

The permissible load is found from equation [4], the reference rating for bearing No. 22332 being (Table 6) 59,800 lb.

$$P = (59{,}800/\sqrt[3]{4000}) \times (100/\sqrt[3]{500}) = 47{,}500 \text{ lb.}$$

Using the chart, the reference rating 59,800 is divided by 10 to give 5,980 and the load corresponding to speed 500 r.p.m. is read as 4,800. The load then is multiplied by 10 to give 48,000 lb. This procedure is equivalent to moving the decimal point simultaneously on the scale of reference rating and on the scale of load.

## 7. PRACTICE IN MOUNTING BALL BEARINGS

**TOLERANCES.**—Ball bearings are manufactured to close tolerances on both internal and external dimensions. The degree of accuracy observed is in no sense an elaborate or unnecessary refinement. A ball bearing is essentially a statically indeterminant structure in which the externally applied load is internally reacted by a series of individual ball contacts. Under these circumstances, the magnitude of the several ball reactions is determined by the combined clearances and deformations existing at the contacts. Uniformity in this respect can be obtained only when a high degree of accuracy is observed in the manufacture of all component parts. This will insure a proper distribution of the load among the balls, and smooth running will result if corresponding care is exercised in the mounting of the bearing.

The unit of measurement and tolerance generally used in the manufacture of bearings is 0.0001 in. Precision of external dimension is necessary to allow accurate fitting and the requisite interchangeability when a bearing is installed. This implies a comparable precision in the tolerances for shaft seat diameter and housing bore. Both inner and outer ring will, through elastic deformation, tend to assume the size and shape of the mating machine part. This tendency is more pronounced in the case of interference or press fits. Inner races will expand measurably to take up the initial internal clearance of the bearing, and outer races will contract with a similar effect, that may be exaggerated when the housing bore is out of round. The combination of shaft and housing fit should be selected with due regard for assembly and disassembly requirements and a consideration of their effect on the internal clearance of the bearing.

**FITS.**—A general principle to be observed in the mounting of ball bearings is the use of an interference or press fit for securing the ring, either inner or outer, which rotates with respect to the direction of the radial load. This is necessitated by the tendency of that ring to progressively creep on its support and to develop wear that is aggravated by a peening effect. Frictional constraint, resulting from a press fit of the ring, is the most effective means of preventing this action, although suitable types of lock-nut and washer are useful as a supplementary provision. The tendency to creep is greater under heavy load and shock, and under these conditions relatively heavier fits should be used. With moderate loads, in the absence of vibration and shock, lighter fits are sufficient, especially when supplemented by a lock-nut. Formerly it was believed that some advantage would accrue by allowing the race that is stationary with respect to the radial load, to have a so-called creeping fit. This view is now discredited by an understanding of the principles affecting fatigue deterioration of the bearing, and practical difficulties encountered in preventing a creeping ring from spinning in the presence of an unbalanced load component.

Insufficient clearance or a tight internal fit generally is conducive to heating of the bearing in operation, and to rough running. The heavier classes of shaft fit will result in normally fitted bearings being internally tight after mounting. However, when a bearing is subjected to heavy load, the accompanying deflection at the load-carrying contacts will serve to relieve the effect of initial tightness. This principle is involved in the action of the so-called preloaded type of bearing, which is particularly useful in meeting exacting requirements of precision, and in minimizing displacement of the bearing under load. With bearings selected for application to precision machinery, it is desirable to employ light press fits in order to avoid disturbing the initial internal fit of the bearing.

The procedure to be followed in assembly and disassembly of the machine parts must be considered in deciding upon the shaft and housing fits to be used. Ordinarily but one bearing on a given shaft should be located axially and all other bearings on the same shaft should have endwise float permitting axial adjustment under thrust load and temperature gradient. The race, either inner or outer, which provides the float naturally must not be the one for which a press fit is required in consideration of the radial load.

To properly meet the various requirements encountered in practice, it is necessary to employ a number of different classes of shaft and housing fit, adapted to the tolerances and the internal fit of bearings. This classification should provide a graduated sequence of fits, adjusted to give a uniform variation, and permitting machining and grinding tolerances suited to design and manufacturing requirements. Such a schedule developed specifically for use with ball and roller bearings is given in Table 8 for shaft fits, and Table 9 for housing fits. Table 10 is a metric conversion table for bearing bores and outside diameters.

**General Requirements.**—Radial loads acting in a fixed direction, require an interference or press fit between shaft and inner race when the shaft rotates, and between the housing and outer race when the housing rotates.

**Table 8.—Shaft Fits and Tolerances**

| | | | | | Push Fit | | | Light Press | | | Press | | | Heavy Press | | | Heavy Press | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Rotating Housing * and Inner Race Having Axial Float on Shaft<br>Automobile Front Wheels and Loose Pulleys | | | Rotating Shaft.† Bearing Lightly Loaded Permits Easy Assembly and Disassembly<br>Machine Tool and Fractional Horsepower Motors, etc., Thrust Bearings in General | | | Rotating Shaft.† Bearing Normally Loaded<br>Ball Bearing Applications in General, Automotive and Industrial Equipment | | | Rotating Shaft.† Bearing Heavily Loaded and Where Shock Conditions Prevail<br>Intended for Larger Sizes of Ball Bearings and Smaller Sizes of Cylindrical and Spherical Roller Bearings | | | Rotating Shaft.† Roller Bearings Heavily Loaded Together with Shock Conditions<br>Intended for Heavy Machinery in General, where Cylindrical or Spherical Roller Bearings are Used | | |
| | | | All Dimensions in Ten Thousandths of an Inch (0.0001 in.) | | | | | | | | | | | | | | | | |
| Nominal Size | | | Bearing Bore Nominal Size | | Shaft Seat Nominal Size | | Average Clearance | Shaft Seat Nominal Size | | Average Interference | Shaft Seat Nominal Size | | Average Interference | Shaft Seat Nominal Size | | Average Interference | Shaft Seat Nominal Size | | Average Interference |
| Over | Including | | | | | | | | | | | | | | | | | | |
| mm. | mm. | in. | Plus | Minus | Minus | Minus | | Plus | Minus | | Plus | Plus | | Plus | Plus | | Plus | Plus | |
| 3 | 6 | 0.2362 | 0 | 3 | 2 | 5 | 2 | .... | ...... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 6 | 10 | .3937 | 0 | 3 | 2 | 6 | 3 | 2 | 1 | 2 | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 10 | 18 | .7087 | 0 | 3 | 2 | 7 | 3 | 2 | 1 | 2 | 4 | 0 | 4 | .... | .... | .... | .... | .... | .... |
| 18 | 30 | 1.1811 | 0 | 4 | 3 | 8 | 4 | 2 | 2 | 2 | 4 | 0 | 4 | 7 | 3 | 7 | .... | .... | .... |
| 30 | 50 | 1.9685 | 0 | 5 | 4 | 10 | 5 | 2 | 2 | 3 | 5 | 1 | 6 | 8 | 4 | 9 | 13 | 7 | 13 |
| 50 | 80 | 3.1496 | 0 | 6 | 4 | 11 | 5 | 2 | 3 | 3 | 6 | 1 | 7 | 9 | 4 | 10 | 15 | 8 | 15 |
| 80 | 120 | 4.7244 | 0 | 8 | 5 | 13 | 5 | 2 | 4 | 3 | 7 | 1 | 8 | 11 | 5 | 12 | 18 | 9 | 18 |
| 120 | 180 | 7.0866 | 0 | 10 | 6 | 15 | 6 | .... | ...... | .... | 8 | 1 | 10 | 13 | 6 | 15 | 20 | 11 | 21 |
| 180 | 250 | 9.8425 | 0 | 12 | 6 | 17 | 6 | .... | ...... | .... | .... | .... | .... | 15 | 7 | 17 | 24 | 12 | 24 |
| 250 | 315 | 12.4016 | 0 | 14 | 7 | 19 | 6 | | ...... | .... | .... | .... | .... | 17 | 8 | 20 | 26 | 13 | 27 |

* *Rotating Housing* refers to a loading condition in which the outer race rotates with respect to the direction of the load. † *Rotating Shaft* refers to a loading condition in which the inner race rotates with respect to the direction of the load.

Radial loads not fixed in direction, but rotating with a moving part, reverse the requirements. An interference fit is used with the outer race for a rotating shaft, and for the inner race with a rotating housing.

A radial load that is the resultant of two components, one fixed in direction and one rotating, requires a compromise in the selection of fits. This should favor the predominating component.

Rotating loads are induced by mechanical unbalance or may result directly from the interaction of machine forces.

The numerical designation used for describing the fit is the mean value of the clearance or interference, measured in units of 0.0001 in.

The quantitative character of any particular class of fit is best judged from an estimate of its average value. The frequency distribution of production parts, both bearings and machine elements, follow characteristic patterns. A negligible percentage of pieces is

**Table 9.—Housing Fits and Tolerances**

| | | | | | Drive Fit | | | Hard Push Fit | | | Push Fit | | | Sliding Fit | | | Loose Fit | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Rotating Housing * and Inner Race Having Axial Float on Shaft. Shock Load Present<br>Automobile Front Wheels and Loose Pulleys | | | Rotating Housing * or Rotating Shaft. Bearing Subjected to Unbalanced Loads and Outer Race Tending to Creep<br>Crank-shafts, Eccentrics, etc. | | | Rotating Shaft.† Bearing Normally Loaded in High Speed Precision Machinery or Heavily Loaded at Low Speed<br>Machine Tools, Electric Motors, etc. | | | Rotating Shaft.† Bearing Applications in General, Both Automatic and Industrial Equipment | | | Rotating Shaft.† Bearings on Line-shaft and Transmission Equipment. Low Speed Without Particular Precision Requirements | | |
| | | | All Dimensions in Ten Thousandths of an Inch (0.0001 in.) | | | | | | | | | | | | | | | | |
| Nominal Size | | | Bearing O. D. Nominal Size | | Housing Bore Nominal Size | | Average Interference | Housing Bore Nominal Size | | Average Clearance | Housing Bore Nominal Size | | Average Clearance | Housing Bore Nominal Size | | Average Clearance | Housing Bore Nominal Size | | Average Clearance |
| Over | Including | | | | | | | | | | | | | | | | | | |
| mm. | mm. | in. | Plus | Minus | Minus | Minus | | Plus | Minus | | Plus | Minus | | Plus | Minus | | Plus | Minus | |
| 10 | 18 | 0.7087 | 0 | 4 | ...... | ...... | .... | .... | ...... | .... | 2 | 2 | 2 | 7 | 0 | 6 | 11 | 0 | 7 |
| 18 | 30 | 1.1811 | 0 | 4 | 4 | 9 | 5 | 1 | 4 | 1 | 3 | 2 | 3 | 8 | 0 | 6 | 13 | 0 | 9 |
| 30 | 50 | 1.9685 | 0 | 4 | 5 | 11 | 6 | 2 | 5 | 1 | 4 | 2 | 4 | 10 | 0 | 7 | 15 | 0 | 10 |
| 50 | 80 | 3.1496 | 0 | 5 | 6 | 13 | 7 | 2 | 6 | 1 | 5 | 2 | 4 | 12 | 0 | 9 | 18 | 0 | 12 |
| 80 | 120 | 4.7244 | 0 | 6 | 6 | 15 | 8 | 2 | 7 | 1 | 6 | 2 | 5 | 14 | 0 | 10 | 21 | 0 | 14 |
| 120 | 150 | 5.9055 | 0 | 7 | 7 | 16 | 8 | 2 | 7 | 1 | 7 | 3 | 6 | 16 | 0 | 11 | 25 | 0 | 16 |
| 150 | 180 | 7.0866 | 0 | 10 | 8 | 18 | 8 | 2 | 9 | 2 | 7 | 3 | 7 | 16 | 0 | 13 | 25 | 0 | 18 |
| 180 | 250 | 9.8425 | 0 | 12 | 9 | 20 | 9 | 2 | 9 | 3 | 9 | 3 | 9 | 18 | 0 | 15 | 28 | 0 | 20 |
| 250 | 315 | 12.4016 | 0 | 14 | 10 | 22 | 9 | 2 | 11 | 4 | 10 | 3 | 11 | 20 | 0 | 17 | 32 | 0 | 23 |
| 315 | 400 | 15.7480 | 0 | 16 | 10 | 24 | 9 | 3 | 11 | 4 | 11 | 3 | 12 | 22 | 0 | 19 | 35 | 0 | 26 |
| 400 | 500 | 19.6850 | 0 | 18 | 11 | 26 | 10 | 3 | 13 | 4 | 13 | 3 | 14 | 25 | 0 | 22 | .... | .... | .... |

* *Rotating Housing* refers to a loading condition in which the outer race rotates with respect to the direction of the load.

† *Rotating Shaft* refers to a loading condition in which the inner race rotates with respect to the direction of the load.

**Table 10.—Metric Conversion Table for Bearing Bores and Outside Diameters**

| Millimeters | Inch | Millimeters | Inches | Millimeters | Inches | Millimeters | Inches | Millimeters | Inches |
|---|---|---|---|---|---|---|---|---|---|
| 5 | 0.1969 | 24 | 0.9449 | 62 | 2.4409 | 140 | 5.5118 | 270 | 10.6299 |
| 6 | .2362 | 25 | .9843 | 65 | 2.5591 | 150 | 5.9055 | 280 | 11.0236 |
| 7 | .2756 | 26 | 1.0236 | 70 | 2.7559 | 160 | 6.2992 | 290 | 11.4173 |
| 8 | .3150 | 30 | 1.1811 | 72 | 2.8346 | 170 | 6.6929 | 300 | 11.8110 |
| 9 | .3543 | 32 | 1.2598 | 75 | 2.9528 | 180 | 7.0866 | 310 | 12.2047 |
| 10 | .3937 | 35 | 1.3780 | 80 | 3.1496 | 190 | 7.4803 | 320 | 12.5984 |
| 11 | .4331 | 37 | 1.4567 | 85 | 3.3465 | 200 | 7.8740 | 340 | 13.3858 |
| 12 | .4724 | 40 | 1.5748 | 90 | 3.5433 | 210 | 8.2677 | 360 | 14.1732 |
| 13 | .5118 | 42 | 1.6535 | 95 | 3.7402 | 215 | 8.4646 | 380 | 14.9606 |
| 14 | .5512 | 45 | 1.7717 | 100 | 3.9370 | 220 | 8.6614 | 400 | 15.7480 |
| 15 | .5906 | 47 | 1.8504 | 105 | 4.1339 | 225 | 8.8583 | 420 | 16.5354 |
| 17 | .6693 | 50 | 1.9685 | 110 | 4.3307 | 230 | 9.0551 | 440 | 17.3228 |
| 19 | .7480 | 52 | 2.0472 | 120 | 4.7244 | 240 | 9.4488 | 460 | 18.1102 |
| 20 | .7874 | 55 | 2.1654 | 125 | 4.9213 | 250 | 9.8425 | 480 | 18.8976 |
| 22 | .8661 | 60 | 2.3622 | 130 | 5.1181 | 260 | 10.2362 | 500 | 19.6850 |

found at either limit of the working tolerance and a comparatively large percentage lies close to the mean or average value. The result of random assembly will be a large percentage of fits close to the mean, with an inconsequential number approaching either extreme value. Accordingly it is erroneous practice to attempt to define a fit by the maximum and minimum values which seldom, if ever, occur. It is desirable to designate a class of fit by some appropriate symbol, and to employ this in specifying the use. An effective comparison of one class of fit with another is made by referring to their mean or average values. A further statistical check will enable a frequency allotment to be made showing the percentage of assemblies having a fit within given limits about the mean. Experience in the application of ball bearings has shown that approximately 95% of the assemblies lie within a range of about one-half that between the extreme limits of the fit.

## 8. LUBRICATION OF BALL BEARINGS

**BASIC REQUIREMENTS** for lubrication of ball bearings include provision for the sliding contact between balls and retainer and the protection of the highly finished surfaces from corrosion. The supporting surfaces of the retainer are essentially plain bearings and must be lubricated as such by a lubricant film. The surface of the balls and raceways is particularly subject to corrosion, which will accelerate fatigue action and result in abrasive wear of the bearing parts.

A comparatively small quantity of lubricant, either oil or grease, will suffice for the requirements of the bearing if it is consistently present and properly distributed to the bearing surfaces. An excess quantity of lubricant is undesirable, and will cause the bearing to run at high temperature and aggravate leakage from the housing.

**OPERATING TEMPERATURE** of a bearing is a controlling factor in selecting the proper grade of lubricant. Load, speed and surrounding conditions directly affect this temperature, as does also the particular type of bearing and the design of the shaft enclosures. The viscosity of an oil and the type and consistency of a grease should be adapted to the operating temperature. Under normal conditions ball bearings will run at a temperature from 10° F. to 60° F. above surroundings.

**TYPE OF LUBRICANT.**—Either grease or oil is successfully used for the lubrication of ball bearings over a wide range of speed and operating conditions. The use of grease simplifies the design from the point of view of shaft enclosures to prevent leakage, but requires care in application to assure the grease gaining access to the bearing and at the same time to prevent overloading. Oil lubrication allows the use of a suitable oil level indicator as a check on the quantity in the housing. Oil generally is to be preferred for speeds above 10,000 r.p.m., and for operating temperatures, above 200° F.

**CONSUMPTION OF OIL OR GREASE** in a ball bearing is relatively small, but proper provision should be made in the housing design for its periodic renewal. Aside from the loss of lubricant, due to evaporation and leakage, both oil and grease are subject to progressive deterioration over a period of time. Contamination of the lubricant is the cumulative result of dirt, moisture or other foreign material gaining access to the housing. Wherever possible, the old lubricant should be removed and the bearing washed out with clean, hot oil, before renewal of the lubricant. In general, the lubricant can be applied and the housing sealed without subsequent attention for a period ranging from six months to two years, according to the severity of operating conditions. The necessity for renewal can best be determined, in the absence of definite information, by periodic inspection during the first year or more of service.

**SPECIFICATIONS** of physical and chemical properties cannot in themselves adequately define the qualities of a lubricant. Materials and methods used in refining and manufacture have an important influence on the performance of the lubricant in service. Many reputable manufacturers have specialized in the development of lubricants suitable for use with ball bearings, and this applies particularly to greases. In consideration of the fact that anti-friction bearings are susceptible to abrasive and corrosive action, only high-grade products should be used for their lubrication, and these should be obtained from established sources prepared to accept responsibility for the use of their product. Table 11 shows the character of oil recommended for various conditions of service.

Greases may be used throughout the range shown in Table 11 provided they are selected to meet the operating temperature of the bearing. The heavier oils are used for the higher bearing temperatures and suitable greases must have a corresponding temperature range. It is suggested that generally a grease with A.S.T.M. penetration 225 to 275 can be used where a light or medium oil is specified, and a stiffer or high temperature grease, A.S.T.M. penetration 150, where heavy and extra heavy oil is called for. See A.S.T.M. Specification for Oils and Greases.

The classification of oils given in Table 11 is arranged according to their viscosities, as

**Table 11.—Oil Lubrication of Ball Bearings**

| Bearing Load,* lb. | Revolution per minute | | | | | |
|---|---|---|---|---|---|---|
| | 100 | 300 | 600 | 1200 | 1800 | 3600 |
| 50 | light | light | light | light | medium | medium |
| 100 | light | light | light | medium | medium | medium |
| 500 | light | light | medium | medium | medium | heavy |
| 1,000 | medium | medium | medium | heavy | heavy | heavy |
| 2,000 | medium | medium | heavy | heavy | extra heavy | ........ |
| 5,000 | medium | heavy | heavy | extra heavy | ........ | ........ |
| 7,500 | heavy | heavy | extra heavy | ........ | ........ | ........ |
| 10,000 | heavy | extra heavy | ........ | ........ | ........ | ........ |
| 15,000 | extra heavy | extra heavy | ........ | ........ | ........ | ........ |

* Loads given in the table are radial loads only. For thrust loads multiply the actual thrust load by 4 and enter the table with this value of load.

determined by a Saybolt Standard Universal Viscosimeter at 100 deg. F., as follows: Extra light, 135–165 sec.; light, 180–220 sec.; medium, 270–330 sec.; heavy, 360–440 sec.; extra heavy, 450–550 sec.; steam cylinder and valve oils, 2000–2500 sec.

**MAINTENANCE.**—When proper judgment is shown in the design of a ball bearing application, successful operation depends largely on the degree of care exercised in maintenance. Mechanical equipment usually deteriorates more rapidly from neglect than from normal wear and tear in service. The precision and finish with which a ball bearing is manufactured is essential to its best performance, and these qualities will be retained only when the bearing is protected from the accumulation of dirt, rusting in the presence of water and corrosion from chemical action. Lubrication adequate in quality and quantity is necessary, and its provision is a responsibility of the user contributing directly to successful operation.

Bearings mounted in equipment which will be stored for an indefinite period before being placed in service should be protected by use of anti-rust compounds or slushing greases. Lubricating greases generally are inadequate for this purpose, and their use is productive of trouble due to the tendency to separate and harden when the equipment remains idle for protracted periods.

## 9. CYLINDRICAL AND SPHERICAL ROLLER BEARINGS

The demand made by heavy duty requirements has led to the development of additional bearing types that will *interchange* with ball bearings both in dimensional standards and design practice. Cylindrical roller bearings, Figs. 10–12, and spherical roller bearings, Fig. 9, serve this purpose and find their principal use under service conditions that lie beyond the inherent capacity of ball bearings. Cylindrical bearings are used on aircraft engine crank-shafts, railway traction motors, machine tool spindles, and in other applications where their load capacity, permissible speed, and precision of manufacture are of special advantage. Spherical roller bearings are used under the heaviest load and shock conditions, and find extensive application on railway journals, steel mill rolls, crusher and pulverizer equipment, and in other heavy duty industrial and production machinery.

**STANDARDS.—Cylindrical Roller Bearings** manufactured to dimensions and tolerances conforming with the single-row ball bearing, and light, medium and heavy series are listed in Table 6.

**Spherical Roller Bearings** are made in dimensions and tolerances conforming with the Int. Stds. Assoc. wide type, and are listed in Table 7. Other series are available in dimensions adapted to particular requirements. Information concerning these can be obtained from the manufacturer.

**BEARING TYPES.—Cylindrical Bearings** take their name from the cylindrical shape of the roller and raceways. Guide flanges integral with the inner race, type N, Fig. 10, or the outer race type, NU, Fig. 12, serve to constrain the roller and prevent skewing. A suitable cage or retainer is used to space the rollers and to retain them when the inner and outer races are separated in handling. The NF, Fig. 11, type has a flange integral with the outer race, which provides for endwise location and is capable of supporting a moderate thrust load component. The types shown are those found in general use. Modified constructions have been developed to serve special purposes.

**Spherical Roller Bearings,** Fig. 9, are named in consideration of the outer race roller path, which is a spherical surface having its center at the geometrical center of the bearing. The rollers are barrel shaped and have a line contact with the inner race roller path. The in-board end of the roller abuts against a guiding surface on the center flange. The cage or retainer is centered on this flange. The bearing is self-aligning and inherently provides for equalization of radial load between the two rows of rollers. This type of bearing affords a maximum load capacity within given space limitation. Although primarily suited for radial load it has a substantial thrust capacity, in both directions, by virtue of the angular contact of the rollers with the raceways.

**DESIGN PRACTICE.**—**Cylindrical Roller Bearings** are used in accordance with the design principles that apply with ball bearings. They may be operated at essentially the same speeds and have corresponding requirements in the matter of lubrication.

The separable feature of the bearing offers attractive possibilities in assembly procedure, and permits the use of press fits in mounting both inner and outer race. The heavier classes of fit generally should be employed, in view of the inherently greater shock- and load-carrying ability of the bearing. Open type bearings N and NU provide a free endwise float within the bearing, and the choice between these two types depends largely on assembly requirements.

The cylindrical bearing is in effect a rigid unit and due consideration should be shown for the problem of limiting shaft and housing deflection, and to maintaining correct alignment in assembly of the machine parts.

**Spherical Roller Bearings** serve as a self-contained, self-aligning, combined thrust and radial load carrier. Operating speeds are limited to lower values than for the cylindrical type. The lubricant used must gain access to the bearing and penetrate to the center guide flange of the inner race. Both grease and oil are successfully used, but careful consideration should be shown in the selection of lubricant to provide properly for the operating temperature and conditions affecting the supply of lubricant to the bearing.

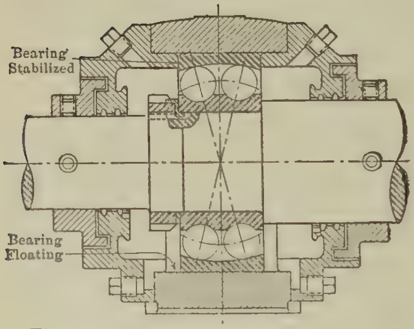


Fig. 16. Pillow Block with Self-aligning Ball Bearing

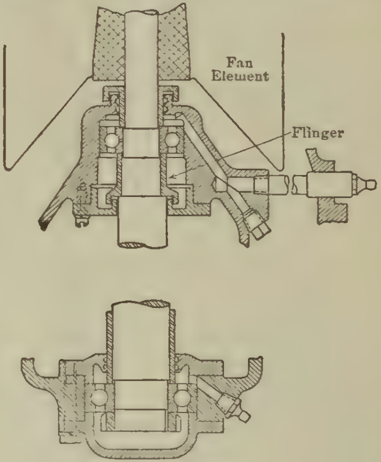


Fig. 17. High-speed Vertical Mounting

This bearing is used to carry the heaviest loads and shocks, and the shaft and housing fits used should be selected accordingly. The aligning feature is of particular value in providing for shaft and housing deflection occurring under load, and at the same time allows for a reasonable degree of incidental misalignment.

**SELECTION OF BEARING SIZE.**—The procedure followed in the selection of cylindrical and spherical bearing sizes is precisely the same as that used with ball bearings. Reference ratings shown in Tables 6 and 7 have been prepared on the same basis, and equations [1] to [5] relating load and speed to reference rating and design life are directly applicable. A graphical solution can be made by means of the chart, Fig. 15, but it should be noted that the speed limit line does not apply for the spherical bearing. When loads and reference ratings lie beyond the range of the chart scales, the decimal point can be shifted simultaneously in both the load and rating, to bring the values within the range of the chart.

In general, cylindrical and spherical bearings are used in the heavier and more complex types of equipment, where experience is of increased importance in design. The bearing manufacturer should be consulted in reference to technical problems of application, so that full advantage can be taken of the experience already gained in highly specialized fields.

## 10. TYPICAL APPLICATIONS

**PILLOW BLOCK.**—Fig. 16 shows a typical pillow block application of a self-aligning ball bearing. The inner ring is a press fit on the shaft seat and the lock-nut clamps the ring against the shaft shoulder. It is important that the shaft fillet at the shoulder clears the corner of the bearing; suitable fillet radii are tabulated in Tables 1 to 7. Manufacturers' catalog data can be consulted for detailed information as to limiting sizes of shaft shoulder, which should provide a flat surface for seating beyond the fillet and at the same time permit access to the face of the ring in order to facilitate removal.

The housing shown is designed for either grease or oil lubrication. Pipe plugs in the top section of the end covers provide for replenishment of the lubricant and drain plugs in the lower section provide for removal of used lubricant. End covers are designed to afford a reservoir capacity for the lubricant. Grooves in the end covers form a shaft enclosure that will prevent leakage from the housing under normal conditions. The inboard grooves have drain slots at the lowest position, to relieve any accumulation of lubricant. The shaft flingers, mounted external of the housing, provide a labyrinth enclosure which is an effective protection against entrance of dust, dirt, or water into the housing. The labyrinth seal depends on a close running clearance and is not subject to deterioration in service.

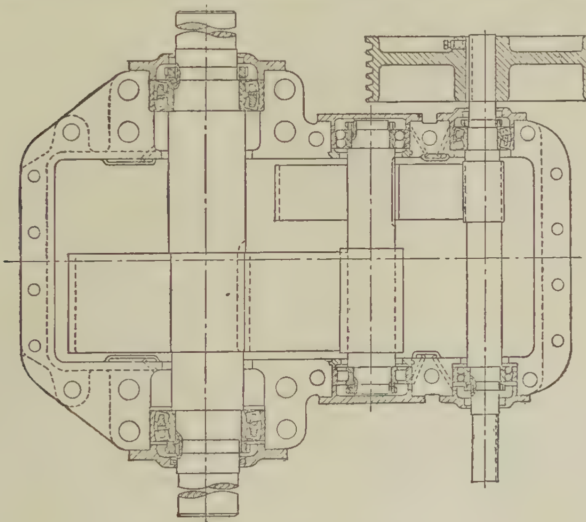


Fig. 18. Heavy-duty Gear Reduction Unit

The alternative designs for free or held bearings allow for axial float or stability. A general principle in the application of bearings is that but one bearing on any shaft is to be stabilized axially, in order to prevent undue thrust reactions as the result of temperature expansion or assembly errors.

**VERTICAL MOUNTING.**—Fig. 17 illustrates a high-speed vertical mounting on a fan, designed for grease lubrication. The lubricant is supplied above the bearing and a reservoir space provided below for its accumulation. Pressure-type grease fittings are used, but it is imperative in such installations to guard carefully against overcharging with grease by indiscriminate application. An internal type of labyrinth seal is provided at the lower shaft enclosure of the upper bearing housing to prevent leakage of lubricant. The external labyrinth flinger at the upper bearing tends to prevent lubricant leakage as the result of air blast induced by the fan element.

**GEAR REDUCTION UNIT.**—Fig. 18 shows a heavy-duty gear reduction unit. The gear case is split on the horizontal center line to facilitate assembly. Spherical roller bearings are used where the loads are heaviest and self-aligning ball bearings to compensate for shaft deflections under load. The open type NM cylindrical roller bearing used on the intermediate shaft provides for free axial float under the relatively heavy load from the adjacent pinion. The bearings are lubricated by oil from the gear case which is trapped on the bosses shown, when thrown from the gears, and fed through the drilled passages to each of the bearings. Leather cuff-type seals seal the case where the shaft passes through.

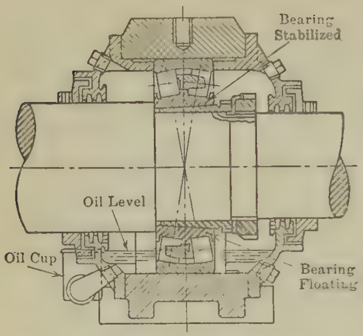


**Fig. 19.** Bearing with Adapter Sleeve

**ADAPTER SLEEVE.**—Fig. 19 shows a convenient type of mounting for the inner race where assembly conditions or service requirements preclude a press fit. The "push in" adapter sleeve is slotted lengthwise so that when it engages the tapered bore of the inner ring it will close in and clamp the shaft seat with an effect equivalent to that of a press fit. The sleeve is withdrawn at disassembly by a removal nut engaging the thread on the shoulder of the sleeve. This construction affords an additional opportunity of controllling internal clearance of the bearing to suit particular requirements.

# ROLLER BEARINGS

By Robert Thurston Kent

Roller bearings are of two general types, straight and tapered. The straight bearing will sustain radial loads only. The tapered bearing will sustain both radial and thrust loads. Straight roller bearings have rollers that are solid steel cylinders, or rollers formed of a flat strip of steel wound in a helix. Needle bearings are one form of straight roller bearings.

## 1. STRAIGHT ROLLER BEARINGS

**WOUND ROLLERS.**—One form of straight roller bearing made by the Hyatt Roller Bearing Co. uses a roller formed by helically winding into a hollow cylinder strips of alloy steel, which then are heat treated and ground to size. Rollers wound right and left hand are assembled alternately in the bearing, and held in proper alignment by a cage retainer. Bearings are made with both solid inner and outer races, with solid outer race and no inner race, and with a split outer race and no inner race. See Tables 1 and 2 for dimensions and capacities. The load ratings in these tables are based on a temperature of 350° F. For temperatures above 350°, the ratings should be multiplied by the following factors:

| Maximum temp., deg. F. . . . . . . | 350 | 400 | 450 | 500 | 550 | 600 |
|---|---|---|---|---|---|---|
| Temp. factor *T*. . . . . . . . . . . . . . . | 1.00 | 0.87 | 0.73 | 0.60 | 0.47 | 0.33 |

For certain classes of service, depending on load fluctuations, shock, overloads, etc., the load ratings should be reduced or increased by multiplying by the following factors:

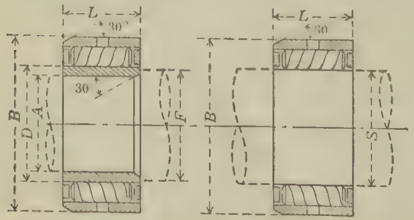


**Fig. 1.** Wound Roller Type Roller Bearing with Solid Outer Race

**Table 1,** 0.67–1.0, agricultural equipment, trailer trucks; 0.80–1.0, hand-operated hoists; 0.80, power-house cranes; 1.0–1.33, centrifugal extractors, electric motors, industrial locomotive journals and transmissions; 1.33, mining machinery, except mine cars; 1.33–2.0, air compressors, hot-saws, sprayers and dusters; 1.0–2.0, dairy machinery; 2.0, house lighting units; 5.0, tractor crawler tracks.

**Table 2,** 0.25, pintle bearings; 0.4–0.67, hand trucks; 0.5–0.67, industrial car caster wheels; 0.80–1.0, hand-operated cranes and hoists; 0.67, steel mill cars, charging cars, ingot cars; 0.67–1.0, agricultural equipment, gravity conveyors, industrial cars, contractors' cars; 0.80–1.0, hand-operated hoists; 1.33, mining machinery, except cars; 2.0–2.8, belt conveyor idlers, rotating shafts; 3.0, soaking pit covers; 3.0–4.0, belt conveyor idlers, stationary shafts. For unusual conditions the manufacturer should be consulted for the proper load rating.

**Lubrication** of wound roller bearings depends on the speed and temperature. The Hyatt Roller Bearing Co. gives a chart showing the range of speeds for various shaft

**Table 1.—Dimensions and Load Capacities of Roller Bearings with Wound Rollers**
**With Solid Outer and Inner Races, and Without Inner Race** (See Fig. 1)
(Hyatt Roller Bearing Co., Kearny, N. J.)

| Bearings With Inner Races | | | | Bearings With or Without Inner Races | | | | Diam. of Shaft, Bearing Without Inner Race, in. | Bearings With or Without Inner Races | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal Inside Diam. of Inner Race | | Nominal Outside Diam. of Inner Race, in. | Minimum Diam. of Shaft Shoulder, in. | Nominal Outside Diam. of Outer Race, in. | Length of Bearing, in. | Rollers | | | Load Rating, pounds, Based on Rotating Shaft. For Stationary Shafts, with Inner Races, Multiply Load Rating by 0.75. For Stationary Shafts Without Inner Races Multiply Load Rating by Factor *F* below. Load Ratings Based on Life of 5000 hr. | | | | | | |
| mm. | in. | | | | | No. | Diam., in. | | Revolutions per minute | | | | | | |
| *A* | | *D* | *F* | *B* | *C* | | | *S* | 50 | 100 | 200 | 500 | 1000 | 1500 | 2000 |
| ..... | ...... | ...... | ....... | 1.1875 | 1 | 6 | 1/4 | 1/2 | 585 | 500 | 395 | 265 | 185 | 150 | 125 |
| ..... | ...... | ...... | ....... | 1.3125 | 1 | 7 | 1/4 | 5/8 | 680 | 570 | 450 | 290 | 200 | 160 | 130 |
| ..... | ...... | ...... | ....... | 1.5625 | 1 | 7 | 5/16 | 3/4 | 810 | 675 | 520 | 345 | 240 | 190 | 155 |
| ..... | ...... | ...... | ....... | 1.6875 | 1 | 8 | 5/16 | 7/8 | 950 | 780 | 595 | 390 | 265 | 205 | 170 |
| ..... | ...... | ...... | ....... | 2.2500 | 1 | 7 | 1/2 | 1 | 1,050 | 885 | 690 | 460 | 310 | 240 | 200 |
| ..... | ...... | ...... | ....... | 2.3750 | 1 | 7 | 1/2 | 1 1/8 | 1,080 | 905 | 710 | 475 | 325 | 255 | 215 |
| 25 | 0.9843 | 1 1/4 | 1 3/16 | 2.4409 | 1 1/8 | 11 | 7/16 | 1 1/4 | 2,080 | 1,700 | 1,300 | 840 | 560 | 440 | 355 |
| 30 | 1.1811 | 1 1/2 | 1 7/16 | 2.4409 | 13/16 | 17 | 5/16 | 1 1/2 | 1,580 | 1,240 | 900 | 560 | 370 | 280 | 230 |
| 30 | 1.1811 | 1 1/2 | 1 7/16 | 2.4409 | 1 1/8 | 17 | 5/16 | 1 1/2 | 2,475 | 1,940 | 1,430 | 880 | 580 | 445 | 365 |
| 35 | 1.3780 | 1 3/4 | 1 5/8 | 2.8346 | 15/16 | 17 | 3/8 | 1 3/4 | 2,075 | 1,620 | 1,215 | 730 | 480 | 380 | 295 |
| 35 | 1.3780 | 1 3/4 | 1 5/8 | 2.8346 | 1 3/16 | 17 | 3/8 | 1 3/4 | 2,950 | 2,300 | 1,710 | 1,050 | 680 | 530 | 420 |
| 40 | 1.5748 | 2 | 1 7/8 | 3.1496 | 1 | 19 | 3/8 | 2 | 2,520 | 1,960 | 1,440 | 880 | 560 | 435 | 340 |
| 40 | 1.5748 | 2 | 1 7/8 | 3.1496 | 1 3/8 | 19 | 3/8 | 2 | 3,950 | 3,080 | 2,230 | 1,360 | 890 | 670 | 525 |
| 45 | 1.7717 | 2 3/16 | 2 1/16 | 3.3465 | 1 1/8 | 20 | 3/8 | 2 3/16 | 3,090 | 2,380 | 1,740 | 1,050 | 660 | 500 | 395 |
| 45 | 1.7717 | 2 3/16 | 2 1/16 | 3.3465 | 1 9/16 | 20 | 3/8 | 2 3/16 | 4,890 | 3,710 | 2,690 | 1,630 | 1040 | 790 | 615 |
| 50 | 1.9685 | 2 3/8 | 2 1/4 | 3.5433 | 1 1/4 | 22 | 3/8 | 2 3/8 | 3,880 | 3,000 | 2,150 | 1,300 | 820 | 610 | 475 |
| 50 | 1.9685 | 2 3/8 | 2 1/4 | 3.5433 | 1 3/4 | 22 | 3/8 | 2 3/8 | 6,100 | 4,700 | 3,325 | 2,000 | 1260 | 945 | 740 |
| 55 | 2.1654 | 2 5/8 | 2 1/2 | 3.9370 | 1 5/16 | 21 | 7/16 | 2 5/8 | 4,570 | 3,500 | 2,520 | 1,525 | 900 | 700 | 545 |
| 55 | 2.1654 | 2 5/8 | 2 1/2 | 3.9370 | 1 13/16 | 21 | 7/16 | 2 5/8 | 7,010 | 5,400 | 3,900 | 2,300 | 1460 | 1070 | 830 |
| 60 | 2.3622 | 2 7/8 | 2 3/4 | 4.3307 | 1 7/16 | 20 | 1/2 | 2 7/8 | 5,650 | 4,340 | 3,150 | 1,875 | 1170 | 855 | 655 |
| 60 | 2.3622 | 2 7/8 | 2 3/4 | 4.3307 | 1 15/16 | 20 | 1/2 | 2 7/8 | 8,350 | 6,300 | 4,625 | 2,800 | 1720 | 1260 | 965 |
| 70 | 2.7559 | 3 5/16 | 3 3/16 | 4.9213 | 1 5/8 | 23 | 1/2 | 3 5/16 | 7,400 | 5,650 | 4,000 | 2,380 | 1450 | 1020 | 750 |
| 70 | 2.7559 | 3 5/16 | 3 3/16 | 4.9213 | 2 3/8 | 23 | 1/2 | 3 5/16 | 11,400 | 8,700 | 6,220 | 3,650 | 2250 | 1570 | 1155 |
| 80 | 3.1496 | 3 3/4 | 3 5/8 | 5.5118 | 1 13/16 | 23 | 9/16 | 3 3/4 | 9,550 | 7,200 | 5,110 | 3,050 | 1870 | 1240 | 900 |
| 80 | 3.1496 | 3 3/4 | 3 5/8 | 5.5118 | 2 5/8 | 23 | 9/16 | 3 3/4 | 14,900 | 11,250 | 8,150 | 4,800 | 2800 | 1950 | 1410 |
| 90 | 3.5433 | 4 1/4 | 4 1/16 | 6.2992 | 2 1/16 | 22 | 11/16 | 4 1/4 | 11,700 | 8,800 | 6,210 | 3,670 | 2200 | 1480 | .... |
| 90 | 3.5433 | 4 1/4 | 4 1/16 | 6.2992 | 2 13/16 | 22 | 11/16 | 4 1/4 | 17,500 | 13,500 | 9,500 | 5,600 | 3400 | 2250 | .... |
| 100 | 3.9370 | 4 3/4 | 4 9/16 | 7.0866 | 2 5/16 | 22 | 3/4 | 4 3/4 | 14,600 | 11,000 | 7,850 | 4,530 | 2660 | 1760 | .... |
| 100 | 3.9370 | 4 3/4 | 4 9/16 | 7 0866 | 3 1/4 | 22 | 3/4 | 4 3/4 | 22,600 | 17,300 | 12,400 | 7,000 | 4100 | 2750 | .... |
| 110 | 4.3307 | 5 1/4 | 5 | 7.8740 | 2 9/16 | 21 | 7/8 | 5 1/4 | 18,500 | 14,000 | 10,100 | 5,700 | 3280 | 2125 | .... |
| 110 | 4.3307 | 5 1/4 | 5 | 7.8740 | 3 1/2 | 21 | 7/8 | 5 1/4 | 27,600 | 21,000 | 14,800 | 8,500 | 4900 | 3180 | .... |
| 120 | 4.7244 | 5 5/8 | 5 3/8 | 8.4646 | 2 13/16 | 21 | 15/16 | 5 5/8 | 21,000 | 16,000 | 11,300 | 6,400 | 3670 | 2500 | .... |
| 120 | 4.7244 | 5 5/8 | 5 3/8 | 8.4646 | 3 7/8 | 21 | 15/16 | 5 5/8 | 31,800 | 24,000 | 17,100 | 9,700 | 5500 | .... | .... |
| 130 | 5.1181 | 6 1/16 | 5 13/16 | 9.0551 | 3 1/8 | 21 | 1 | 6 1/16 | 27,700 | 20,900 | 14,750 | 7,980 | 4410 | .... | .... |
| 130 | 5.1181 | 6 1/16 | 5 13/16 | 9.0551 | 4 1/4 | 21 | 1 | 6 1/16 | 39,800 | 30,000 | 21,200 | 11,450 | 6330 | .... | .... |
| 140 | 5.5118 | 6 5/8 | 6 5/16 | 9.8425 | 3 1/4 | 22 | 1 1/16 | 6 5/8 | 28,200 | 22,300 | 15,900 | 8,800 | 4800 | .... | .... |
| 140 | 5.5118 | 6 5/8 | 6 5/16 | 9.8425 | 4 3/4 | 22 | 1 1/16 | 6 5/8 | 44,000 | 36,000 | 25,500 | 14,400 | 7630 | .... | .... |
| 150 | 5.9055 | 7 1/16 | 6 3/4 | 10.6326 | 3 1/2 | 21 | 1 3/16 | 7 1/16 | 32,700 | 26,400 | 18,500 | 10,500 | 5400 | .... | .... |
| 150 | 5.9055 | 7 1/16 | 6 3/4 | 10.6326 | 4 3/4 | 21 | 1 3/16 | 7 1/16 | 46,500 | 37,500 | 26,250 | 14,950 | 7750 | .... | .... |
| 160 | 6.2992 | 7 5/8 | 7 1/4 | 11.4201 | 3 7/8 | 21 | 1 1/4 | 7 5/8 | 33,600 | 28,400 | 20,800 | 11,300 | 5900 | .... | .... |
| 160 | 6.2992 | 7 5/8 | 7 1/4 | 11.4201 | 4 7/8 | 21 | 1 1/4 | 7 5/8 | 48,600 | 39,000 | 27,900 | 15,300 | 7900 | ... | .... |

**Factors for Stationary Shafts, without Inner Races**

| | | | | | |
|---|---|---|---|---|---|
| Brinell Hardness of Shaft........... | 400 | 450 | 500 | 550 | 600 |
| Rockwell Hardness of Shaft......... | C42 | C46 | C50 | C54 | C58 |
| Factor *F*..................... | 2.00 | 1.82 | 1.47 | 1.33 | 1.00 |

## Table 2.—Dimensions and Load Capacities of Roller Bearings with Wound Rollers

With Split Outer Race and No Inner Race (See Fig. 2)

(Hyatt Roller Bearing Co., Kearny, N. J.)

| Nominal Diam. of Shaft, in. | Nominal Outside Diam. of Race, in. | Bearing Length, in. | Limits for Housing Bore | | Rollers | | Load Ratings in Pounds. Based on Rotating Shaft. For Stationary Shafts, Multiply Load Rating by Factor *F* Below. Load Ratings Based on Life of 5000 hr. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | High | Low | No. | Diam., in. | Revolutions per Minute | | | | |
| *A* | *B* | *C* | in. | in. | | | 25 | 100 | 300 | 500 | 1000 |
| 1/2 | 1 1/16 | 1 | 0.500 | 0.498 | 6 | 1/4 | 65 | 60 | 50 | 45 | 35 |
| 1/2 | 1 1/16 | 2 | 0.500 | 0.498 | 6 | 1/4 | 155 | 145 | 125 | 105 | 80 |
| 5/8 | 1 3/16 | 1 | 0.625 | 0.623 | 7 | 1/4 | 80 | 75 | 60 | 55 | 40 |
| 5/8 | 1 3/16 | 2 | 0.625 | 0.623 | 7 | 1/4 | 190 | 175 | 150 | 130 | 95 |
| 3/4 | 1 7/16 | 1 | 0.750 | 0.748 | 7 | 5/16 | 95 | 85 | 70 | 60 | 50 |
| 3/4 | 1 7/16 | 2 1/2 | 0.750 | 0.748 | 7 | 5/16 | 275 | 250 | 205 | 175 | 140 |
| 7/8 | 1 9/16 | 1 | 0.875 | 0.873 | 8 | 5/16 | 110 | 100 | 80 | 65 | 55 |
| 7/8 | 1 9/16 | 2 1/2 | 0.875 | 0.873 | 8 | 5/16 | 335 | 305 | 250 | 205 | 170 |
| 1 | 1 15/16 | 1 | 1.000 | 0.997 | 7 | 3/8 | 130 | 120 | 105 | 90 | 65 |
| 1 | 1 15/16 | 2 | 1.000 | 0.997 | 7 | 3/8 | 350 | 325 | 275 | 240 | 180 |
| 1 | 1 15/16 | 4 | 1.000 | 0.997 | 7 | 3/8 | 780 | 730 | 610 | 530 | 400 |
| 1 1/8 | 2 3/16 | 2 | 1.125 | 1.122 | 8 | 7/16 | 390 | 365 | 305 | 265 | 200 |
| 1 1/8 | 2 3/16 | 4 | 1.125 | 1.122 | 8 | 7/16 | 865 | 810 | 680 | 590 | 440 |
| 1 1/4 | 2 7/16 | 2 | 1.250 | 1.247 | 8 | 1/2 | 425 | 400 | 330 | 285 | 210 |
| 1 1/4 | 2 7/16 | 4 | 1.250 | 1.247 | 8 | 1/2 | 950 | 890 | 740 | 635 | 470 |
| 1 1/2 | 2 13/16 | 2 | 1.500 | 1.497 | 8 | 9/16 | 505 | 470 | 385 | 330 | 240 |
| 1 1/2 | 2 13/16 | 3 | 1.500 | 1.497 | 8 | 9/16 | 820 | 760 | 620 | 530 | 390 |
| 1 1/2 | 2 13/16 | 5 | 1.500 | 1.497 | 8 | 9/16 | 1440 | 1340 | 1090 | 935 | 690 |
| 1 3/4 | 3 1/16 | 2 | 1.750 | 1.747 | 9 | 9/16 | 580 | 535 | 435 | 370 | 265 |
| 1 3/4 | 3 1/16 | 5 | 1.750 | 1.747 | 9 | 9/16 | 1650 | 1515 | 1240 | 1060 | 765 |
| 2 | 3 1/2 | 3 | 2.000 | 1.997 | 9 | 5/8 | 1060 | 980 | 785 | 665 | 475 |
| 2 | 3 1/2 | 5 | 2.000 | 1.997 | 9 | 5/8 | 1870 | 1725 | 1380 | 1170 | 840 |
| 2 1/4 | 4 1/4 | 3 | 2.250 | 2.247 | 8 | 7/8 | 1180 | 1090 | 870 | 730 | 515 |
| 2 1/4 | 4 1/4 | 6 | 2.250 | 2.247 | 8 | 7/8 | 2500 | 2315 | 1850 | 1550 | 1100 |
| 2 1/2 | 4 5/8 | 3 | 2.500 | 2.497 | 8 | 15/16 | 1300 | 1200 | 940 | 780 | 550 |
| 2 1/2 | 4 5/8 | 5 | 2.500 | 2.497 | 8 | 15/16 | 2290 | 2100 | 1650 | 1380 | 965 |
| 2 1/2 | 4 5/8 | 7 | 2.500 | 2.497 | 8 | 15/16 | 3280 | 3025 | 2360 | 1980 | 1385 |
| 2 3/4 | 4 7/8 | 4 | 2.750 | 2.747 | 9 | 15/16 | 1940 | 1775 | 1385 | 1160 | 790 |
| 2 3/4 | 4 7/8 | 6 | 2.750 | 2.747 | 9 | 15/16 | 3000 | 2750 | 2150 | 1800 | 1225 |
| 3 | 5 1/4 | 4 | 3.000 | 2.996 | 9 | 1 | 2100 | 1915 | 1480 | 1290 | 845 |
| 3 | 5 1/4 | 7 | 3.000 | 2.996 | 9 | 1 | 3840 | 3505 | 2710 | 2260 | 1525 |
| 3 1/2 | 6 1/8 | 4 | 3.500 | 3.496 | 12 | 1 1/8 | 2860 | 2570 | 1980 | 1650 | ...... |
| 3 1/2 | 6 1/8 | 7 | 3.500 | 3.496 | 12 | 1 1/8 | 5240 | 4740 | 3630 | 3020 | ...... |
| 4 | 6 5/8 | 4 | 4.000 | 3.996 | 14 | 1 1/8 | 3370 | 3040 | 2310 | 1885 | ...... |
| 4 | 6 5/8 | 7 | 4.000 | 3.996 | 14 | 1 1/8 | 6170 | 5570 | 4225 | 3460 | ...... |
| 4 1/2 | 7 3/8 | 4 | 4.500 | 4.496 | 14 | 1 1/4 | 3730 | 3300 | 2510 | 2050 | ...... |
| 4 1/2 | 7 3/8 | 7 | 4.500 | 4.496 | 14 | 1 1/4 | 6800 | 6100 | 4580 | 3740 | ...... |
| 5 | 7 7/8 | 4 | 5.000 | 4.995 | 15 | 1 1/4 | 3940 | 3520 | 2610 | 2110 | ...... |
| 5 | 7 7/8 | 7 | 5.000 | 4.995 | 15 | 1 1/4 | 7200 | 6400 | 4780 | 3870 | ...... |

Factors for Stationary Shafts

| | | | | | | |
|---|---|---|---|---|---|---|
| Brinell Hardness of Shaft....... | 100 | 200 | 300 | 400 | 500 | 600 |
| Rockwell Hardness of Shaft..... | B60 | B94 | C31 | C42 | C50 | C58 |
| Factor *F*..................... | 2.0 | 1.43 | 1.0 | 0.715 | 0.527 | 0.408 |

## Table 3.—Specifications for Oil for Wound-roller Roller Bearings

Neutralization value for all oils, 0.05 Mg. KOH gram, max.

| Temp. Range, deg. F. | Speed | | | Speed | | | Speed | | | Speed | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | High | Medium | Low | High | Medium | Low | High | Medium | Low | High | Medium | Low |
| | Viscosity, Saybolt Universal, seconds at 100° F. | | | Flash Point, min., deg. F. | | | Color, A.P.A. No. max. | | | Pour Test, deg. F., max. | | |
| Below 32 | 145–175 | 145–175 | 265–340 | 320 | 320 | 340 | 4 | 4 | 6 | 0 | 0 | 0 |
| 32–150 | 265–340 | 560–900 | 560–900 | 340 | 385 | 385 | 6 | 7 | 7 | 45 | 45 | 32 |
| 150–250 | 500–1000 | 500–1000 | 500–1000 | 425 | 425 | 425 | 7 | 7 | 7 | ..... | ..... | ..... |
| 250–500 | 150–190* | 150–190* | 150–190* | 540 | 540 | 540 | ..... | ..... | ..... | ..... | ..... | ..... |

* At 210° F.

**Table 4.—Specifications for Grease for Wound Roller Roller Bearings**

| Temp. Range, deg. F. | Speed |  | Speed |  | For All Greases |
|---|---|---|---|---|---|
|  | Medium | Low | Medium | Low |  |
|  | Soap, percent |  | Viscosity of Mineral Oil, Saybolt Universal, seconds |  |  |
| Below 32 | ....... | 5–8 | ....... | 275–325 | Free fatty acids, 0.2%, max.<br>Uncombined lime, 0.10%, max.<br>Water, 1.0%, max.<br>Sand, silicate or grit, 0.02% max. |
| 32–150 | 5–8 | 9–12 | 275–325 | 475–525 |  |

diameters, by which a bearing can be classed as low, medium or high speed. From this chart the following figures are taken:

| Shaft diam, in. | $^1/_4$ | $^1/_2$ | 1 | $1\,^1/_2$ | 2 | $2\,^1/_2$ | 3 | 4 | 5 | 6 | 8 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Low speed, max. r.p.m. | 865 | 755 | 570 | 420 | 310 | 250 | 200 | 165 | 145 | 120 | 90 | 72 |
| Medium speed, max. r.p.m | 3140 | 2820 | 2295 | 1875 | 1535 | 1245 | 1030 | 785 | 645 | 535 | 380 | 325 |

Tables 3 and 4 give the specifications for oil and grease for these bearings, as recommended by the maker. In general, the lightest oil that will stay in the housing should be used. Lubricants heavier than light liquid grease are inadvisable, but when their use is necessary, speeds should be under 300 r.p.m., with temperatures between 50 and 100° F.

**SOLID ROLLER BEARINGS.**—Solid roller bearings comprise a series of solid cylindrical rollers confined between inner and outer raceways, and held in position relative to each other by cages or spacers. The inner raceway frequently is grooved to prevent endwise movement of the rollers. In some cases the inner raceway may be the shaft or spindle itself. The outer raceway may be a plain ring. The rollers then are free to move endwise across the face of the raceway, thus permitting endwise movement of the shaft, or the outer raceway may have a single lip or two lips, forming a groove in the raceway, thus locating the shaft endwise. The outer raceway may be turned to the surface of a sphere, and enclosed in a ring with a spherical interior surface, making a self-aligning bearing. Various forms with their dimensions are shown in Table 5. Table 6 gives dimensions and shows the form of lock nut and lock washer for fastening the bearings to the shaft.

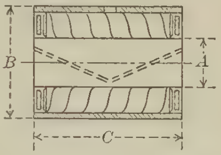


Fig. 2. Wound Roller Type Roller Bearing with Split Outer Race

## 2. NEEDLE BEARINGS

**NEEDLE OR QUILL BEARINGS.**—Needle bearings are straight roller bearings in which diameter of roller is not over $^1/_8 \times$ roller length. Separate outer and inner raceways may be used, or the inner raceway may be the shaft and the outer raceway integral with the structure. These bearings are used where high load-carrying capacity is required in a small space. Load-carrying capacity depends on hardness of the raceways and rollers. Bantam Ball Bearing Co. recommends a hardness, Rockwell C scale, of 56 to 60 for rollers, and of 58 to 60 for raceways. For hardness of less than 58–60 of either raceway, the rated load should be multiplied by the following factors:

| Rockwell hardness, C scale | 60 | 58 | 56 | 54 | 52 | 50 |
|---|---|---|---|---|---|---|
| Rated load factor | 1.00 | 0.96 | 0.91 | 0.81 | 0.70 | 0.50 |

The following formulas may be used in the design of needle bearings: Let $L$ = rated safe load on bearing, lb.; $D$ = diam. of shaft or inner raceway, in.; $D_p$ = pitch diam., in.; $d$ = diam. of roller, in.; $l$ = effective length of roller, in.; $C$ = circumferential clearance on pitch diameter; $S$ = speed of bearing, r.p.m.; for oscillating bearings $S$ may be taken as 8; $K$ = a constant (Table 7). Then

$$L = (10{,}000\,Nld)/\sqrt[3]{S} \quad . \quad . \quad . \quad [1]; \quad N = \pi(D + d)/d \quad . \quad . \quad . \quad . \quad [2]$$

$$D_p = Kd + (C/\pi) \quad . \quad . \quad . \quad . \quad [3]; \quad D = D_p - d \quad . \quad . \quad . \quad . \quad . \quad [4]$$

$C$ ranges from a minimum value of 0.0001 in. per roller to values of greater than $d$. For rotation, $C$ should be increased with $N$ by 0.5 $d$ when $N$ = 20 to 0.9 $d$ when $N$ = 50. For oscillating motion, $C$ may be kept close to minimum values.

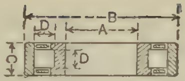
An approximate value of $N$ is found by assuming $D$ and $d$. The value of $K$ corresponding to $N$ then is taken from Table 7 and $D_p$ calculated (formula [3]); the maximum value of $D$ then is found by formula [4]. Diametral clearance should be subtracted from the value of $D$ so found. Bantam Ball Bearing Co. recommends the following

| Shaft diam., in. | up to 2 | 2–4 | 4–6 |
|---|---|---|---|
| Diametral clearance, in., min. | 0.0005 | .0010 | .0015 |
| Diametral clearance, in., max. | 0.0015 | .0020 | .0025 |

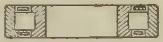
(*Continued on* p. 13–89)

## Table 5.—Dimensions and Load Capacities of Solid Roller Bearings

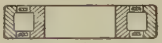
(Norma-Hoffman Bearings Corp., Stamford, Conn.)



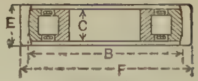
Standard



1-Lip



2-Lip



Self-aligning

| Dimensions—All Types | | | | Standard and 1-Lipped Bearings | | | | | | | 2-Lipped and Self-Aligning Bearings | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Bore, A | | Outside Diam. B, in. | Width C, in. | Rollers | | Speed, r.p.m. | | | | | Rollers | | Overall Width, E, in. | Overall Diam., F, in. | Speed, r.p.m. | | | | |
| mm. | in. | | | Diam. D, in. | No. | 100 | 500 | 1000 | 2000 | 4000 | Diam. D, in. | No. | | | 100 | 500 | 1000 | 2000 | 3000 |
| | | | | | | Load, lb. | | | | | | | | | Load, lb. | | | | |
| Heavy Series | | | | | | | | | | | | | | | | | | | | |
| 20 | 0.7874 | 2.8347 | 0.748 | 7/16 | 10 | 4,850 | 2,680 | 2,130 | 1,690 | 1340 | 7/16 | 7 | 0.827 | 3.1496 | 3,890 | 2,270 | 1,800 | 1,430 | 1250 |
| 25 | 0.9843 | 3.1496 | .827 | 1/2 | 10 | 5,770 | 3,370 | 2,680 | 2,120 | 1690 | 1/2 | 7 | .906 | 3.5433 | 4,900 | 2,870 | 2,270 | 1,800 | 1580 |
| 30 | 1.1811 | 3.5433 | .906 | 9/16 | 10 | 7,090 | 4,150 | 3,290 | 2,610 | 2070 | 9/16 | 7 | .984 | 3.9370 | 6,030 | 3,520 | 2,800 | 2,220 | 1940 |
| 35 | 1.3780 | 3.9370 | .984 | 5/8 | 10 | 8,550 | 5,000 | 3,970 | 3,150 | 2500 | 5/8 | 7 | 1.063 | 4.3307 | 7,270 | 4,250 | 3,370 | 2,680 | 2340 |
| 40 | 1.5748 | 4.3307 | 1.063 | 5/8 | 12 | 10,260 | 6,000 | 4,760 | 3,780 | 3000 | 5/8 | 8 | 1.142 | 4.7244 | 8,550 | 5,000 | 3,970 | 3,150 | 2750 |
| 45 | 1.7717 | 4.7244 | 1.142 | 11/16 | 12 | 12,100 | 7,070 | 5,610 | 4,450 | 3540 | 11/16 | 8 | 1.220 | 5.1181 | 10,080 | 5,890 | 4,680 | 3,710 | 3240 |
| 50 | 1.9685 | 5.1181 | 1.220 | 3/4 | 12 | 14,090 | 8,240 | 6,540 | 5,190 | 4120 | 3/4 | 8 | 1.299 | 5.5118 | 11,740 | 6,860 | 5,450 | 4,320 | 3780 |
| 55 | 2.1654 | 5.5118 | 1.299 | 13/16 | 12 | 16,210 | 9,480 | 7,520 | 5,970 | ...... | 13/16 | 8 | 1.378 | 5.9055 | 13,500 | 7,900 | 6,270 | 4,970 | 4340 |
| 60 | 2.3622 | 5.9055 | 1.378 | 7/8 | 12 | 18,490 | 10,810 | 8,580 | 6,810 | ...... | 7/8 | 8 | 1.457 | 6.2992 | 15,400 | 9,010 | 7,150 | 5,670 | 4960 |
| 70 | 2.7559 | 7.0866 | 1.654 | 1 | 12 | 23,340 | 13,650 | 10,830 | 8,590 | ...... | 1 | 8 | 1.732 | 7.5984 | 19,440 | 11,370 | 9,020 | 7,160 | 6360 |
| 80 | 3.1496 | 7.8740 | 1.890 | 1 1/8 | 12 | 28,670 | 16,770 | 13,310 | 10,560 | ...... | 1 1/8 | 8 | 1.969 | 8.3858 | 23,890 | 13,970 | 11,090 | 8,800 | ...... |
| 90 | 3.5433 | 8.8582 | 2.126 | 1 1/4 | 12 | 34,500 | 20,180 | 16,010 | ........ | ...... | 1 1/4 | 8 | 2.283 | 9.4488 | 28,740 | 16,810 | 13,340 | 10,590 | ...... |
| 100 | 3.9370 | 10.4330 | 2.362 | 1 3/8 | 12 | 40,770 | 23,840 | 18,920 | ........ | ...... | 1 3/8 | 9 | 2.520 | 10.0236 | 35,660 | 20,860 | 16,550 | 13,140 | ...... |
| Medium Series | | | | | | | | | | | | | | | | | | | | |
| 10 | 0.3937* | 1.3780* | 0.433* | 3/16* | 10* | 1,050* | 615* | 490* | 390* | 310* | ...... | .... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 15 | .5906 | 1.6535 | .512 | 1/4 | 9 | 1,530 | 895 | 710 | 565 | 445 | 1/4 | 8 | 0.591 | 1.8898 | 1,440 | 845 | 670 | 530 | 465 |
| 20 | .7874 | 2.0473 | .591 | 1/4 | 12 | 2,040 | 1,190 | 945 | 750 | 595 | 1/4 | 10 | .669 | 2.2835 | 1,700 | 990 | 790 | 625 | 545 |
| 25 | .9843 | 2.4410 | .669 | 5/16 | 12 | 3,010 | 1,760 | 1,400 | 1,110 | 880 | 5/16 | 9 | .748 | 2.6772 | 2,380 | 1,390 | 1,110 | 880 | 770 |
| 30 | 1.1811 | 2.8347 | .748 | 3/8 | 12 | 4,190 | 2,450 | 1,950 | 1,540 | 1230 | 3/8 | 9 | .827 | 3.1496 | 3,320 | 1,940 | 1,540 | 1,220 | 1070 |
| 35 | 1.3780 | 3.1496 | .827 | 3/8 | 13 | 4,540 | 2,660 | 2,110 | 1,670 | 1330 | 3/8 | 10 | .906 | 3.5433 | 3,850 | 2,250 | 1,780 | 1,420 | 1240 |
| 40 | 1.5748 | 3.5433 | .906 | 7/16 | 12 | 5,500 | 3,220 | 2,550 | 2,020 | 1610 | 7/16 | 10 | .984 | 3.9370 | 5,040 | 2,950 | 2,340 | 1,860 | 1620 |
| 45 | 1.7717 | 3.9370 | .984 | 1/2 | 12 | 6,920 | 4,050 | 3,210 | 2,550 | 2020 | 1/2 | 10 | 1.063 | 4.3307 | 6,060 | 3,540 | 2,810 | 2,230 | 1950 |
| 50 | 1.9685 | 4.3307 | 1.063 | 9/16 | 12 | 8,510 | 4,980 | 3,950 | 3,130 | 2490 | 9/16 | 9 | 1.142 | 4.7244 | 7,450 | 4,360 | 3,460 | 2,740 | 2400 |
| 55 | 2.1654 | 4.7244 | 1.142 | 9/16 | 13 | 9,220 | 5,390 | 4,280 | 3,400 | ...... | 9/16 | 10 | 1.220 | 5.1181 | 8,160 | 4,770 | 3,790 | 3,000 | 2620 |
| 60 | 2.3622 | 5.1181 | 1.220 | 5/8 | 12 | 10,260 | 6,000 | 4,760 | 3,780 | ...... | 5/8 | 10 | 1.299 | 5.5118 | 9,410 | 5,500 | 4,370 | 3,470 | 3030 |
| 70 | 2.7559 | 5.9055 | 1.378 | 5/8 | 15 | 12,830 | 7,500 | 5,950 | 4,720 | ...... | 5/8 | 12 | 1.457 | 6.2992 | 11,110 | 6,500 | 5,160 | 4,090 | 3580 |

* Standard type only.

**Table 5.—Dimensions and Load Capacities of Solid Roller Bearings.—*Continued***

(Norma-Hoffman Bearings Corp., Stamford, Conn.)

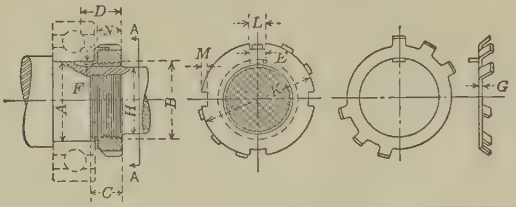
| Dimensions—All Types | | | | Standard and 1-Lipped Bearings | | | | | | | 2-Lipped and Self-Aligning Bearings | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Bore, *A* | | Outside Diam. *B*, in. | Width *C*, in. | Rollers | | Speed, r.p.m. | | | | | Rollers | | Overall Width, *E*, in. | Overall Diam., *F*, in. | Speed, r.p.m. | | | | |
| mm. | in. | | | Diam. *D*, in. | No. | 100 | 500 | 1000 | 2000 | 4000 | Diam. *D*, in. | No. | | | 100 | 500 | 1000 | 2000 | 3000 |
| | | | | | | Load, lb. | | | | | | | | | Load, lb. | | | | |
| | | | | | | | | | Medium Series—*Continued* | | | | | | | | | | |
| 80 | 3.1496 | 6.6929 | 1.535 | 3/4 | 15 | 17,610 | 10,300 | 8,170 | 6,490 | ...... | 3/4 | 11 | 1.614 | 7.2047 | 15,260 | 8,930 | 7,080 | 5,620 | ...... |
| 90 | 3.5433 | 7.4803 | 1.693 | 7/8 | 15 | 23,110 | 13,510 | 10,730 | ...... | ...... | 7/8 | 11 | 1.850 | 7.9921 | 19,260 | 11,260 | 8,940 | 7,090 | ...... |
| 100 | 3.9370 | 8.4646 | 1.850 | 1 | 14 | 27,220 | 15,920 | 12,630 | ...... | ...... | 1 | 10 | 2.008 | 9.0551 | 23,330 | 13,650 | 10,830 | 8,590 | ...... |
| 110 | 4.3307 | 9.4488 | 1.969 | 1 1/8 | 14 | 33,440 | 19,560 | 15,520 | ...... | ...... | 1 1/8 | 10 | 2.126 | 10.0394 | 28,670 | 16,770 | 13,300 | ...... | ...... |
| 120 | 4.7244* | 10.2362* | 2.165* | 1 1/8* | 16* | 38,220* | 22,350* | 17,750* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 130 | 5.1181* | 11.0236* | 2.283* | 1 1/4* | 15* | 43,110* | 25,210* | 20,000* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 140 | 5.5118* | 11.8110* | 2.441* | 1 3/8* | 15* | 50,950* | 29,800* | 23,650* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| | | | | | | | | | Light Series | | | | | | | | | | |
| 10* | 0.0397* | 1.1811* | 0.354* | 3/16* | 8* | 630* | 370* | 295* | 235* | 185* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 15* | .5905* | 1.3780* | .433* | 3/16* | 10* | 790* | 460* | 365* | 290* | 230* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 20 | .7874 | 1.8504 | .551 | 1/4 | 10 | 1,270 | 745 | 590 | 470 | 370 | 1/4 | 9 | 0.630 | 2.0866 | 1,210 | 705 | 560 | 445 | 390 |
| 25 | .9843 | 2.0473 | .591 | 1/4 | 12 | 1,530 | 895 | 710 | 565 | 445 | 1/4 | 10 | .669 | 2.2835 | 1,400 | 820 | 650 | 515 | 450 |
| 30 | 1.1811 | 2.4410 | .630 | 5/16 | 12 | 2,260 | 1,320 | 1,050 | 835 | 660 | 5/16 | 10 | .709 | 2.6772 | 2,070 | 1,210 | 960 | 765 | 665 |
| 35 | 1.3780 | 2.8347 | .669 | 5/16 | 14 | 2,640 | 1,540 | 1,220 | 975 | 770 | 5/16 | 11 | .748 | 3.1496 | 2,350 | 1,375 | 1,090 | 865 | 760 |
| 40 | 1.5748 | 3.1496 | .709 | 3/8 | 12 | 3,150 | 1,840 | 1,460 | 1,160 | 920 | 3/8 | 11 | .787 | 3.5433 | 2,880 | 1,690 | 1,340 | 1,060 | 930 |
| 45 | 1.7717 | 3.3465 | .748 | 3/8 | 14 | 3,670 | 2,150 | 1,700 | 1,350 | 1070 | 3/8 | 11 | .827 | 3.7402 | 3,280 | 1,910 | 1,520 | 1,210 | 1050 |
| 50 | 1.9685 | 3.5433 | .787 | 3/8 | 16 | 4,190 | 2,450 | 1,950 | 1,540 | 1230 | 3/8 | 12 | .866 | 3.9370 | 3,670 | 2,150 | 1,700 | 1,350 | 1180 |
| 55 | 2.1654 | 3.9370 | .827 | 3/8 | 17 | 4,450 | 2,610 | 2,070 | 1,640 | 1300 | 3/8 | 13 | .906 | 4.3307 | 3,930 | 2,300 | 1,820 | 1,450 | 1270 |
| 60 | 2.3622 | 4.3307 | .866 | 3/8 | 18 | 4,720 | 2,760 | 2,190 | 1,740 | 1380 | 3/8 | 15 | .946 | 4.7244 | 4,320 | 2,530 | 2,010 | 1,590 | 1390 |
| 70 | 2.7559 | 4.9213 | .945 | 7/16 | 18 | 6,180 | 3,620 | 2,870 | 2,280 | ...... | 7/16 | 14 | 1.024 | 5.3150 | 5,500 | 3,210 | 2,550 | 2,020 | 1770 |
| 80 | 3.1496 | 5.5118 | 1.024 | 1/2 | 18 | 7,790 | 4,550 | 3,610 | 2,870 | ...... | 1/2 | 14 | 1.102 | 5.9055 | 6,920 | 4,050 | 3,210 | 2,550 | ...... |
| 90 | 3.5433 | 6.2992 | 1.181 | 9/16 | 19 | 10,100 | 5,910 | 4,690 | 3,720 | ...... | 9/16 | 14 | 1.260 | 6.8110 | 8,780 | 5,130 | 4,070 | 3,230 | ...... |
| 100 | 3.9370 | 7.0866 | 1.339 | 5/8 | 20 | 12,830 | 7,500 | 5,950 | ...... | ...... | 5/8 | 15 | 1.417 | 7.5984 | 10,900 | 6,370 | 5,060 | 4,020 | ...... |
| 110 | 4.3307 | 7.8740 | 1.496 | 3/4 | 18 | 15,850 | 9,270 | 7,360 | ...... | ...... | 3/4 | 13 | 1.654 | 8.4646 | 13,500 | 7,980 | 6,330 | ...... | ...... |
| 120* | 4.2744* | 8.4646* | 1.575* | 7/8* | 18* | 20,800* | 12,160* | 9,660* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 130* | 5.1181* | 9.0551* | 1.575* | 15/16* | 18* | 23,450* | 13,720* | 10,900* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 140* | 5.5118* | 9.8426* | 1.654* | 1 * | 18* | 27,560* | 16,120* | 12,800* | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

* Standard type only.

### Table 6.—Dimensions of Lock-nuts and Lock Washers, and Shaft Machining Dimensions

(Norma-Hoffman Bearings Corporation, Stamford, Conn.)

All dimensions in inches



| Nominal Shaft Diam., A | | Threads | | Keyway | | | Thickness of Lock Washer, G | Shaft Extension, Max. Diam., H | Lock-nut | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| mm. | in. | Outside Diam., B | Length, C | Useful Length, D | Width, E | Depth, F | | | Diam., K | Width of Slot, L | Depth of Slot, M | Thickness, N | Threads per Inch |
| 10 | 0.3937 | 0.391 | 5/16 | 11/32 | 1/8 | 1/16 | 0.032 | 5/16 | 3/4 | 1/8 | 1/16 | 7/32 | 32 |
| 15 | .5906 | .586 | 15/32 | 7/16 | 1/8 | 5/64 | .042 | 1/2 | 1 | 1/8 | 1/16 | 5/16 | 32 |
| 20 | .7874 | .781 | 1/2 | 1/2 | 3/16 | 5/64 | .042 | 22/32 | 1 3/8 | 3/16 | 3/32 | 3/8 | 32 |
| 25 | .9843 | .969 | 17/32 | 9/16 | 3/16 | 3/32 | .050 | 7/8 | 1 9/16 | 3/16 | 3/32 | 13/32 | 32 |
| 30 | 1.1811 | 1.172 | 17/32 | 9/16 | 3/16 | 3/32 | .050 | 1 1/16 | 1 3/4 | 3/16 | 3/32 | 13/32 | 18 |
| 35 | 1.3780 | 1.375 | 9/16 | 19/32 | 3/16 | 3/32 | .050 | 1 1/4 | 2 1/16 | 3/16 | 3/32 | 7/16 | 18 |
| 40 | 1.5748 | 1.562 | 9/16 | 5/8 | 5/16 | 3/32 | .058 | 1 15/32 | 2 1/4 | 1/4 | 3/32 | 7/16 | 18 |
| 45 | 1.7717 | 1.766 | 9/16 | 21/32 | 5/16 | 3/32 | .058 | 1 11/16 | 2 17/32 | 1/4 | 3/32 | 7/16 | 18 |
| 50 | 1.9685 | 1.968 | 5/8 | 23/32 | 5/16 | 3/32 | .058 | 1 7/8 | 2 11/16 | 1/4 | 3/32 | 1/2 | 18 |
| 55 | 2.1654 | 2.156 | 5/8 | 23/32 | 5/16 | 3/32 | .0625 | 2 1/16 | 2 31/32 | 1/4 | 1/8 | 1/2 | 18 |
| 60 | 2.3624 | 2.359 | 21/32 | 3/4 | 5/16 | 3/32 | .0625 | 2 1/4 | 3 5/32 | 1/4 | 1/8 | 17/32 | 18 |
| 70 | 2.7559 | 2.750 | 11/16 | 27/32 | 5/16 | 3/32 | .0625 | 2 5/8 | 3 5/8 | 1/4 | 1/8 | 9/16 | 18 |
| 80 | 3.1496 | 3.141 | 3/4 | 7/8 | 3/8 | 1/8 | .072 | 3 | 4 5/32 | 3/8 | 1/8 | 19/32 | 11 |
| 90 | 3.5433 | 3.531 | 7/8 | 1 | 3/8 | 1/8 | .094 | 3 3/8 | 4 21/32 | 3/8 | 5/32 | 11/16 | 11 |
| 100 | 3.9370 | 3.922 | 15/16 | 1 1/8 | 3/8 | 1/8 | .094 | 3 25/32 | 5 3/16 | 3/8 | 5/32 | 3/4 | 11 |
| 110 | 4.3307 | 4.328 | 1 | 1 3/16 | 3/8 | 5/32 | .125 | 4 3/16 | 5 23/32 | 1/2 | 3/16 | 25/32 | 11 |
| 120 | 4.7244 | 4.719 | 1 1/32 | 1 7/32 | 3/8 | 5/32 | .125 | 4 9/16 | 6 1/8 | 1/2 | 3/16 | 13/16 | 11 |
| 130 | 5.1181 | 5.109 | 1 3/32 | 1 5/16 | 1/2 | 5/32 | .125 | 4 15/16 | 6 3/4 | 5/8 | 1/4 | 7/8 | 11 |

### Table 7.—Values of *K*

| *N* | *K* | *N* | *K* | *N* | *K* | *N* | *K* | *N* | *K* | *N* | *K* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 3.236036 | 24 | 7.661074 | 37 | 11.791250 | 51 | 16.243850 | 64 | 20.380089 | 78 | 24.836174 |
| 11 | 3.549509 | 25 | 7.978935 | 38 | 12.109316 | 52 | 16.561564 | 65 | 20.697669 | 79 | 25.153708 |
| 12 | 3.863689 | 26 | 8.296583 | 39 | 12.427348 | 53 | 16.879250 | 66 | 21.015224 | 80 | 25.471218 |
| 13 | 4.178586 | 27 | 8.614006 | 40 | 12.745348 | 54 | 17.199862 | 67 | 21.336424 | 81 | 25.788702 |
| 14 | 4.493978 | 28 | 8.931192 | 41 | 13.064441 | 55 | 17.517599 | 68 | 21.654949 | 82 | 26.106171 |
| 15 | 4.809773 | 29 | 9.248982 | 42 | 13.381251 | 56 | 17.835304 | 69 | 21.972565 | 83 | 26.423612 |
| 16 | 5.125839 | 30 | 9.566632 | 43 | 13.699154 | 57 | 18.152981 | 70 | 22.290153 | 84 | 26.748188 |
| 17 | 5.442316 | 31 | 9.884511 | 44 | 14.017024 | 58 | 18.470632 | 71 | 22.607719 | 85 | 27.065753 |
| 18 | 5.758710 | 32 | 10.202520 | 45 | 14.334862 | 59 | 18.788252 | 72 | 22.925264 | 86 | 27.383306 |
| 19 | 6.075626 | 33 | 10.519702 | 46 | 14.660235 | 60 | 19.105846 | 73 | 23.242782 | 87 | 27.700831 |
| 20 | 6.392635 | 34 | 10.838034 | 47 | 14.972682 | 61 | 19.427555 | 74 | 23.560273 | 88 | 28.018339 |
| 21 | 6.709479 | 35 | 11.156268 | 48 | 15.290520 | 62 | 19.744845 | 75 | 23.883449 | 89 | 28.335836 |
| 22 | 7.026733 | 36 | 11.474469 | 49 | 15.608326 | 63 | 20.062479 | 76 | 24.201045 | 90 | 28.653295 |
| 23 | 7.344090 | | | 50 | 15.926103 | | | 77 | 24.518626 | | |

### Table 8.—Dimensions of Rollers for Quill Bearings

(Bantam Ball Bearing Co., South Bend, Ind.)

All dimensions in inches

| Max. Diam. | Minus Tolerance | Max. Length | Minus Tolerance | Max. Diam. | Minus Tolerance | Max. Length | Minus Tolerance | Max. Diam. | Minus Tolerance | Max. Length | Minus Tolerance |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.0627 | 0.0004 | 0.395 | 0.020 | 0.1582 | 0.0002 | 1.015 | 0.015 | 0.3125 | 0.0002 | 2.500 | 0.020 |
| .0627 | .0004 | .666 | .020 | .1875 | .0002 | 1.000 | .010 | .3750 | .0002 | 3.250 | .020 |
| .0939 | .0004 | .489 | .010 | .2182 | .0002 | 0.755 | .020 | .5000 | .0002 | 4.500 | .020 |
| .0939 | .0003 | .975 | .020 | .2500 | .0002 | 1.250 | .015 | .7500 | .0002 | 6.500 | .020 |
| .1250 | .0002 | .750 | .020 | .2502 | .0004 | 1.750 | .010 | 1.0000 | .0002 | 9.000 | .020 |

Bore of outer raceway = $D_p + d +$ diametral clearance. The manufacturing tolerance of the outer raceway should be $+ 0, - 0.0005$ in.

Table 8 gives standard sizes of rollers for quill bearings. Several rows of rollers may be placed in one bearing to obtain the necessary length of roller. The rows should be separated by hardened and ground retaining rings. Longitudinal clearances of from 0.010 to 0.030 in. are required for each set of rollers.

## 3. TAPERED ROLLER BEARINGS

The principle of the tapered roller bearing is essentially a construction of the rolling members and raceways on the elements of a cone, so that lines coincident with the working surfaces of rollers and races all meet at a common point on the axis of the bearing. See Fig. 3. True rolling motion thus is obtained, and the bearing will handle radial or thrust loads or any combination of them. The essential parts are an inner raceway, or cone, an outer raceway or cup, tapered rollers, and a cage or roller retainer, Fig. 4. The accompanying tables of dimensions and capacities are selected as representing the more generally used bearings. Data regarding other sizes and types may be obtained from manufacturers.

**THE STANDARD BEARING** is made in three basic series, namely heavy, medium and light. These are established by maintaining fundamental relations between the various components of the bearing. The standard, or heavy series, has wide races and long rollers; the light series has narrow races and short rollers; the medium series is an approximate mean between the other two. See Table 9. The standard type of bearing is a general service bearing. It is widely used in automotive vehicles, machinery, farm implements, mining machinery, etc.

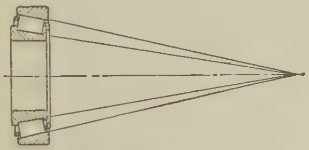


FIG. 3. Basic Principle of Tapered Roller Bearing

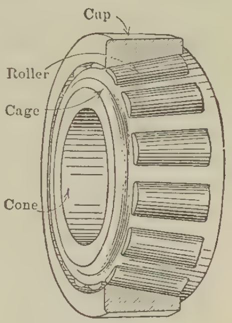


FIG. 4. Standard Single-row Tapered Roller Bearing

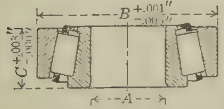


**Table 9.—Dimensions and Load Capacities of Standard Single-row Tapered Roller Bearings**

(Timken Roller Bearing Co., Canton, Ohio)

| Bore | Heavy Series | | | | Medium Series | | | | Light Series | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Dimensions, in. | | Load rating, lb., at 500 r.p.m. | | Dimensions, in. | | Load rating, lb., at 500 r.p.m. | | Dimensions, in. | | Load rating, lb., at 500 r.p.m. | |
| *A* | *B* | *C* | Radial | Thrust | *B* | *C* | Radial | Thrust | *B* | *C* | Radial | Thrust |
| 0.375 | ....... | ....... | ...... | ...... | ....... | ....... | ...... | ...... | 1.2595 | 0.394 | 255 | 205 |
| .500 | ....... | ....... | ...... | ...... | ....... | ....... | ...... | ...... | 1.3775 | .433 | 290 | 255 |
| .625 | ....... | ....... | ...... | ...... | ....... | ....... | ...... | ...... | 1.625 | .5625 | 490 | 300 |
| .750 | ....... | ....... | ...... | ...... | 1.938 | 0.7813 | 840 | 440 | 1.8504 | .5662 | 580 | 405 |
| .875 | ....... | ....... | ...... | ...... | 2.240 | .7625 | 1010 | 610 | 2.047 | .591 | 610 | 480 |
| 1.000 | 2.615 | 0.9375 | 1,625 | 810 | 2.375 | .7813 | 1010 | 650 | 2.3125 | .750 | 1010 | 650 |
| 1.125 | 2.8593 | 1.1875 | 2,150 | 1400 | 2.615 | .9375 | 1625 | 810 | 2.500 | .8125 | 1115 | 770 |
| 1.250 | 3.125 | 1.1563 | 2,265 | 1620 | 2.750 | .9375 | 1715 | 920 | 2.717 | .7813 | 1065 | 800 |
| 1.375 | 3.3125 | 1.1563 | 2,470 | 1320 | 3.000 | .9375 | 1805 | 1075 | 2.844 | .8125 | 1165 | 920 |
| 1.500 | 3.4375 | 1.1875 | 2,465 | 1480 | 3.125 | 1.000 | 1805 | 1075 | 3.000 | .8125 | 1330 | 1045 |
| 1.625 | 3.750 | 1.250 | 2,595 | 1445 | 3.3125 | 1.1875 | 2480 | 1950 | 3.1496 | .8268 | 1720 | 925 |
| 1.750 | 4.000 | 1.375 | 3,615 | 2020 | 3.4375 | 1.1875 | 2465 | 1480 | 3.3464 | .8125 | 1800 | 1080 |
| 1.875 | 4.250 | 1.4375 | 3,795 | 2205 | 3.6718 | 1.1875 | 2690 | 1785 | 3.500 | .8125 | 1800 | 1130 |
| 2.000 | 4.375 | 1.500 | 3,795 | 2205 | 3.875 | 1.1875 | 2690 | 1785 | 3.6718 | .8125 | 1875 | 1245 |
| 2.250 | 4.875 | 1.500 | 4,140 | 2815 | 4.125 | 1.1875 | 2855 | 1875 | 3.937 | .8268 | 1955 | 1355 |
| 2.500 | 5.375 | 1.625 | 5,280 | 3750 | 4.4375 | 1.1875 | 3005 | 2365 | 4.3307 | .8661 | 2095 | 1855 |
| 2.750 | 5.909 | 1.750 | 7,015 | 4475 | 4.875 | 1.1875 | 3210 | 2420 | 4.6875 | 1.000 | ...... | ...... |
| 3.000 | 6.375 | 1.875 | 7,315 | 4890 | 5.375 | 1.1875 | 3540 | 3080 | 5.000 | 1.0625 | 2230 | 1970 |
| 3.250 | 6.625 | 1.875 | 7,315 | 4890 | 5.5115 | 1.4375 | 4460 | 3530 | 5.375 | 1.1875 | 3540 | 3080 |
| 3.500 | 7.125 | 1.875 | 7,600 | 5740 | 6.000 | 1.5625 | 4925 | 4265 | 5.625 | 1.1875 | ...... | ...... |
| 3.750 | 7.500 | 2.250 | 10,400 | 6820 | 6.375 | 1.5625 | 5060 | 4710 | 5.875 | 1.250 | 3740 | 3610 |
| 4.000 | 7.875 | 2.4375 | 10,400 | 6820 | 6.625 | 1.625 | 6070 | 5600 | ....... | ....... | ...... | ...... |
| 4.250 | 8.375 | 2.625 | 12,250 | 7820 | ....... | ....... | ...... | ...... | ....... | ....... | ...... | ...... |

**THE STEEP ANGLE BEARING,** Fig. 5 is used in power take-offs, worms and bevel gear shafts, etc., where the thrust load predominates, although there is an appreciable radial load. It is similar to the standard bearing except that the angles of both the raceway and rollers are quite steep. See Table 10.

**FLANGED CUP BEARINGS,** Fig. 6, are used where extreme precision is required as on machine tool spindles. The bearing housing is bored straight through, and a seat faced at right angles to the bore. This face provides a backing for the flange of the cup. This bearing also is used where there is insufficient space for conventional bearing-mounting methods, and where the housing is of manganese steel or other material difficult to machine. Table 11 gives dimensions and capacities.

**MULTIPLE ROW BEARINGS** provide maximum load carrying capacity in small space, resist thrust loads from both directions and maintain a precise and rigid mounting or a floating assembly as desired. They are made in several types as follows:

**Two-row Double-cup Single-cone Bearings,** Fig. 7. Three assemblies are possible. 1. A spacer is introduced between the cones to establish the set-up or running clearances at the time the bearing is assembled. 2. The cones are butted together, thus

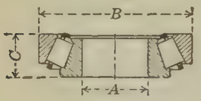


FIG. 5. Single-row Steep Angle Tapered Roller Bearings

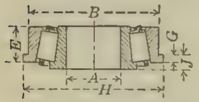


FIG. 6. Single-row Flanged-cup Tapered Roller Bearings

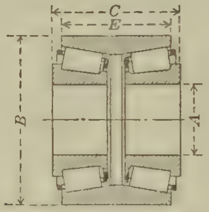


FIG. 7. Two-row Double-cup Single-cone Tapered Roller Bearings

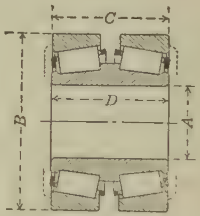


FIG. 8. Two-row Double-cone Single-cup Tapered Roller Bearings

### Table 10.—Steep Angle Tapered Roller Bearings—Single Row

(Timken Roller Bearing Co., Canton, Ohio)

| Dimensions, in. (Fig. 5) | | | Load Rating, lb., at 500 r.p.m. | | Dimensions, in. (Fig. 5) | | | Load Rating, lb., at 500 r.p.m. | |
|---|---|---|---|---|---|---|---|---|---|
| *A* | *B* | *C* | Radial | Thrust | *A* | *B* | *C* | Radial | Thrust |
| 0.750 | 2.125 | 0.875 | 1020 | 1185 | 1.500 | 3.4843 | 1.000 | 1765 | 2700 |
| .875 | 2.5625 | .875 | 1125 | 1635 | 1.750 | 3.750 | 2.2187 | 2145 | 3115 |
| 1.000 | 2.6875 | .875 | 1125 | 1635 | 2.000 | 4.375 | 1.1875 | 2295 | 3980 |
| 1.125 | 2.8593 | .9687 | 1535 | 1815 | 2.500 | 5.513 | 1.4375 | 3995 | 6780 |
| 1.250 | 3.125 | 1.000 | 1710 | 2240 | 3.000 | 7.000 | 2.1875 | 6525 | 9770 |

### Table 11.—Dimensions and Load Capacities of Flanged-cup Roller Bearings

(Timken Roller Bearing Co., Canton, Ohio)

| Dimensions, inches. (Fig. 6) | | | | | | Load Rating, lb., at 500 r.p.m. | |
|---|---|---|---|---|---|---|---|
| *A* | *B* | *E* | *G* | *H* | *J* | Radial | Thrust |
| 3.6250 | 6.0000 | 1.1875 | 0.2500 | 6.2500 | 0.6250 | 4,925 | 4,265 |
| 3.7500 | 6.2500 | 1.1875 | .2813 | 6.9063 | .7188 | 6,070 | 5,600 |
| 3.8750 | 6.3750 | 1.0313 | .2500 | 6.6250 | .6563 | 5,060 | 4,710 |
| 4.0000 | 7.1250 | 1.5000 | .3125 | 7.4375 | .6875 | 7,600 | 5,740 |
| 4.0000 | 7.5000 | 1.7500 | .3750 | 7.8750 | .8750 | 10,400 | 6,820 |
| 4.2500 | 8.3750 | 2.1250 | .4375 | 8.8125 | .9375 | 12,250 | 7,820 |
| 4.5000 | 7.5000 | 1.3750 | .3125 | 7.8125 | .8125 | 8,445 | 6,880 |
| 5.0000 | 8.5000 | 1.3750 | .3125 | 8.8125 | .8125 | 9,195 | 8,790 |
| 5.0000 | 9.2500 | 1.9375 | .4375 | 9.6875 | 1.0000 | 13,350 | 9,670 |
| 5.5000 | 9.5000 | 1.7500 | .3750 | 9.8750 | 0.8750 | 11,970 | 10,375 |
| 5.7500 | 10.0000 | 1.8750 | .4375 | 10.4375 | 1.1875 | 14,400 | 11,495 |
| 6.5000 | 11.3750 | 1.8750 | .4375 | 11.8125 | 1.0625 | 16,350 | 15,000 |
| 7.5000 | 12.5000 | 1.8125 | .4375 | 12.9375 | 1.2500 | 17,670 | 18,125 |

establishing the proper running clearance. 3. No cone spacer is used and the proper clearance is obtained by some form of adjusting device. Types 1 and 2 usually are preferred if quick removal and replacement of bearings is necessary, or if operating conditions do not necessitate take-up after a long period of service. With this type of bearing, shaft expansion is provided for by locking the cone in position and permitting the cup to float in the housing. The opposite end of the shaft usually is fixed in position, and movement is confined to one end. Both conventional and steep-angle assemblies are possible with this bearing. See Table 12.

**Two-row Double-cone Single-cup Bearings,** Fig. 8, are used when the capacity of a two-row bearing is required, with a simpler mounting than two standard single-row bearings. They also are used on the fixed end of a shaft when allowance must be made for shaft expansion. The bearing used on the opposite end then usually is a floating double-cup unit. See Table 13.

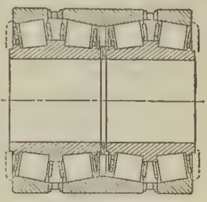


Fig. 9. Four-row Tapered Roller Bearing

**Four-row Bearings,** Fig. 9, are used where maximum capacity is required in a limited space, as in rolling mills, cement plants, rubber mills and other units which develop extreme pressures. It comprises four sets of rollers. The arrangement is similar to two double-cone bearings set side by side with a double cup carrying the load from the two inner sets of rollers and a single cup carrying the load imposed on the two outer sets of rollers. Spacer rings are recommended in all applications. Manufacturers should be consulted regarding application of this type of bearing.

**Table 12.—Dimensions and Load Capacities of Two-row Roller Bearings, Double Cup, Single Cones**

(Timken Roller Bearing Co., Canton, Ohio)

| Dimensions, in. (Fig. 7) | | | | Load Rating, lb., at 500 r.p.m. | | Dimensions, in. (Fig. 7) | | | | Load Rating, lb., at 500 r.p.m. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| *A* | *B* | *C* | *E* | Radial | Thrust | *A* | *B* | *C* | *E* | Radial | Thrust |
| 1.2500 | 2.7500 | 2.6250 | 2.2500 | 3,430 | 920 | 6.0000 | 12.1250 | 7.8750 | 5.7500 | 44,920 | 14,360 |
| 1.5000 | 3.1510 | 1.8125 | 1.3750 | 2,660 | 1,045 | 6.5000 | 11.3750 | 5.6250 | 4.3750 | 32,700 | 15,000 |
| 1.7500 | 3.5480 | 2.0000 | 1.7500 | 3,600 | 1,080 | 7.0000 | 11.2500 | 5.3750 | 3.6250 | 24,720 | 10,500 |
| 2.0000 | 3.6718 | 2.5625 | 2.0625 | 5,380 | 1,785 | 8.0000 | 12.5000 | 5.0000 | 3.5000 | 24,040 | 7,390 |
| 2.2500 | 3.9370 | 1.9375 | 1.5625 | 3,910 | 1,355 | 9.0000 | 14.0000 | 6.0000 | 4.3750 | 40,480 | 13,105 |
| 2.5000 | 4.8750 | 3.1250 | 2.5000 | 8,280 | 2,815 | 10.0000 | 16.6250 | 7.0312 | 5.5000 | 56,380 | 18,370 |
| 2.7500 | 5.0000 | 3.1875 | 2.5625 | 8,620 | 3,070 | 11.0000 | 18.5000 | 7.8750 | 5.8750 | 73,020 | 27,035 |
| 3.0000 | 6.0000 | 3.7500 | 3.0000 | 11,390 | 4,545 | 12.0000 | 17.4980 | 6.0000 | 5.0000 | 42,660 | 15,800 |
| 4.0000 | 7.1250 | 4.1250 | 3.3750 | 15,200 | 5,740 | 13.0000 | 19.0000 | 5.2500 | 3.5000 | 32,200 | 15,815 |
| 4.2500 | 6.2500 | 2.1250 | 1.5625 | 5,020 | 2,980 | 14.0000 | 19.0000 | 5.2500 | 3.5000 | 32,200 | 15,815 |
| 4.5000 | 7.5000 | 4.1875 | 3.1875 | 16,890 | 6,880 | 16.0000 | 22.6250 | 5.9375 | 3.9375 | 50,740 | 24,735 |
| 4.7500 | 9.8750 | 4.8750 | 3.5000 | 24,860 | 10,330 | 18.0000 | 28.7460 | 10.0000 | 7.0000 | 58,520 | 61,000 |
| 5.0000 | 9.2500 | 5.6250 | 4.5000 | 26,700 | 9,670 | 20.0000 | 29.0000 | 7.3426 | 4.5000 | 98,900 | 45,950 |
| 5.5000 | 9.5000 | 5.1875 | 4.1875 | 23,940 | 10,375 | | | | | | |

**Table 13.—Dimensions and Load Capacities of Two-row Roller Bearings, Double Cone, Single Cups**

(Timken Roller Bearing Co., Canton,. Ohio)

| Dimension, in. (Fig. 8) | | | | Load Rating, lb., at 500 r.p.m. | | Dimension, in. (Fig. 8) | | | | Load Rating, lb., at 500 r.p.m. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| *A* | *B* | *C* | *D* | Radial | Thrust | *A* | *B* | *C* | *D* | Radial | Thrust |
| 1.4375 | 2.6875 | 1.4492 | 1.5000 | 2,000 | 870 | 8.0000 | 12.5000 | 5.2500 | 5.2500 | 35,340 | 18,125 |
| 1.6875 | 2.1875 | 1.3750 | 1.2500 | 2,070 | 1,070 | 9.0000 | 14.0000 | 4.7500 | 4.7500 | 40,480 | 13,105 |
| 2.0000 | 3.6718 | 1.9690 | 2.0940 | 3,750 | 1,245 | 10.0000 | 14.3720 | 3.6500 | 3.6560 | 26,820 | 9,540 |
| 2.5000 | 5.5130 | 2.6020 | 2.5980 | 7,990 | 6,780 | 11.0000 | 18.5000 | 6.6875 | 6.5625 | 73,020 | 27,035 |
| 3.0000 | 6.3750 | 4.0000 | 4.2500 | 14,630 | 4,890 | 12.0000 | 17.2500 | 4.7500 | 3.8750 | 29,280 | 12,580 |
| 3.5000 | 7.5000 | 4.6250 | 5.0000 | 20,800 | 6,820 | 13.0000 | 19.0000 | 4.1250 | 3.7500 | 32,200 | 15,815 |
| 4.0000 | 7.5000 | 4.6250 | 5.0000 | 20,800 | 6,820 | 14.0000 | 20.2500 | 5.0000 | 4.3750 | 46,360 | 20,015 |
| 4.5000 | 9.2500 | 5.5000 | 6.0000 | 26,700 | 9,670 | 16.0000 | 22.6250 | 5.1563 | 4.5000 | 50,740 | 24,735 |
| 5.0000 | 11.6250 | 5.8750 | 6.2500 | 43,760 | 17,080 | 18.0000 | 33.9960 | 14.5000 | 14.5000 | 309,600 | 109,350 |
| 5.5000 | 8.7500 | 2.3850 | 2.1250 | 10,430 | 4,470 | 20.0000 | 33.0000 | 10.5000 | 10.5000 | 207,200 | 97,410 |
| 6.0000 | 11.7500 | 4.2500 | 4.3750 | 31,910 | 10,420 | 26.0000 | 42.0000 | 12.6563 | 12.2812 | 389,125 | 118,370 |
| 7.0000 | 11.3750 | 6.2500 | 6.2500 | 32,700 | 15,000 | 30.0000 | 48.0000 | 15.6250 | 15.2500 | 534,200 | 186,500 |

**Table 14.—Variation of Capacity with Speed of Tapered Roller Bearings.**
(Timken Roller Bearing Co., Canton, Ohio)

| R.p.m. | *F* | R.p.m. | *F* | R.p.m. | *F* | R.p.m. | *F* | R.p.m. | *F* | R.p.m. | *F* | R.p.m. | *F* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 100 | 1.621 | 300 | 1.166 | 500 | 1.000 | 900 | 0.838 | 1300 | 0.751 | 2000 | 0.660 | 3500 | 0.558 |
| 150 | 1.435 | 350 | 1.113 | 600 | 0.947 | 1000 | .812 | 1400 | .734 | 2250 | .637 | 4000 | .536 |
| 200 | 1.317 | 400 | 1.069 | 700 | .904 | 1100 | .789 | 1500 | .719 | 2500 | .617 | 4500 | .517 |
| 250 | 1.231 | 450 | 1.032 | 800 | .869 | 1200 | .769 | 1750 | .687 | 3000 | .584 | 5000 | .501 |

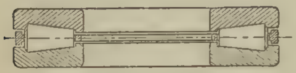
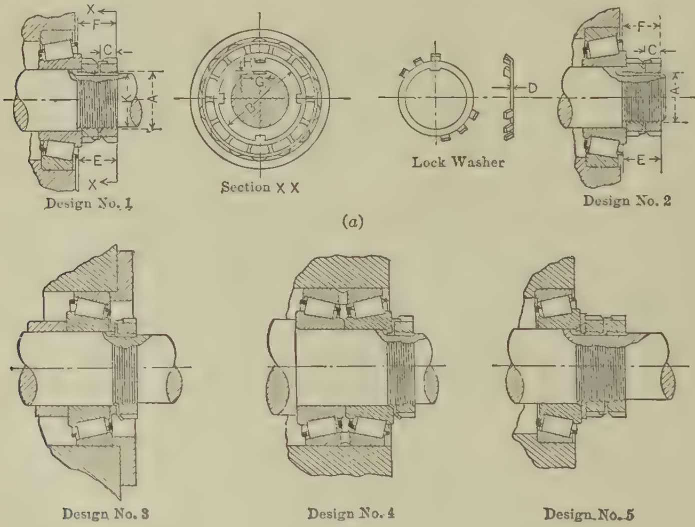


Fig. 10. Tapered Roller Thrust Bearing

**Thrust Bearings** to withstand thrust loads only without radial loads are shown in Fig. 10.

**LOAD FACTORS FOR TAPERED ROLLER BEARINGS.**—The load ratings for tapered roller bearings as given in the tables can be converted to ratings at speeds other than 500 r.p.m. by application of the factors given in Table 14.

**Life and Service Factors.**—The rating of the bearings given in Tables 9 to 13, are based on a life expectancy of 3000 hours of service. For longer life, the loads given in the tables must be reduced by an appropriate factor. Also for certain classes of service involving shock, inequality of loading, intermittent operation, etc., the load rating must be modified by an application factor. The two factors are combined in a service factor by which the rated load should be multiplied as follows: 0.8, steel mill machinery; 0.67, belt drives, farm tractor transmissions, gas or oil engines for farm use, gear reduction drives, I-beam trolleys, railway section cars, shop trucks and trailers; 0.57, automotive wheels and differentials; 0.50, dough mixers, elevators, industrial steam engines, fans and blowers, farm machinery, except tractor transmissions, garage compressors, glass making machinery, grain milling machinery, highway trailers, I-beam cranes and hoists, ice-making machinery, laundry machinery, leather working machinery, locomotives, mine and industrial, logging engines, machine tools, pulverizing and screening equipment, pumps, deep-well and fire, railway car drives, road building machinery, saw mill and woodworking machinery, stokers, textile machinery; 0.4, special motors and generators; 0.33, air compressors, clay working machinery, dredge pumps, printing machinery, rubber machinery; 0.25, paper mill machinery and pumps.



Design No. 1 Section X X Lock Washer Design No. 2

(*a*)

Design No. 3 Design No. 4 Design No. 5

(*b*)

Fig. 11. Timken Lock Nuts

METHODS OF MOUNTING TAPERED ROLLER BEARINGS.—Fig. 11 shows various methods of mounting tapered roller bearings. Design 1 is for those cases where the maximum possible shaft extension diameter is desired. In Design 2 it is possible to use a smaller size nut than in Design 1, and yet have the necessary cone backing. Design 3 uses a single lock-nut and tongued washer to clamp another member through the cones.

**Table 15.—Shaft Lock-nuts and Washers for Tapered Roller Bearings**

See Fig. 11. Thread tolerances and fits in accordance with A.S.A. coarse-thread series, medium fit, Class 3.

(Timken Roller Bearing Co., Canton, Ohio)

| Lock-nut Dimensions, in. | | | Shaft Dimensions, in. | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Outside Diam., *B* | Width, *C* | Threads per Inch | Outside Diam. of Threads, *A* | Thread Length, *E* | Length of Full Depth of Keyway, *F* | Width of Key-way, *G* | Depth of Key-way, *H* | Max. Diam. of Shaft, Extension *I* | Lock Washer Thickness, in. |
| 3/4 | 7/32 | 32 | 0.391 | 19/32 | 19/32 | 1/8 | 3/32 | 5/16 | 0.042 |
| 7/8 | 5/16 | 32 | .469 | 25/32 | 25/32 | 1/8 | 3/32 | 13/32 | .042 |
| 1 | 5/16 | 32 | .586 | 13/16 | 13/16 | 1/8 | 3/32 | 1/2 | .058 |
| 1 1/8 | 11/32 | 32 | .664 | 7/8 | 7/8 | 1/8 | 3/32 | 9/16 | .058 |
| 1 3/8 | 3/8 | 32 | .781 | 29/32 | 29/32 | 3/16 | 3/32 | 23/32 | .042 |
| 1 3/16 | 13/32 | 32 | .969 | 1 | 1 | 3/16 | 1/8 | 7/8 | .062 |
| 1 3/4 | 13/32 | 18 | 1.173 | 1 | 1 | 3/16 | 1/8 | 1 1/16 | .062 |
| 2 1/16 | 7/16 | 18 | 1.376 | 1 1/16 | 1 1/16 | 3/16 | 1/8 | 1 1/4 | .058 |
| 2 1/4 | 7/16 | 18 | 1.563 | 1 1/16 | 1 1/16 | 5/16 | 1/8 | 1 15/32 | .072 |
| 2 17/32 | 7/16 | 18 | 1.767 | 1 1/16 | 1 1/16 | 5/16 | 1/8 | 1 11/16 | .072 |
| 2 11/16 | 1/2 | 18 | 1.968 | 1 3/16 | 1 3/16 | 5/16 | 1/8 | 1 7/8 | .072 |
| 2 21/32 | 1/2 | 18 | 2.157 | 1 3/16 | 1 3/16 | 5/16 | 1/8 | 2 1/16 | .072 |
| 3 5/32 | 17/32 | 18 | 2.360 | 1 9/32 | 1 9/32 | 5/16 | 5/32 | 2 1/4 | .082 |
| 3 3/8 | 9/16 | 18 | 2.548 | 1 11/32 | 1 11/32 | 5/16 | 5/32 | 2 7/16 | .082 |
| 3 5/8 | 9/16 | 18 | 2.751 | 1 11/32 | 1 11/32 | 5/16 | 5/32 | 2 5/8 | .082 |
| 3 7/8 | 19/32 | 12 | 2.933 | 1 13/32 | 1 13/32 | 5/16 | 3/16 | 2 25/32 | .095 |
| 4 5/32 | 19/32 | 12 | 3.137 | 1 13/32 | 1 13/32 | 3/8 | 3/16 | 3 | .095 |
| 4 13/32 | 5/8 | 12 | 3.340 | 1 15/32 | 1 15/32 | 3/8 | 3/16 | 3 3/16 | .095 |
| 4 21/32 | 11/16 | 12 | 3.527 | 1 5/8 | 1 5/8 | 3/8 | 7/32 | 3 3/8 | .125 |
| 4 15/16 | 23/32 | 12 | 3.730 | 1 11/16 | 1 11/16 | 3/8 | 7/32 | 3 9/16 | .125 |
| 5 3/16 | 3/4 | 12 | 3.918 | 1 3/4 | 1 3/4 | 3/8 | 7/32 | 3 25/32 | .125 |
| 5 7/16 | 3/4 | 12 | 4.122 | 1 3/4 | 1 3/4 | 3/8 | 7/32 | 3 15/16 | .125 |
| 5 23/32 | 25/32 | 12 | 4.325 | 1 13/16 | 1 13/16 | 3/8 | 7/32 | 4 3/16 | .140 |
| 6 1/8 | 13/16 | 12 | 4.716 | 1 29/32 | 1 29/32 | 3/8 | 1/4 | 4 9/16 | .165 |
| 6 3/4 | 7/8 | 12 | 5.106 | 2 1/32 | 2 1/32 | 1/2 | 1/4 | 4 15/16 | .165 |
| 7 3/32 | 1 3/16 | 12 | 5.497 | 2 21/32 | 2 21/32 | 5/8 | 1/4 | 5 5/16 | .165 |
| 7 1/16 | 1 1/4 | 12 | 5.888 | 2 13/16 | 2 13/16 | 5/8 | 9/32 | 5 23/32 | .203 |
| 8 1/16 | 1 9/32 | 8 | 6.284 | 2 7/8 | 2 7/8 | 5/8 | 5/16 | 6 1/16 | .203 |
| 8 21/32 | 1 11/32 | 8 | 6.659 | 3 | 3 | 3/4 | 5/16 | 6 7/16 | .203 |
| 9 1/16 | 1 13/32 | 8 | 7.066 | 3 1/8 | 3 1/8 | 3/4 | 5/16 | 6 7/8 | .203 |
| 9 15/32 | 1 13/32 | 8 | 7.472 | 3 1/8 | 3 1/8 | 3/4 | 5/16 | 7 1/4 | .203 |
| 9 27/32 | 1 1/2 | 8 | 7.847 | 3 5/16 | 3 5/16 | 7/8 | 5/16 | 7 5/8 | .203 |

A small spacer is necessary between the front face of the cone to prevent interference by the washer with the bearing cage. The diameter of the spacer must be such as to provide cage clearance. In Design 4 the nut should provide sufficient backing to clamp the assembly solidly. Design 5 is a typical automotive application. A hardened tongued washer is used between the inner nut and cone face. The length of thread and keyway must be increased by the thickness of the hardened washer. Table 15 gives dimensions of lock-nuts and tongued washers. For data concerning the theory of tapered roller bearings and their application to various services consult the Timken Engineering Journal published by the Timken Roller Bearing Co., Canton, Ohio.

**LUBRICATION.**—Table 16 gives the specifications of lubricants recommended for tapered roller bearings by the Timken Roller Bearing Co. In general, bearings less than 6 in. outside diam., operating at speeds of less than 1000 r.p.m., without unusual conditions may be lubricated with a lime or soda base grease of medium consistency. With moisture present, only a lime base should be used. For speeds above 1000 r.p.m., oil is more satisfactory, but grease may be used on the smaller bearings. Grease is recommended for bearings over 6 in. outside diam. only up to 500 r.p.m. High-speed bearings are best lubricated with light oil and a drop feed or wick system, with provision to drain excess oil to avoid heat due to churning. The drain should be so located that the lowest point of the bearing is not immersed more than 1/8 in. A circulating oil system is recommended for conditions of extreme heat. For unusual conditions, advice as to lubrication should be obtained from the makers of the bearings.

For general machinery applications, grease should be compounded from a high grade soap and a refined mineral oil, free from acid, alkali, and fillers that would tend to score the bearing. A bright copper or steel plate, submerged in the grease for 24 hr. should show no discoloration. Ash and moisture should not exceed 2%. Oil from which the grease

### Table 16.—Lubricants for Tapered Roller Bearings

| Oil |  |  | Grease |  |  |  |
|---|---|---|---|---|---|---|
| Specification No. | Viscosity, Saybolt Universal |  | Specification No. | Base | Consistency, Karns-Maag Method |  |
|  | Seconds | Deg. F. |  |  | mm. | Deg. F. |
| 1 | 75–500 | 100 | 10 | Lime | 20–40 | 75 |
| 2 | 100–700 | 100 | 11 | Lime | 15–30 | 75 |
| 3 | 60–100 | 210 | 12 | Lime or Soda | 20–50 | 75 |
| 4 | 100 & Up | 210 | 13 | Lime or Soda | 20–40 | 75 |
| 5 | 75–300 | 100 | 14 | Soda | 15–30 | 75 |
| 6* | 75–100 | 210 | 15 | Soda; Short Fiber | 30–50 | 75 |
| 7* | 100–175 | 210 | 16 | Soda | 20–40 | 75 |
| 8* | 100–250 | 210 | 17* | Water Resistant | 20–30 | 75 |
| 9* | 175–250 | 210 | 18* | Lime or Soda | 20–30 | 75 |
| .. | ....... | ... | 19* | Water Resistant | 15–20 | 75 |
| .. | ....... | ... | 20* | Lime or Soda | 15–25 | 75 |
| .. | ....... | ... | 21* | Lime or Soda | 30–45 | 75 |

**Oil Lubrication—Recommended Specification**

| Speed, r.p.m. | Bearings up to 6 in. Outside Diam. |  |  |  | Bearings Over 6 in. Outside Diam. |  |  |  |
|---|---|---|---|---|---|---|---|---|
|  | Maximum Temperature, deg. F. |  |  |  | Maximum Temperature, deg. F. |  |  |  |
|  | 32 | 125 | 180 | Over 180 | 32 | 125 | 180 | Over 180 |
|  | Recommended Specification No. |  |  |  | Recommended Specification No. |  |  |  |
| 0–500†........ | ... | ... | ... | ... | 6‡ | 7 | 8 | 9 |
| 0–500.......... | 1 | 2 | 3 | 4 | 1‡ | 2 | 3 | 4 |
| 500–1000...... | 1 | 2 | 3 | 4 | 1 | 2 | 3 | 4 |
| Over 1000..... | 5 | 2 | 3 | 4 | 5 | 2 | 3 | 4 |

**Grease Lubrication—Recommended Specification**

| Speed, r. p. m. | Moisture Present |  |  |  |  |  |  |  | No Moisture Present |  |  |  |  |  |  |  |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  | Bearings up to 6 in. Outside Diam. |  |  |  | Bearings over 6 in. Outside Diam. |  |  |  | Bearings up to 6 in. Outside Diam. |  |  |  | Bearings over 6 in. Outside Diam. |  |  |  |
|  | Maximum Temperature, deg. F. |  |  |  |  |  |  |  | Maximum Temperature, deg. F. |  |  |  |  |  |  |  |
|  | 32 | 125 | 180 | Over 180 | 32 | 125 | 180 | Over 180 | 32 | 125 | 180 | Over 180 | 32 | 125 | 180 | Over 180 |
|  | Recommended Specification No. |  |  |  |  |  |  |  | Recommended Specification No. |  |  |  |  |  |  |  |
| 0–500†........ | .... | .... | .... | ..... | .... | 17 | 19 | ** | .... | .... | .... | ..... | 21 | 18 | 20 | ** |
| 0–500.......... | .... | 10 | 11 | ** | .... | 10 | 11 | ** | 12 | 13 | 14 | ** | 12 | 13 | 14 | ** |
| 500–1000...... | .... | 10 | 11 | ** | ** | ** | ** | ** | 12 | 13 | 14 | ** | ** | ** | ** | ** |
| Over 1000..... | .... | 10 | 11 | ** | ** | ** | ** | ** | 15 | 16 | 14 | ** | ** | ** | ** | ** |

* Extreme pressure lubricant. † Heavy duty applications. ‡ No moisture present.
** Not recommended.

is compounded should have a minimum flash point of 340° F., a minimum fire point of 380° F., and a Saybolt Universal viscosity of 200 sec. at 100° F. For heavy duty applications the greases should follow the same general specifications. The oil from which they are compounded should conform to the following: Flash point, min., 375° F.; fire point, min., 425° F.; min. Saybolt Universal viscosity, 75 sec. at 210° F.; pour test, max., 40° F.

# Section 14

# GEARING

By Earle Buckingham

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# GEARING

Rewritten by **Earle Buckingham**

## 1. FRICTION GEARING

**FRICTION GEARING.**—If two cylinders with parallel axes are in contact and one of them rotates upon its axis, the friction between the surfaces will drive the other cylinder. The driving capacity depends upon the coefficient of friction of the material of the wheels, the pressure with which they are held in contact, and the length of the cylinders. In order to secure good frictional conditions, one member only of a friction gearing is of iron; the other is of a composition giving a high coefficient of friction with iron. The composition wheel is always the driving member, and its slip on the driven tends to distribute the wear uniformly over its surface. Friction gearing cannot be used if positive driving is essential, as it is always liable to slip. It may be used where the power to be transmitted is not very great; where the speed is so high that toothed wheels would be noisy; where the shafts are frequently to be put in and out of gear or to have their relative directions of motion reversed; or where the velocity ratio is to be changed while the machinery is in motion.

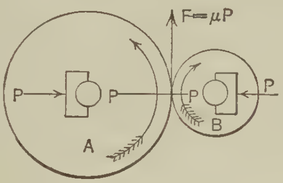

Fig. 1. Forces in Plain Friction Gearing

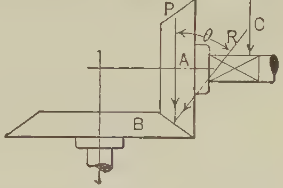

Fig. 2. Forces in Bevel Friction Gearing

**POWER TRANSMITTED BY FRICTION WHEELS.**—The tangential force that can be transmitted by two friction gears is equal to the product of the total normal pressure and the coefficient of friction. Thus, in Fig. 1 let $P$ = total normal pressure, lb.; and $\mu$ = the coefficient of friction; then, $F$, the resultant tangential force $= \mu P$. If $V$ = velocity of the surface of the friction wheels, ft. per min., the horsepower transmitted is Hp. $= \mu PV/33000$. If $d$ = diameter of the driver, in.; $l$ = length of the face, in.; $w$ = allowable load per inch of face, lb.; $N$ = revolutions per minute,

$$\text{Hp.} = (\mu wl \times 3.1416\, dN) \div (12 \times 33000) = 0.000008\, \mu wldN.$$

In *bevel wheels*, Fig. 2, the resultant $R$ of the applied force $P$ presses the wheels together, and $R = P \div \cos\theta$. The velocity of the mean circumference of the driver may be taken as the velocity of transmission.

**WEDGE-FACED OR GROOVED FRICTION WHEELS** give greater resistance to slipping with a given radial pressure. In Fig. 3, $P$ = the radial force applied to the grooved surfaces, lb.; $F$ = the tangential force transmitted, lb.; $R/2$ = the reaction on each face; and $2\theta$ = the angle of the groove; the wedge is held in equilibrium by the force $P$, the reactions $R/2$ and the frictional resistance $\mu R/2$ due to the wedging action. Then $R = F/\mu$; $P = F\sin\theta/\mu + F\cos\theta$; $F = \mu P/(\sin\theta + \mu\cos\theta)$. As before, the tangential force that can be transmitted by the gears is the product of the total normal pressure and the coefficient of friction. The number of wedges has no influence on the power transmitted, but decreases the wear by distributing it over several surfaces.

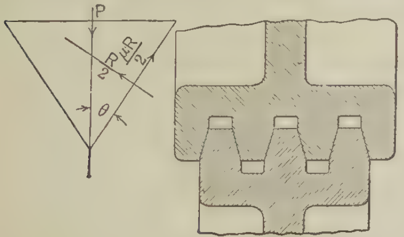

Fig. 3. Forces in Grooved Friction Wheels

The angle $2\theta$ of the grooves varies from 30° to 40°, but to avoid sticking should never be less than 30°. (See Kimball & Barr, Machine Design.)

**POWER TRANSMITTED BY FRICTION DRIVES.** (W. F. M. Goss, *Trans.* A. S. M. E., 1907.)—Driving wheels of each of the materials named in Table 1 were tested in peripheral contact with driving wheels of iron, aluminum and type metal. All the wheels were 16 in. diam.; the face of the driving wheels was 1 $^3/_4$ in., and that of the driven wheels $^1/_2$ in. Records were made of the pressure of contact, of the coefficient of friction developed, and of the percentage of slip resulting from the development of the coefficient of friction. Curves were plotted showing the relation of the coefficient and the slip for pressures of 150 and 400 lb. per inch width of face in contact. Another series of tests was made in which the slip was maintained constant at 2% and the pressures were varied. In most of the combinations it was found that with constant slip the coefficient of friction diminished very slightly as the pressure of contact was increased, so that it may be considered practically constant for all pressures between 150 and 400 lb. per sq. in.

The crushing strength of each material under the conditions of the test was determined by running each combination with increasing loads until a load was found under which the wheel failed before 15,000 revolutions had been made. The results showed the failure of the several fiber wheels under loads per inch of width as follows: Straw fiber, 750 lb.; leather fiber, 1200 lb.; tarred fiber, 1200 lb.; leather, 750 lb.; sulphite fiber, 700 lb. One-fifth of these pressures is taken as a safe working load. The coefficient of friction approaches its maximum value when the slip between driver and driven wheel is 2%. The safe working horsepower of the drive is calculated on the basis of 60% the coefficient developed at a pressure of 150 lb. per inch of width, a reduction of 40% being made to cover possible decrease of the coefficient in actual service and also to cover loss due to friction of the journals. From these data Table 1 is constructed showing the horsepower that may be transmitted by driving wheels of the several materials named when in frictional contact with iron, aluminum and type metal.

The formula for horsepower is Hp. = $(\pi d/12) \times (WPN \times 0.6f/33000) = KdWN$, in which $d$ = diam., in.; $W$ = width of face, in.; $P$ = safe working pressure, lb. per in. of width; $N$ = revs. per min.; $f$ = coefficient of friction; 0.6 = a factor for the decrease of the coefficient in service and for the loss in journal friction; $K$ = a coefficient including $P$, $f$ and the numerical constants.

**Table 1.—Coefficients of Friction and Horsepower of Friction Drives**

| | On Iron | | On Aluminum | | On Type Metal | |
|---|---|---|---|---|---|---|
| | $f$ | $K$ | $f$ | $K$ | $f$ | $K$ |
| Straw fiber | 0.255 | 0.00030 | 0.273 | 0.00033 | 0.186 | 0.00022 |
| Leather fiber | 0.309 | 0.00059 | 0.297 | 0.00057 | 0.183 | 0.00035 |
| Tarred fiber | 0.150 | 0.00029 | 0.183 | 0.00035 | 0.165 | 0.00031 |
| Sulphite fiber | 0.330 | 0.00037 | 0.318 | 0.00035 | 0.309 | 0.00034 |
| Leather | 0.135 | 0.00016 | 0.216 | 0.00026 | 0.246 | 0.00029 |

## 2. TOOTHED WHEEL GEARING

**DEFINITIONS OF TOOTH PARTS.** (See Fig. 4.)—**Addendum Circle** is the circle which limits the tops of the teeth.

**Pitch-circle** is the trace of the pitch cylinders on which the tooth curves are formed.

**Dedendum or Root Circle** is the circle which limits the bottom of the tooth.

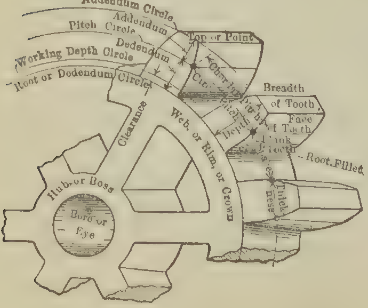

FIG. 4. Nomenclature of Gearing

**Clearance** is the amount by which the tops of the teeth of one wheel clear the bottoms of the spaces of the other, as they pass the line of centers.

**Working Depth Circle** is a circle of radius equal to that of the dedendum circle plus the clearance.

**Face** is that part of the tooth lying between the pitch and addendum circles.

**Flank** is that part of the tooth lying between the pitch and dedendum circles.

**Thickness** of the tooth is its width measured on the pitch circle.

**Width of Space** is the space between the teeth measured on the pitch-circle.

**Backlash** is the difference between the thickness of a tooth and the space into which it meshes, measured on the pitch-circles.

**Pitch Diameter** is the diameter of the pitch-circle. It is the diameter that is used in making all calculations for the size of gears.

**Circular Pitch** is the distance in inches between similar points of adjacent teeth, measured along the pitch-circle. It is the thickness of the tooth plus the width of a space, measured on the pitch-circle.

**Diametral Pitch** is the number of teeth on the gear per inch of diameter of the pitch-circle.

Circular pitch × diametral pitch = $\pi$.

**Diameter Pitch or Module** is the distance in inches obtained by dividing the diameter of the pitch-circle by the number of teeth in the gear.

**Chordal Pitch** is the length of a straight line drawn from the pitch points of two adjacent teeth. Chordal pitch = diam. of pitch-circle × sine (180° ÷ No. of teeth). Chordal pitch is used with chain or sprocket wheels, to conform to the pitch of the chain.

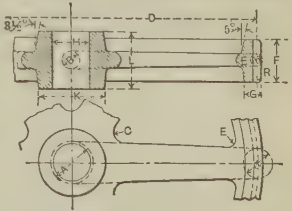

Fig. 5. Proportions of Spur Gears

**Angles of Action** are the angles through which the gears turn while a pair of teeth are in action. If the diameters are not alike, these angles will be inversely as the radii of the pitch-circles, since the arcs which subtend them are equal.

**Angle of Approach** is the angle through which a gear turns from the beginning of contact of a pair of teeth until contact reaches the pitch point.

**Angle of Recess** is the angle through which a gear turns while the contact point of a pair of teeth moves from the pitch point to the point where the teeth pass out of contact. Angle of Approach + Angle of Recess = Angle of Action.

**Line of Action** is a line normal to the tooth curves at their point of contact.

**Angle of Obliquity** is the angle which any line of action makes with the common tangent to two pitch-circles. This is constant for the involute system, and is commonly called the pressure angle.

**STANDARD PROPORTIONS FOR SPUR GEARS.** (Brown & Sharpe Mfg. Co., Providence.)—A regular system of proportioning gears must be followed, if the gears are to have good proportions, sufficient strength without excessive weight, and uniform appearance when the gears are used in sets. The following formulas, and Table 2, refer to Fig. 5, and are for gears of from 4 in. diameter, 20 diametral pitch to 72 in. diameter, 1 ½ diametral pitch. $D$ = pitch diameter, $P$ = diametral pitch, $d$ = dedendum, $f$ = addendum, and $t$ = thickness of tooth at pitch line. The significance of the other factors in the formulas is shown in Fig. 5.

**Table 2.—Values of $\{(0.95/P) + 0.28\}$ for Finding $A$**

| Diametral pitch | 1 | 1 1/2 | 2 | 2 1/2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $(0.95/P) + 0.28$ | 1.23 | 0.915 | 0.755 | 0.660 | 0.597 | 0.520 | 0.470 | 0.44 | 0.42 | 0.40 | 0.39 |
| Diametral pitch | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 |
| $(0.95/P) + 0.28$ | 0.38 | 0.37 | 0.36 | 0.354 | 0.352 | 0.345 | 0.340 | 0.337 | 0.334 | 0.332 | 0.330 |

### Formulas for Proportions of Spur Gears

Proportions of Arm.

$A = 0.138D\{(0.95/P) + 0.28\} + 0.67.$

$a = {}^2/_3 A.$

$B = 0.4A.$

$b = 0.4a$ or ${}^2/_3 B.$

$C = {}^1/_4 A.$

Proportions of Hub.*

$K = 1.5H + {}^1/_4$ in.† $L = 1.5H.$

Proportions of Rim.

* $F = (8/P) + 0.25.$

$E = {}^1/_4 A.$

$G = (d + f) + t + 0.06.$

$R = {}^1/_{32}$ in. (to 20 pitch incl.); ${}^3/_{64}$ in. (to 14 pitch incl.); ${}^1/_{16}$ in. (to 8 pitch incl.); ${}^3/_{32}$ in. (to 4 pitch incl.); ${}^1/_8$ in. (above 4 pitch).

**To Determine Whether Gears Shall Be Solid, Web, or Arm.**— Gears to be used together should be consistent as to construction. Table 3 shows the maximum diameter that should be used in each style of gear. Diameters greater than those shown in the "Web" column should be made with 6 arms, of the proportions given by formulas above.

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* Suitable for general practice. For light change gears $F$ should be less than given by formula. The hole $H$ should vary to suit conditions, but can usually be made to agree with one of the following formulas:

$$H = (4/P) + 0.5; = (5/P) + 0.5; = (6/P) + 0.5.$$

† Where heavy service is required, or large set-screws or keyways are to be used, make $K = 1.625H + {}^1/_4$ or reinforce hub over keyway or around set-screw.

Table 3.—Maximum Permissible Diameter, In., of Solid and Web Gears

| Diametral Pitch | Solid Gears |  | Web Gears |  | Diametral Pitch | Solid Gears |  | Web Gears |  |
|---|---|---|---|---|---|---|---|---|---|
|  | Change | Combination | Change | Combination |  | Change | Combination | Change | Combination |
| 4 | 5.50 | 6.00 | 9.00 | 9.50 | 10 | 3.80 | 3.80 | 5.80 | 6.00 |
| 5 | 4.80 | 5.00 | 7.50 | 8.00 | 12 | 3.67 | 3.67 | 5.67 | 5.75 |
| 6 | 4.50 | 4.75 | 7.30 | 7.75 | 14 | 3.29 | 3.29 | 5.30 | 5.50 |
| 7 | 4.30 | 4.57 | 6.86 | 7.00 | 16 | 3.13 | ...... | 4.15 | ...... |
| 8 | 4.25 | 4.25 | 6.75 | 6.88 | 18 | 3.00 | ...... | 4.55 | ...... |
| 9 | 4.00 | 4.00 | 6.67 | 6.75 | 20 | 2.80 | ...... | 4.40 | ...... |

For ordinary service, "combination" gears should be used, the "change" gears being adapted only for light service.

**GEAR WHEEL ARMS.**—The proportions of sections for gear wheel arms shown in Fig. 6 are taken from Leutwiler's Machine Design. It is also suggested that the section modulus of the arm at the center of the hub be computed by the formula $I/c = WR/nS$, where $W$ = load on the gear; $R$ = length of arm, or radius of the gear, in.; $S$ = allowable fiber stress of the material; $n$ = number of arms in the gear; $I$ = moment of inertia of section; $c$ = distance of extreme fiber from axis. This formula for the elliptical section gives $I/c = h\sqrt[3]{20WR/nS}$. The dimensions of the arm at the pitch-line can be approximately 0.7 of those at the center.

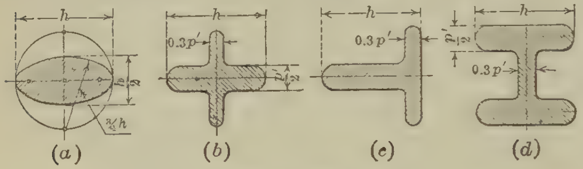

Fig. 6. Forms of Gear Wheel Arms

The number of arms suggested are: 1. For gears up to 16 to 20 in. diameter, 4 or 5 arms; 2. For gears 20 to 60 in. diameter, 6 arms; 3. For gears 60 to 96 in. diameter, 8 arms; 4. For gears over 96 in. diameter, 10 or 12 arms. For small gears web centers of thickness approximately one-half of the circular pitch are recommended.

**WIDTH OF TEETH.**—The width of the faces of teeth generally is made from 2 to 3 times the circular pitch, that is, from 6.28 to 9.42 divided by the diametral pitch. There is no standard rule for width.

The following sizes are given in a stock list of cut gears in "Grant's Gears":

|  |  |  |  |  |  |  |
|---|---|---|---|---|---|---|
| Diametral pitch.... | 3 | 4 | 6 | 8 | 12 | 16 |
| Face, inches....... | 3 & 4 | 2 1/2 | 1 3/4 & 2 | 1 1/4 & 1 1/2 | 3/4 & 1 | 1/2 & 5/8 |

The Walker Company gives:

|  |  |  |  |  |  |  |  |  |  |  |  |  |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Circular pitch, in. | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/2 | 2 | 2 1/2 | 3 | 4 | 5 | 6 |
| Face, in........ | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/2 | 4 1/2 | 6 | 7 1/2 | 9 | 12 | 16 | 20 |

## 3. FORMS OF THE TEETH

In order that the teeth of gears and pinions may run together smoothly and with a constant relative velocity, it is necessary that their working faces shall be formed of certain curves called odontoids. The essential property of these curves, of which there is an almost infinite variety, is that when two teeth are in contact the common normal to the tooth curves at their point of contact must pass through the pitch point, or point of contact of the two pitch circles. Any system of such curves can be best defined by the shape of the basic rack, *i.e.*, the rack form that is conjugate to all gears of the system. Because of its greater simplicity of production and control, the involute system, whose basic rack form comprises straight lines, is the one most widely used for power transmission. Other forms are used for special purposes. The cycloidal and pin-tooth forms are widely used in watches, clocks, etc. The four following forms have been standardized by the American Standards Assoc.

1. **THE 14 1/2-DEGREE COMPOSITE SYSTEM** was developed originally by Brown & Sharpe Mfg. Co., for use with its formed gear milling cutters, and has been commonly known as the Brown & Sharpe 14 1/2-deg. involute system. It has proved to be a very satisfactory gear-tooth system for general purposes. Fig. 7 shows the form of the basic

rack and the standard tooth proportions. Gears of this system are generally produced by formed milling cutters, but hobs made to this basic rack form are sometimes used.

**Formed Milling Cutters for 14 $^1/_2$-Degree Composite System.**—Theoretically, a different cutter is required for every different diameter of gear, even though the pitch remains the same. In practice, however, it has been found that the same cutter can be used for several sizes of gears having the same pitch, without serious error. See Table 4. By this plan 8 cutters will cut all gears of 12 teeth and over, of any one pitch. Thus, it takes 8 cutters to cut all 4-pitch gears of 12 teeth or more. A No. 8,

### Table 4.—List of Gear Cutters and Number of Teeth Cut

(Brown and Sharpe Mfg. Co., Providence, R. I.)

| Cutter No. | No. of Teeth | Cutter No. | No. of Teeth | Cutter No. | No. of Teeth | Cutter No. | No. of Teeth |
|---|---|---|---|---|---|---|---|
| 1 | 135–rack | 3 | 35–54 | 5 | 21–25 | 7 | 14–16 |
| 2 | 55–134 | 4 | 26–34 | 6 | 17–20 | 8 | 12–13 |

### Table 5.—Relation of Dimensions of Parts of Gears

(Brown & Sharpe Mfg. Co., Providence, R. I.)

**Notation**—$P$ = diametral pitch; $p$ = circular pitch; $N$ = number of teeth; $D$ = pitch diameter; $d$ = outside diameter; $a$ = addendum; $t$ = thickness of tooth at pitch-line; $r$ = root; $c$ = clearance; $W$ = working depth; $H$ = whole depth.

| Diametral Pitch | | | Circular Pitch | | |
|---|---|---|---|---|---|
| Given | To Find | Formula | Given | To Find | Formula |
| $p$ | $P$ | $P = 3.1416 \div p$ | $P$ | $p$ | $p = 3.1416 \div P$ |
| $D, N$ | $P$ | $P = N \div D$ | $D, N$ | $p$ | $p = D \div 0.3183\,N$ |
| $d, N$ | $P$ | $P = (N + 2) \div d$ | $d, N$ | $p$ | $p = d \div (0.3183\,N + 2)$ |
| $N, P$ | $D$ | $D = N \div P$ | $N, p$ | $D$ | $D = Np \times 0.3183$ |
| $N, d$ | $D$ | $D = dN \div (N + 2)$ | $N, d$ | $D$ | $D = dN \div (N + 2)$ |
| $d, P$ | $D$ | $D = d - (2 \div P)$ | $d, p$ | $D$ | $D = d - 0.6366\,p$ |
| $a, N$ | $D$ | $D = aN$ | $a, N$ | $D$ | $D = aN$ |
| $N, P$ | $d$ | $d = (N + 2) \div P$ | $N, p$ | $d$ | $d = 0.3183\,p(N + 2)$ |
| $D, P$ | $d$ | $d = D + (2 \div P)$ | $D, p$ | $d$ | $d = D + 0.6366\,p$ |
| $D, N$ | $d$ | $d = (N + 2) \div (N \div D)$ | $N, a$ | $d$ | $d = a(N + 2)$ |
| $N, a$ | $d$ | $d = (N + 2)a$ | $D, p$ | $N$ | $N = 3.1416\,D \div p$ |
| $D, P$ | $N$ | $N = DP$ | $p$ | $t$ | $t = p \div 2$ |
| $d, P$ | $N$ | $N = dP - 2$ | $p$ | $a$ | $a = D \div N = 0.3183\,p$ |
| $P$ | $t$ | $t = 1.5708 \div P$ | $p$ | $r$ | $r = a + c = 0.3683\,p$ |
| $P$ | $a$ | $a = 1 \div P = D \div N$ | $p$ | $W$ | $W = 0.6366\,p$ |
| $P$ | $r$ | $r = a + c = 1.157 \div P$ | $p$ | $H$ | $H = W + c = 0.6866\,p$ |
| $P$ | $W$ | $W = 2 \div P$ | $p$ | $c$ | $c = 0.05\,p$ |
| $P$ | $H$ | $H = W + c = 2.157 \div P$ | $t$ | $c$ | $c = 0.1\,t$ |
| $P$ | $c$ | $c = 0.157 \div P$ | | | |
| $t$ | $c$ | $c = 0.1\,t$ | | | |

### Table 6.—Diameter of Pitch-Line of Wheels from 10 to 100 Teeth of 1 in. Circular Pitch

For diameter of wheels of any other pitch than 1 in., multiply the figures in the table by the pitch. Given the diameter and the pitch, to find the number of teeth: Divide the diameter by the pitch, look in the table under diameter for the figure nearest to the quotient, and the number of teeth will be found opposite.

| No. Teeth | Diam., in. | No. Teeth | Diam., in. | No. Teeth | Diam., in. | No. Teeth | Diam., in. | No. Teeth | Diam., in. | No. Teeth | Diam., in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 3.183 | 26 | 8.276 | 41 | 13.051 | 56 | 17.825 | 71 | 22.600 | 86 | 27.375 |
| 11 | 3.501 | 27 | 8.594 | 42 | 13.369 | 57 | 18.144 | 72 | 22.918 | 87 | 27.693 |
| 12 | 3.820 | 28 | 8.913 | 43 | 13.687 | 58 | 18.462 | 73 | 23.236 | 88 | 28.011 |
| 13 | 4.138 | 29 | 9.231 | 44 | 14.006 | 59 | 18.781 | 74 | 23.555 | 89 | 28.329 |
| 14 | 4.456 | 30 | 9.549 | 45 | 14.324 | 60 | 19.099 | 75 | 23.873 | 90 | 28.648 |
| 15 | 4.775 | 31 | 9.868 | 46 | 14.642 | 61 | 19.417 | 76 | 24.192 | 91 | 28.966 |
| 16 | 5.093 | 32 | 10.186 | 47 | 14.961 | 62 | 19.735 | 77 | 24.510 | 92 | 29.285 |
| 17 | 5.411 | 33 | 10.504 | 48 | 15.279 | 63 | 20.054 | 78 | 24.828 | 93 | 29.603 |
| 18 | 5.730 | 34 | 10.823 | 49 | 15.597 | 64 | 20.372 | 79 | 25.146 | 94 | 29.921 |
| 19 | 6.048 | 35 | 11.141 | 50 | 15.915 | 65 | 20.690 | 80 | 25.465 | 95 | 30.239 |
| 20 | 6.366 | 36 | 11.459 | 51 | 16.234 | 66 | 21.008 | 81 | 25.783 | 96 | 30.558 |
| 21 | 6.685 | 37 | 11.777 | 52 | 16.552 | 67 | 21.327 | 82 | 26.101 | 97 | 30.876 |
| 22 | 7.003 | 38 | 12.096 | 53 | 16.870 | 68 | 21.645 | 83 | 26.419 | 98 | 31.194 |
| 23 | 7.321 | 39 | 12.414 | 54 | 17.189 | 69 | 21.963 | 84 | 26.738 | 99 | 31.512 |
| 24 | 7.639 | 40 | 12.732 | 55 | 17.507 | 70 | 22.282 | 85 | 27.056 | 100 | 31.831 |
| 25 | 7.958 | | | | | | | | | | |

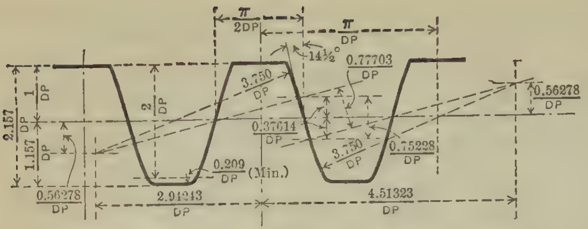

Fig. 7. Basic Rack for 14 1/2-deg. Composite System

5-pitch cutter cuts only 5-pitch gears having 12 or 13 teeth. On each cutter is stamped the numbers of teeth at the limits of its range, and the number of the cutter. When required, the error introduced by the use of range cutters can be minimized by cutting the teeth in mating gears to slightly different depths. See Chap. IV, Spur Gears, Earle Buckingham. (New York, McGraw-Hill Book Co.)

**2. THE 14 1/2-DEGREE FULL DEPTH TOOTH INVOLUTE SYSTEM** was developed for use with gear hobs, for both spur and helical gears. As the useful tooth form cannot exist below the base circle of the involute form, this system should not be used for small numbers of teeth unless a suitable change is made in the tooth proportions of mating gears. See Chap. V., Spur Gears, Earle Buckingham. (New York, McGraw-Hill Book Co.) This system has proved to be a satisfactory form for general purposes for both spur and helical gears. The form of the basic rack and the standard tooth proportions are shown in Fig. 8. The standard tooth proportions of this system are the same as those for the 14 1/2-deg. composite system, and Tables 5, 6, and 9 also apply to it. When the number of teeth in any gear of this system is less than 35, undercut will be present. When the number of teeth is less than 22, the amount of undercut may be enough to reduce the angle of action appreciably.

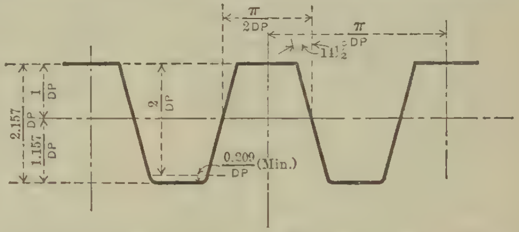

Fig. 8. Basic Rack for 14 1/2-deg. Full Depth Involute System

**3. THE 20-DEGREE FULL DEPTH TOOTH INVOLUTE SYSTEM** was developed for use with hobs and also for use with Fellows gear shapers. The increased pressure

**Table 7.—Tooth Proportions for Spur Gears.*** (See Figs. 7, 8, 9 and 10)

$N$ = Number of Teeth; $DP$ = Diametral Pitch; $CP$ = Circular Pitch

| Part | Basic Rack for 14 1/2° Full Depth Involute, 14 1/2° Composite | | Basic Rack for 20° Full Depth Involute | | Basic Rack for 20° Stub Tooth Involute | |
|---|---|---|---|---|---|---|
| | In Terms of Diametral Pitch, in. | In Terms of Circular Pitch, in. | In Terms of Diametral Pitch, in. | In Terms of Circular Pitch, in. | In Terms of Diametral Pitch, in. | In Terms of Circular Pitch, in. |
| 1. Addendum | $1/DP$ | $0.3183 \times CP$ | $1/DP$ | $0.3183 \times CP$ | $0.8/DP$ | $0.2546 \times CP$ |
| 2. Minimum Dedendum † | $1.157/DP$ | $0.3683 \times CP$ | $1.157/DP$ | $0.3683 \times CP$ | $1/DP$ | $0.3183 \times CP$ |
| 3. Working Depth | $2/DP$ | $0.6366 \times CP$ | $2/DP$ | $0.6366 \times CP$ | $1.6/DP$ | $0.5092 \times CP$ |
| 4. Minimum Total Depth † | $2.157/DP$ | $0.6866 \times CP$ | $2.157/DP$ | $0.6866 \times CP$ | $1.8/DP$ | $0.5729 \times CP$ |
| 5. Pitch Diameter | $N/DP$ | $0.3183 \times N \times CP$ | $N/DP$ | $0.3183 \times N \times CP$ | $N/DP$ | $0.3183 \times N \times CP$ |
| 6. Outside Diameter | $(N + 2)/DP$ | $0.3183 \times (N + 2) \times CP$ | $(N + 2)/DP$ | $0.3183 \times (N + 2) \times CP$ | $(N + 16)/DP$ | $(0.3183 \times N + 0.5092) \times CP$ |
| 7. Basic Tooth Thickness on Pitch Line | $1.5708/DP$ | $0.5 \times CP$ | $1.5708/DP$ | $0.5 \times CP$ | $1.5708/DP$ | $0.5 \times CP$ |
| 8. Minimum Clearance †‡ | $0.157/DP$ | $0.05 \times CP$ | $0.157/DP$ | $0.05 \times CP$ | $0.2/DP$ | |
| 9. Radius of Fillet | 1 1/3 × Clearance | | 1 1/2 × Clearance | | 1 1/2 × Clearance | |

* The term Diametral Pitch is used up to 1 $DP$ inclusive and the term Circular Pitch is used for 3 in. $CP$ and over. † A suitable working tolerance should be considered in connection with all minimum recommendations. ‡ Minimum clearance refers to the clearance between the top of the gear tooth and the bottom of the mating gear space, and is specified as *minimum* to allow for necessary cutter clearance for all methods of producing gears. At the present time (1936) this value cannot be standardized.

angle results in stronger tooth forms, and its use is rapidly increasing for general industrial purposes. It is also widely used in automotive construction. The form of the basic rack and the standard tooth proportions are shown in Fig. 9. The standard tooth proportions of this system are the same as those for the 14 1/2-deg. composite system, and Tables 5, 6, and 9 also apply to it. When the number of teeth in any gear of this system is less than 17, undercut will be present. When the number of teeth is less than 14, this undercut may be enough to reduce the angle of action appreciably.

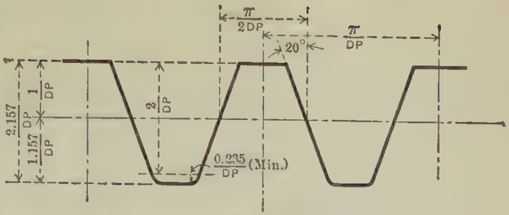

Fig. 9. Basic Rack for 20-deg. Full Depth Involute System

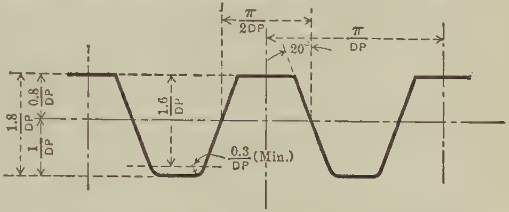

Fig. 10. Basic Rack for 20-deg. Stub Involute System

**4. THE 20-DEGREE STUB TOOTH SYSTEM** originally was developed for heavy duty, generally slow speed, machinery applications in order to obtain the strongest practical tooth forms. It was later adopted for automotive uses where the numbers of teeth were small. Several different stub tooth systems, differing but slightly from each other, have been developed. The basic rack form and tooth proportions shown in Fig. 10 will interchange and operate with all of the 20-deg. stub tooth systems in general use, the only difference being in the amount of clearance. The dimensions of the tooth parts of this system are given in Table 8.

**FELLOWS STUB TOOTH GEAR SYSTEM.**—The 20-deg. stub tooth gear system developed by the Fellows Gear Shaper Co., has been widely used for automobile trans-

Table 8.—Dimensions of Tooth Parts. 20-degree Stub Tooth System

| Diametral Pitch | Circular Pitch | Thickness of Tooth on Pitch Line | Addendum | Working Depth of Tooth | Depth of Space Below Pitch Line | Whole Depth of Tooth |
|---|---|---|---|---|---|---|
| 1 | 3.1416 | 1.5708 | 0.8000 | 1.6000 | 1.0000 | 1.8000 |
| 1 1/4 | 2.5133 | 1.2566 | .6400 | 1.2800 | 0.8000 | 1.4400 |
| 1 1/2 | 2.0944 | 1.0472 | .5333 | 1.0667 | .6667 | 1.2000 |
| 1 3/4 | 1.7952 | 0.8976 | .4571 | 0.9143 | .5714 | 1.0285 |
| 2 | 1.5708 | .7854 | .4000 | .8000 | .5000 | 0.9000 |
| 2 1/4 | 1.3963 | .6981 | .3556 | .7111 | .4444 | .8000 |
| 2 1/2 | 1.2566 | .6283 | .3200 | .6400 | .4000 | .7200 |
| 3 | 1.0472 | .5236 | .2667 | .5333 | .3333 | .6000 |
| 3 1/2 | 0.8976 | .4488 | .2286 | .4571 | .2857 | .5143 |
| 4 | .7854 | .3927 | .2000 | .4000 | .2500 | .4500 |
| 4 1/2 | .6981 | .3491 | .1778 | .3556 | .2222 | .4000 |
| 5 | .6283 | .3142 | .1600 | .3200 | .2000 | .3600 |
| 6 | .5236 | .2618 | .1333 | .2667 | .1667 | .3000 |
| 7 | .4488 | .2244 | .1143 | .2286 | .1429 | .2572 |
| 8 | .3927 | .1963 | .1000 | .2000 | .1250 | .2250 |
| 9 | .3491 | .1745 | .0889 | .1778 | .1111 | .2000 |
| 10 | .3142 | .1571 | .0800 | .1600 | .1000 | .1800 |
| 12 | .2618 | .1309 | .0667 | .1333 | .0833 | .1500 |
| 14 | .2244 | .1122 | .0571 | .1143 | .0714 | .1286 |
| 16 | .1963 | .0982 | .0500 | .1000 | .0625 | .1125 |
| 18 | .1745 | .0873 | .0444 | .0889 | .0556 | .1000 |
| 20 | .1571 | .0785 | .0400 | .0800 | .0500 | .0900 |
| 24 | .1309 | .0654 | .0333 | .0667 | .0417 | .0750 |
| 28 | .1122 | .0561 | .0286 | .0571 | .0357 | .0643 |
| 32 | .0982 | .0491 | .0250 | .0500 | .0313 | .0563 |
| 36 | .0873 | .0436 | .0222 | .0444 | .0278 | .0500 |
| 40 | .0785 | .0393 | .0200 | .0400 | .0250 | .0450 |
| 48 | .0654 | .0327 | .0167 | .0333 | .0208 | .0375 |
| 56 | .0561 | .0280 | .0143 | .0286 | .0179 | .0322 |
| 64 | .0491 | .0245 | .0125 | .0250 | .0156 | .0281 |
| 72 | .0436 | .0218 | .0111 | .0222 | .0139 | .0250 |
| 80 | .0393 | .0196 | .0100 | .0200 | .0125 | .0225 |

### Table 9.—Dimensions of Tooth Parts*

(Brown & Sharpe Mfg. Co., Providence, R. I.)

| Diametral Pitch | Circular Pitch | Thickness of Tooth on Pitch Line | Addendum and Module | Working Depth of Tooth | Depth of Space Below Pitch Line | Whole Depth of Tooth |
|---|---|---|---|---|---|---|
| $P$ | $p$ | $t$ | $a$ | $W$ | $a + c$ | $H$ |
| 1/2 | 6.2832 | 3.1416 | 2.0000 | 4.0000 | 2.3142 | 4.3142 |
| 3/4 | 4.1888 | 2.0944 | 1.3333 | 2.6666 | 1.5428 | 2.8761 |
| 1 | 3.1416 | 1.5708 | 1.0000 | 2.0000 | 1.1571 | 2.1571 |
| 1 1/4 | 2.5133 | 1.2566 | 0.8000 | 1.6000 | 0.9257 | 1.7257 |
| 1 1/2 | 2.0944 | 1.0472 | .6666 | 1.3333 | .7714 | 1.4381 |
| 1 3/4 | 1.7952 | 0.8976 | .5714 | 1.1429 | .6612 | 1.2326 |
| 2 | 1.5708 | .7854 | .5000 | 1.0000 | .5785 | 1.0785 |
| 2 1/4 | 1.3963 | .6981 | .4444 | 0.8888 | .5143 | 0.9587 |
| 2 1/2 | 1.2566 | .6283 | .4000 | .8000 | .4628 | .8628 |
| 2 3/4 | 1.1424 | .5712 | .3636 | .7273 | .4208 | .7844 |
| 3 | 1.0472 | .5236 | .3333 | .6666 | .3857 | .7190 |
| 3 1/2 | 0.8976 | .4488 | .2857 | .5714 | .3306 | .6163 |
| 4 | .7854 | .3927 | .2500 | .5000 | .2893 | .5393 |
| 5 | .6283 | .3142 | .2000 | .4000 | .2314 | .4314 |
| 6 | .5236 | .2618 | .1666 | .3333 | .1928 | .3595 |
| 7 | .4488 | .2244 | .1429 | .2857 | .1653 | .3081 |
| 8 | .3927 | .1963 | .1250 | .2500 | .1446 | .2696 |
| 9 | .3491 | .1745 | .1111 | .2222 | .1286 | .2397 |
| 10 | .3142 | .1571 | .1000 | .2000 | .1157 | .2157 |
| 11 | .2856 | .1428 | .0909 | .1818 | .1052 | .1961 |
| 12 | .2618 | .1309 | .0833 | .1666 | .0964 | .1798 |
| 13 | .2417 | .1208 | .0769 | .1538 | .0890 | .1659 |
| 14 | .2244 | .1122 | .0714 | .1429 | .0826 | .1541 |
| 15 | .2094 | .1047 | .0666 | .1333 | .0771 | .1438 |
| 16 | .1963 | .0982 | .0625 | .1250 | .0723 | .1348 |
| 17 | .1848 | .0924 | .0588 | .1176 | .0681 | .1269 |
| 18 | .1745 | .0873 | .0555 | .1111 | .0643 | .1198 |
| 19 | .1653 | .0827 | .0526 | .1053 | .0609 | .1135 |
| 20 | .1571 | .0785 | .0500 | .1000 | .0579 | .1079 |
| 22 | .1428 | .0714 | .0455 | .0909 | .0526 | .0980 |
| 24 | .1309 | .0654 | .0417 | .0833 | .0482 | .0898 |
| 26 | .1208 | .0604 | .0385 | .0769 | .0445 | .0829 |
| 28 | .1122 | .0561 | .0357 | .0714 | .0413 | .0770 |
| 30 | .1047 | .0524 | .0333 | .0666 | .0386 | .0719 |
| 32 | .0982 | .0491 | .0312 | .0625 | .0362 | .0674 |
| 34 | .0924 | .0462 | .0294 | .0588 | .0340 | .0634 |
| 36 | .0873 | .0436 | .0278 | .0555 | .0321 | .0599 |
| 38 | .0827 | .0413 | .0263 | .0526 | .0304 | .0568 |
| 40 | .0785 | .0393 | .0250 | .0500 | .0289 | .0539 |
| 42 | .0748 | .0374 | .0238 | .0476 | .0275 | .0514 |
| 44 | .0714 | .0357 | .0227 | .0455 | .0263 | .0490 |
| 46 | .0683 | .0341 | .0217 | .0435 | .0252 | .0469 |
| 48 | .0654 | .0327 | .0208 | .0417 | .0241 | .0449 |
| 50 | .0628 | .0314 | .0200 | .0400 | .0231 | .0431 |
| 56 | .0561 | .0280 | .0178 | .0357 | .0207 | .0385 |
| 60 | .0524 | .0262 | .0166 | .0333 | .0193 | .0360 |

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* To obtain the size of any part of a diametral pitch not given in the table, divide the corresponding part of 1 diametral pitch by the pitch required.

### Table 10.—Fellows Stub Tooth System

| Diametral Pitch | Thickness of Tooth, in. | Addendum, in. | Working Depth, in. | Depth of Space Below Pitch Line, in. | Clearance, in. | Whole Depth of Tooth, in. |
|---|---|---|---|---|---|---|
| 4/5 | 0.3927 | 0.2000 | 0.4000 | 0.2500 | 0.0500 | 0.4500 |
| 5/7 | .3142 | .1429 | .2858 | .1786 | .0357 | .3214 |
| 6/8 | .2618 | .1250 | .2500 | .1562 | .0312 | .2812 |
| 7/9 | .2244 | .1111 | .2222 | .1389 | .0278 | .2500 |
| 8/10 | .1963 | .1000 | .2000 | .1250 | .0250 | .2250 |
| 9/11 | .1745 | .0909 | .1818 | .1136 | .0227 | .2045 |
| 10/12 | .1571 | .0833 | .1667 | .1041 | .0208 | .1875 |
| 12/14 | .1309 | .0714 | .1429 | .0893 | .0179 | .1607 |

missions. The proportions of the teeth are based on two diametral pitches, one of which is used to obtain the dimensions of the addendum and dedendum, while the other is used for the dimensions of the pitch circle and tooth-thickness. These gears are designated by a fraction as $^4/_5$, $^{10}/_{12}$ pitch, etc. The numerator designates the pitch determining the number of teeth and tooth thickness. The denominator designates the pitch determining the height of the teeth. The clearance is (0.25/diametral pitch). Table 10 gives dimensions of Fellows stub tooth gears.

## 4. RULES FOR CALCULATING THE SPEED OF GEARS

The relations of the size and speed of driving and driven gear-wheels are the same as those of belt pulleys. In calculating for gears, multiply or divide by the diameter of the pitch-circle or by the number of teeth.

If $D$ = diam. of driving wheel, $d$ = diam. of driven, $R$ = rev. per minute of driver, $r$ = rev. per min. of driven, $RD = rd$;

$$R = rd \div D; \quad r = RD \div d; \quad D = dr \div R; \quad d = DR \div r.$$

If $N$ = No. of teeth of driver and $n$ = No. of teeth of driven, $NR = nr$;

$$N = nr \div R; \; n = NR \div r; \; R = rn \div N; \; r = RN \div n.$$

To find the number of revolutions of the last wheel at the end of a train of spur-wheels, all of which are in a line and mesh into one another, when the revolutions of the first wheel and the number of teeth or the diameter of the first and last are given: Multiply the revolutions of the first wheel by its number of teeth or its diameter, and divide the product by the number of teeth or the diameter of the last wheel.

To find the number of teeth in each wheel for a train of spur-wheels, each to have a given velocity: Multiply the number of revolutions of the driving wheel by its number of teeth, and divide the product by the number of revolutions each wheel is to make.

To find the number of revolutions of the last wheel in a train of wheels and pinions, when the revolutions of the first or driver, and the diameter, the teeth, or the circumference of all the drivers and pinions are given: Multiply the diameter, the circumference, or the number of teeth of all the driving wheels together, and this continued product by the number of revolutions of the first wheel, and divide this product by the continued product of the diameter, the circumference, or the number of teeth of all the driven wheels, and the quotient will be the number of revolutions of the last wheel.

Example.—1. A train of wheels consists of four wheels each 12 in. diameter of pitch-circle, and three pinions 4, 4, and 3 in. diameter. The large wheels are the drivers, and the first makes 36 revs. per min. Required the speed of the last wheel.

*Solution:*
$$\frac{36 \times 12 \times 12 \times 12}{4 \times 4 \times 3} = 1296 \text{ r.p.m.}$$

2. What is the speed of the first large wheel if the pinions are the drivers, the 3-in. pinion being the first driver and making 36 revs. per min.?

*Solution:*
$$\frac{36 \times 3 \times 4 \times 4}{12 \times 12 \times 12} = 1 \text{ r.p.m.}$$

## 5. ELLIPTICAL GEARS

Elliptical gears give a variable velocity ratio throughout the revolution. They represent a cheap way of obtaining a quick-return motion, and are so used on planers, shapers, slotters and similar cutting tools, and also on pumps, shears, punches and other machines which do their work mostly on the forward stroke of a reciprocating member. To determine the pitch ellipses for any given set of conditions, let $c_1 c_2$, Fig. 11, be the given distance between shaft centers. Divide $c_1 c_2$ at $p$ so that $c_1 p/p c_2$ equals the extreme velocity ratio. Lay off $pa = pc_1$ for the other focus of the lower ellipse. As the ellipses must be equal, lay off $c_1 b = c_2 a$, and $b$ is the other focus of the upper ellipse. The extremity $f$ of the minor axis is found by making $af = fc_2 = {}^1/_2\, c_1 c_2$; any other point on the ellipse, such as $d$, is found by making $ad + dc_2 = c_1 c_2$. The two free foci $a$ and $b$ remain at a constant distance apart during rotation, and are sometimes connected by a link. If $V = R^2/r^2$, $A = {}^1/_2$ major axis of the ellipse and $B = {}^1/_2$ the minor axis, then an equation which gives the ratio of the semi-major and semi-minor axes for a given range of velocity ratio is

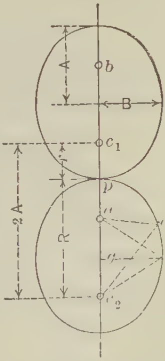

Fig. 11. Laying Out Pitch Ellipses of Elliptical Gears

$$B/A = 2\sqrt[4]{V} \div (1 + \sqrt{V}).$$

See Unwin, Machine Design, and Logue, American Machinist Gear Book.

## 6. EPICYCLIC GEAR TRAINS

Epicyclic Gear Trains may be defined as trains of wheels in which some or all of the wheels have a motion compounded of a rotation about an axis and a revolution, or translation, of that axis. (Schwamb, Merrill & James, Elements of Mechanism.) They are used to produce high velocity ratios with few parts. The link is usually called the train arm; it often rotates upon the axis of the first wheel of the train. The last wheel of the train may or may not be placed upon this axis. The name epicyclic has no connection with the form of the gear teeth which may be of the cycloidal, involute or any other system.

**METHODS OF SOLUTION OF GEAR TRAINS.**—These trains may be solved by two methods: 1. By resolving the resultant motion into its components and then allowing these to take place in succession, the motion of the arm being considered first. For example: If in Fig. 12, *A* has 100 teeth and *B* has 50; and if *A* makes +5 turns about the fixed axis *a* while the arm *D* makes −6 turns, the number of turns of *B* is found as follows:

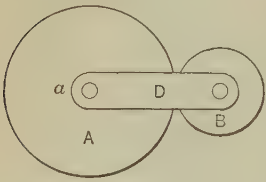

FIG. 12

Suppose the mechanism locked so that the wheels cannot turn in relation to the arm, and then turn the whole −6 turns about *a*, the number of turns which the arm is to make. Next consider the train unlocked, and put the wheel for which the motion is given, in this case *A*, where it should be. *A* is to make +5 turns, but it has been turned −6 turns with the arm when the train was locked; it must now be turned +11 in order that its resultant turns shall be +5. This will cause *B* to make −22 turns, since the value of the train (turns *B*)/(turns *A*) = − 2. Next, take the algebraic sum of the above component motions. Thus:

| | *A* | *B* | Arm |
|---|---|---|---|
| 1. Train locked........................... | − 6 | − 6 | −6 |
| 2. Train unlocked, arm fixed................ | +11 | −22 | 0 |
| 3. Resultant turns......................... | + 5 | −28 | −6 |

If in the same train it were required to find the number of turns which the arm must make in order that *B* shall make −28 turns, while *A* makes +5, let $x$ be the number of turns of the arm and proceed as above.

1. With the train locked, turn the whole $x$ turns.

2. With the train unlocked, and arm fixed, turn *A* through $(-x+5)$ turns, which will cause *B* to turn $(-2)(5-x)$.

3. Add these component motions and equate the turns of *B* thus found with the given number of turns, whence $x$ may be found. Thus:

| | *A* | *B* | Arm |
|---|---|---|---|
| 1. Train locked.................................. | $x$ | $x$ | $x$ |
| 2. Train unlocked, arm fixed........................ | $-x+5$ | $(-2)(5-x)$ | 0 |
| 3. Resultant turns................................ | $+5$ | $(-2)(5-x)+x$ | $x$ |

But turns of $B = -28 = (-2)(5-x) + x$ and $x = -6$ = turns of arm.

The second method of solving these trains is by means of a general equation which applies to any epicyclic train.

$$\frac{\text{Turns of } B \text{ relative to arm}}{\text{Turns of } A \text{ relative to arm}} = \frac{\text{absolute turns of } B - \text{turns of arm}}{\text{absolute turns of } A - \text{turns of arm}}.$$

In this equation it will be seen that the first term can always be expressed in terms of the number of teeth on the gears, as it is simply the value of the train assuming the arm fixed. It is absolutely essential that this value of the train be expressed as + or −, depending on whether the gears considered turn in the same or in opposite directions relative to the arm. Thus in the previous problem,

$$\frac{\text{Relative turns } B}{\text{Relative turns } A} = -2 = \frac{\text{absolute turns } B - \text{turns arm}}{\text{absolute turns } A - \text{turns arm}} = \frac{B-(-6)}{5-(-6)},$$

where *B* in the last term represents the absolute turns of *B*; hence

$$-2 = \frac{B+6}{5+6} \quad \text{or} \quad B = -28 \text{ turns.}$$

With the data as in the second case of the first method, where $x$ represents the turns of the arm, and *B* and *A* the absolute turns of *B* and *A*, respectively,

$$-\frac{2}{1} = \frac{B - \text{arm}}{A - \text{arm}} = \frac{-28-x}{5-x},$$

$$-10 + 2x = -28 - x, \quad \text{and} \quad x = -6 \text{ turns.}$$

**FORMULAS FOR DETERMINING THE ACTION OF EPICYCLIC TRAINS.** (F. J. Bostock, *Am. Mach.*, May 16, 1907.)—See Fig. 13. **Notation.**—$R$ = driving gear or arm; $r$ = number of teeth in driving gear; $N$ = driven gear or arm; $n$ = number of teeth in driven gear or arm; $I, S, M$ = intermediate gears; $i, s, m$ = number of teeth in intermediate gears; $F$ = fixed gears; $f$ = number of teeth in the fixed gears;

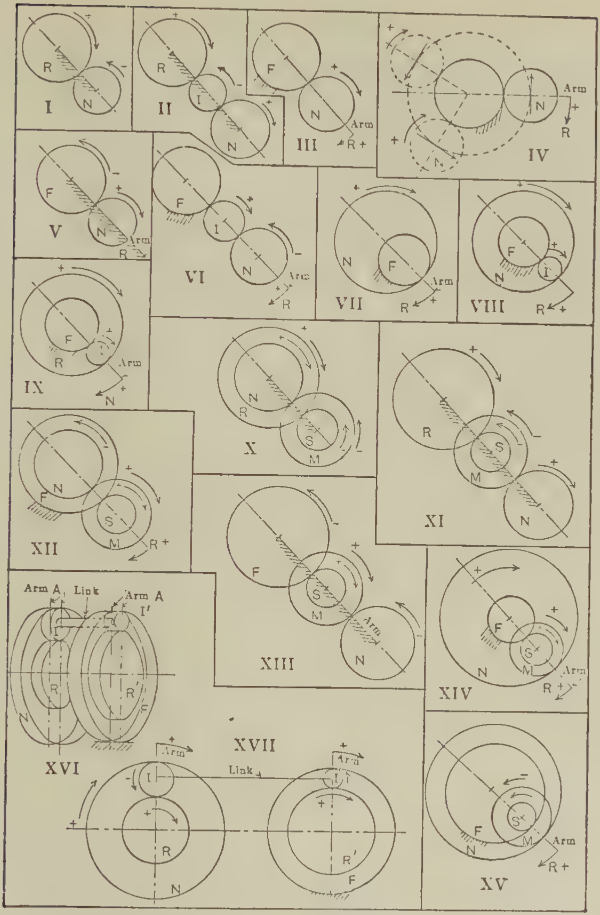

Fig. 13. Epicyclic Trains

$V_n$ = angular velocity of driven gear; $V_r$ = angular velocity of driving gear. The cross-hatching denotes that part of the apparatus that is fixed.

Formulas.—(The Roman numerals refer to the illustrations, Fig. 13.)

I. $V_n = -V_r \times (r/n)$; II. $V_n = V_r \times (r/n)$;
III. $V_n = V_r \times (1 + f/n)$; VI. $V_n = V_r \times (1 - f/n)$;
VII. $V_n = V_r \times (1 - f/n)$; VIII. $V_n = V_r \times (1 + f/n)$;
IX. $V_n = V_r \times \{r \div (r + f)\}$; X. $V_n = V_r \times (rm \div sn)$;
XII. $V_n = V_r \times \{1 - (fm \div sn)\}$; XIV. $V_n = V_r \times \{1 + (fm \div sn)\}$;
XV. $V_n = V_r \times \{1 - (fm \div sn)\}$;
XVII. $V_n = V_r \times \{(r'n - rf) \div n(r' + f)\}$.

Examples.—In Fig. 13 (III), let $f = 30$, $n = 25$; then in one revolution of $R$, $N$ will make

$$(1 + f/n) = 1 + 30/25 = 2\,{}^1/_5 \text{ revolutions.}$$

In Fig. 13 (VII), let $f = 60$, $n = 65$:

$$\frac{\text{Velocity of } N}{\text{Velocity of } R} = \frac{1 - f/n}{1} = \frac{1 - 60/65}{1} = \frac{5}{65} = \frac{1}{13}.$$

In Fig. 13 (XII), let $f = 61$, $n = 60$, $s = 40$, $m = 41$:

$$\frac{\text{Velocity of } N}{\text{Velocity of } R} = \frac{1 - (fm/sn)}{1} = \frac{1 - (61 \times 41) \div (40 \times 60)}{1} = \frac{1 - (2501/2400)}{1} = -\frac{1}{24} \text{ approx.}$$

Note that the motion of $N$ is in a negative direction.

## 7. INTERNAL GEAR DRIVES

An internal gear is one with the teeth cut into the bore instead of outside of the rim, as is the case with a spur gear. The internal gear possesses several advantages when properly applied, and is frequently used for reduction gear and planetary gear combinations. In many respects, it can be treated as a spur gear. The same basic rack forms can be used, and the mating pinion is a spur gear. The center distance on an internal gear drive is equal to the difference between the pitch radius of the internal gear and the pitch radius of the mating pinion. With an internal gear drive, both members rotate in the same direction. With its shorter center distance, the internal gear is more compact than the corresponding spur gear combination. Also, the tooth action is inherently smoother than that of a spur gear drive. The tooth forms of the internal gear are stronger than those of the corresponding spur gear; they are stronger than the basic rack tooth form. For convenience in computing their beam strength, however, it is satisfactory to use the rack tooth form factor.

When the internal gear drive is used as a reduction drive, employing a small pinion and a large internal gear, the question of design as regards the tooth shape and action does not require special consideration. It is only when the number of teeth in the pinion closely approaches the number of teeth in the gear that it is necessary to lay out the teeth to see that proper tooth action is present without interference as the teeth come out of engagement. As a general rule, the smallest permissible difference between the number of teeth in the pinion and internal gear, to avoid this interference, is 7 teeth for the stub tooth form and 12 teeth for the full depth tooth forms. Smaller differences may sometimes be used, but this requires special design of tooth forms and proportions.

Internal gears may be cut with special formed milling cutters and the use of an internal milling cutter attachment or they may be generated on a Fellows gear shaper. Such generated gears may have straight or helical teeth. See The Internal Gear, published by the Fellows Gear Shaper Company, Springfield, Vt.

## 8. STRENGTH OF SPUR GEARS

**LOAD CARRYING ABILITY OF SPUR GEARS.**—With all gear drives, there are three possible limiting factors, any one or more of which may limit the capacity of the drive. These factors are: heat of operation, beam strength of the gear teeth and surface failure, or wear of the tooth surfaces. With spur gears, heat of operation is seldom the limiting factor except when the speeds are very high or the amount of lubrication is excessive. The rate at which the heat will be dissipated from the surface of the gear case is a somewhat uncertain factor, depending largely upon the conditions of air circulation around the gear case. Under the most favorable conditions, where there is free circulation of air around the exposed surface of the case, $Q = L/35\,A$, where $Q$ = difference between room temperature and temperature of the gear case, deg. F., $L$ = power loss in ft.-lb. per min., $A$ = area of exposed surface of gear case, sq. ft. If the gear case is set in a pocket with poor circulation of air about it, then $Q = L/24\,A$. If the rise in temperature is excessive, suitable oil-cooling systems or other means of cooling should be provided.

**STRENGTH OF SPUR GEARS.**—The load carrying ability of a pair of spur gears may be limited by either the beam strength of the tooth or the surface endurance limits of the materials. The lower of these two values should be used to establish the load-carrying ability of any given drive. $W = 33{,}000\ \text{Hp.}/V$, where $W$ = transmitted pitch-line load, lb., $V$ = pitch-line velocity, ft. per min., Hp. = horsepower.

The dynamic load is the maximum momentary load set up by the operating or dynamic conditions.

$$\text{Dynamic load, } W_d = [0.05\,V(FC + W)/(0.05\,V + \sqrt{FC + W})] + W,$$

where $W_d$ = total dynamic tooth load, lb., $F$ = width of face, in., $C$ = deformation constant from Table 11.

In order to use the foregoing formula, it is necessary to have some measure of the error in action of the gears. From tests of gears made in many plants, it appears that this error on well-cut industrial gears ranges from about 0.002 in. on gears of 6-diametral pitch and finer, to about 0.004 in. on gears of 2-diametral pitch. On gears cut with particular care, these errors are reduced by about one-half the above. On ground gears made with extreme care, and on cut gears where every effort is made to secure accuracy, these errors are reduced to even smaller amounts. The foregoing conditions of error are tabulated in Table 12, and classified as Class 1, Class 2, Class 3, respectively.

Table 11.—Values of $C$*

| Materials | Tooth Form | Error in Action of Gears, in. | | | | | |
|---|---|---|---|---|---|---|---|
| | | 0.0005 | 0.001 | 0.002 | 0.003 | 0.004 | 0.005 |
| Cast iron and cast iron....... | 14 1/2° | 400 | 800 | 1600 | 2400 | 3200 | 4000 |
| Cast iron and steel.......... | 14 1/2° | 550 | 1100 | 2200 | 3300 | 4400 | 5500 |
| Steel and steel.............. | 14 1/2° | 800 | 1600 | 3200 | 4800 | 6400 | 8000 |
| Cast iron and cast iron....... | 20° f.d.† | 415 | 830 | 1660 | 2490 | 3320 | 4150 |
| Cast iron and steel.......... | 20° f.d. | 570 | 1140 | 2280 | 3420 | 4560 | 5700 |
| Steel and steel.............. | 20° f.d. | 830 | 1660 | 3320 | 4980 | 6640 | 8300 |
| Cast iron and cast iron....... | 20° sb.‡ | 430 | 860 | 1720 | 2580 | 3440 | 4300 |
| Cast iron and steel.......... | 20° sb. | 590 | 1180 | 2360 | 3540 | 4720 | 5900 |
| Steel and steel.............. | 20° sb. | 860 | 1720 | 3440 | 5160 | 6880 | 8600 |

* The values given for cast iron also are used for the ordinary semi-steels and bronze, since the modulus of elasticity of all of these materials is practically the same. † f.d. = full depth. ‡ sb. = stub.

Table 12.—Maximum Error in Action Between Gears

| Diametral Pitch | Class 1 | Class 2 | Class 3 | Diametral Pitch | Class 1 | Class 2 | Class 3 |
|---|---|---|---|---|---|---|---|
| 1 | 0.0048 in. | 0.0024 in. | 0.0012 in. | 4 | 0.0026 in. | 0.0013 in. | 0.0007 in. |
| 2 | .0040 | .0020 | .0010 | 5 | .0022 | .0011 | .0006 |
| 3 | .0032 | .0016 | .0008 | 6 and finer | .0020 | .0010 | .0005 |

As a matter of fact, noise of operation is usually a very good test of the accuracy of gears. For example, gears to run at pitch-line velocities of 5000 ft. per min. and over cannot have errors much greater than about 0.0005 in., else the noise of operation will render them unsatisfactory. Table 13 gives some measure of the accuracy required at the different velocities. This table is a guide for the selection of the proper class of gear for specific speed conditions. For extreme quietness of operation, a higher order of accuracy than shown is necessary, even at the lower speeds. The values given should keep the noise of operation and the dynamic load within reasonable limits.

Table 13.—Maximum Permissible Error in Action Between Gears

| Velocity, ft. per min. | Error, in. | Velocity, ft. per min. | Error, in. | Velocity, ft. per min. | Error, in. |
|---|---|---|---|---|---|
| 250 | 0.0037 | 1750 | 0.0017 | 3250 | 0.0009 |
| 500 | .0032 | 2000 | .0015 | 3500 | .0008 |
| 750 | .0028 | 2250 | .0014 | 4000 | .0007 |
| 1000 | .0024 | 2500 | .0012 | 4500 | .0006 |
| 1250 | .0021 | 2750 | .0011 | 5000 and over | .0005 |
| 1500 | .0019 | 3000 | .0010 | ........... | ..... |

**BEAM STRENGTH OF TEETH.**—The Lewis equation is used to determine the beam strength of the tooth. The safe static strength should not be less than the dynamic load. A margin of safety of from two to four times the applied load is desirable. Let $W_s$ = safe static load on gear teeth, lb., $p$ = circular pitch, in., $F$ = face width, in., $S$ = flexural endurance limit (stress range from zero to maximum), lb. per sq. in., $y$ = tooth form factor from Table 14. Then

$$W_s = SpFy.$$

It would appear that the flexural endurance limit of the material would be a satisfactory value to use for the limiting stress for determining the safe static load. Endurance

Table 14.—Values of $y$ in Lewis Formula

| No. of Teeth | 14 1/2° | 20° Full Depth | 20° Stub | No. of Teeth | 14 1/2° | 20° Full Depth | 20° Stub |
|---|---|---|---|---|---|---|---|
| 12 | 0.067 | 0.078 | 0.099 | 27 | 0.100 | 0.111 | 0.136 |
| 13 | .070 | .083 | .103 | 30 | .102 | .114 | .139 |
| 14 | .072 | .088 | .108 | 34 | .104 | .118 | .142 |
| 15 | .075 | .092 | .111 | 38 | .107 | .122 | .145 |
| 16 | .077 | .094 | .115 | 43 | .110 | .126 | .147 |
| 17 | .080 | .096 | .117 | 50 | .112 | .130 | .151 |
| 18 | .083 | .098 | .120 | 60 | .114 | .134 | .154 |
| 19 | .087 | .100 | .123 | 75 | .116 | .138 | .158 |
| 20 | .090 | .102 | .125 | 100 | .118 | .142 | .161 |
| 21 | .092 | .104 | .127 | 150 | .120 | .146 | .165 |
| 23 | .094 | .106 | .130 | 300 | .122 | .150 | .170 |
| 25 | .097 | .108 | .133 | Rack | .124 | .154 | .175 |

tests on steel indicate that this limit value follows quite closely the Brinell hardness number up to a stress of about 100,000 lb. per sq. in., although the structure of the material also plays an essential part. Table 15 gives suggested values to use for $S$.

**Table 15.—Values of $S$ in Lewis Formula**

| Material | Brinell Hardness No. | $S$ | Material | Brinell Hardness No. | $S$ |
|---|---|---|---|---|---|
| Gray iron......... | 160 | 12,000 | Steel............. | 280 | 70,000 |
| Semi-steel......... | 200 | 18,000 | Steel............. | 320 | 80,000 |
| Phosphor-bronze... | 100 | 24,000 | Steel............. | 360 | 90,000 |
| Steel............. | 200 | 50,000 | Steel............. | 400 and over | 100,000 |
| Steel............. | 240 | 60,000 | | | |

**LIMIT LOAD FOR WEAR.**—The limit load for wear on properly lubricated gears depends upon the surface endurance limits of the materials, the radii of curvature of the mating profiles, as well as on the hardness and structure of the surfaces of the mating teeth. A harder material in the pinion will cold-work the surfaces of a softer and malleable material of the mating gear, and thus increase its surface endurance limit materially. If $W_w$ = limiting tooth load for wear, lb. (not to be less than the dynamic load for continuous service without appreciable wear), $D$ = pitch diameter of pinion, in., $n$ = number of teeth in pinion, $N$ = number of teeth in gear, $\alpha$ = pressure angle of gears, $F$ = face width of gears, in., $Q$ = ratio factor, $K$ = load-stress factor, $E_1$, $E_2$ = moduli of elasticity of materials, $S_w$ = surface endurance limit of material, lb. per sq. in., then

$$Q = 2N/(N + n) \text{ for spur gears;}$$

$$Q = 2N/(N - n) \text{ (for internal gears);}$$

$$K = \frac{S_w{}^2 \sin \alpha}{1.40}\left(\frac{1}{E_1} + \frac{1}{E_2}\right);$$

$$W_w = DFKQ.$$

The value of $S_w$ for steel appears to vary quite consistently with its Brinell hardness numbers. The results of tests to date (1936) give, for steel, the values in Table 16.

**Table 16.—Values of $S_w$ for Steel**

| Brinell Hardness No. | $S_w$ | Brinell Hardness No. | $S_w$ | Brinell Hardness No. | $S_w$ | Brinell Hardness No. | $S_w$ |
|---|---|---|---|---|---|---|---|
| 150 | 50,000 | 300 | 110,000 | 450 | 170,000 | 550 | 210,000 |
| 200 | 70,000 | 350 | 130,000 | 500 | 190,000 | 600 | 230,000 |
| 250 | 90,000 | 400 | 150,000 | | | | |

Table 17 gives values of $K$ for various combinations of materials for 14 1/2° and 20° gears, taking into consideration the cold-working received in operation. "Cast iron" also includes the ordinary semi-steels. Some of the high-test semi-steels and other special alloys of cast iron have a greater modulus of elasticity than cast iron, and the value of $K$ for such materials must be calculated directly from their own specific properties.

**References:** Dynamic Loads on Gear Teeth, A.S.M.E. Research Publication, 1932. Am. Gear Mfrs. Assoc. Recommended Practice for Computing the Allowable Tooth Loads of Metal Spur Gears, 1932. Spur Gear Tooth Forms, B-6b-1932, Am. Stds. Assoc.

**TYPES OF PHYSICAL WEAR.**—Tests on gears and the analysis of gears badly worn in service have made apparent five distinct types of physical wear, as follows:

**Pitting,** evidenced by particles or flakes of material sheared out of the surface of the material. It is a fatigue phenomenon, and is caused by loads or pressures exceeding the surface endurance limit of the metal. The static stress conditions can be determined quite closely by the use of the Hertz equations (see p. 7–49). In operation, this problem is complicated by the tendency to plastic flow of the material. The Hertz equations for the limit load for wear apply to this type of wear only.

**Abrasion,** evidenced by the scratching or lapping of the surface of the material. It is caused by the presence of foreign matter in the lubricant, which may consist of particles of the material which have been sheared out by the pitting, or of grit picked up by the lubricant.

**Scoring,** evidenced by grooves or cuts in the surface of the material. It is caused by the projecting edges or corners of improperly finished surfaces which break through the film of lubricant and cut or score the mating surface.

Failure of the lubricant is not responsible for any of the three foregoing types of wear.

Table 17.—Values of $K$

| Pinion | | Gear | | $S_w$ | $K$, 14 1/2° | $K$, 20° |
|---|---|---|---|---|---|---|
| Material | Brinell Hardness No. | Material | Brinell Hardness No. | | | |
| Steel | 150 | Steel | 150 | 50,000 | 30 | 41 |
| Steel | 200 | Steel | 150 | 60,000 | 43 | 58 |
| Steel | 250 | Steel | 150 | 70,000 | 58 | 79 |
| Steel | 200 | Steel | 200 | 70,000 | 58 | 79 |
| Steel | 250 | Steel | 200 | 80,000 | 76 | 103 |
| Steel | 300 | Steel | 200 | 90,000 | 96 | 131 |
| Steel | 250 | Steel | 250 | 90,000 | 96 | 131 |
| Steel | 300 | Steel | 250 | 100,000 | 119 | 162 |
| Steel | 350 | Steel | 250 | 110,000 | 144 | 196 |
| Steel | 300 | Steel | 300 | 110,000 | 144 | 196 |
| Steel | 350 | Steel | 300 | 120,000 | 171 | 233 |
| Steel | 400 | Steel | 300 | 125,000 | 186 | 254 |
| Steel | 350 | Steel | 350 | 130,000 | 201 | 275 |
| Steel | 400 | Steel | 350 | 140,000 | 233 | 318 |
| Steel | 450 | Steel | 350 | 145,000 | 250 | 342 |
| Steel | 400 | Steel | 400 | 150,000 | 268 | 366 |
| Steel | 500 | Steel | 400 | 155,000 | 286 | 391 |
| Steel | 600 | Steel | 400 | 160,000 | 305 | 417 |
| Steel | 450 | Steel | 450 | 170,000 | 344 | 470 |
| Steel | 600 | Steel | 450 | 175,000 | 364 | 497 |
| Steel | 500 | Steel | 500 | 190,000 | 430 | 588 |
| Steel | 600 | Steel | 600 | 230,000 | 630 | 861 |
| Steel | 150 | Cast iron | ... | 50,000 | 44 | 60 |
| Steel | 200 | Cast iron | ... | 70,000 | 87 | 119 |
| Steel | 250 and higher | Cast iron | ... | 90,000 | 144 | 196 |
| Steel | 150 | Ph. bronze | ... | 50,000 | 46 | 62 |
| Steel | 200 | Ph. bronze | ... | 70,000 | 87 | 119 |
| Steel | 250 and higher | Ph. bronze | ... | 85,000 | 135 | 204 |
| Cast iron | ... | Cast iron | ... | 90,000 | 193 | 284 |

Scuffing, or Galling, evidenced by dragging of the surface of the material. It is caused by the plastic flow of the surface of the material because of the failure of lubrication. This is responsible for the ridges and hollows found at the pitch line of the teeth of badly worn gears made of malleable materials, as the sliding action is towards the pitch line on the driven gear and away from it on the driving pinion.

Seizing, evidenced by the actual sticking together of two surfaces or the presence of small particles of material which have been brazed or welded to the surface. It is caused by failure of lubrication accompanied by high local heating. The action of sulphur, chlorine, etc., compounded with oils to prevent seizing is primarily that of an anti-flux to prevent this welding. The successful use of sulphur in cutting compounds is primarily due to its action as an anti-flux which prevents the building-up or welding of the material to the edge of the tool.

## 9. NON-METALLIC GEARS

In order to reduce noise and vibration in high-speed gears, one gear of the pair is often made of some non-metallic material, as rawhide, fiber or phenolic materials under various trade names, as Bakelite, Micarta, Formica, Textolite, etc.

**RAWHIDE PINIONS.**—Pinions of rawhide are in common use for gearing shafts driven by electric motors to other shafts which carry machine cut gears of cast iron or steel.

**FIBER GEARS** are used for light loads and high speeds. Fiber is comparatively inexpensive, can be purchased in convenient stock sizes, but is usually difficult to machine. In milling or cutting the teeth, better results will be secured by reversing the ordinary direction of rotation of the cutting tool.

**PHENOLIC LAMINATED GEARS** are made of a non-metallic material developed for gear requirements. It is a product of special heavy duck of uniform weave, thickness and tensile strength, bonded together with a phenolic compound by heating under pressure. This material is self-supporting and does not require metal flanges. It is also water-, moisture-, steam-, acid-, and alkali-proof.

The physical characteristics of these non-metallic materials are so different from those of metals that they should be considered in a class apart by themselves. Due to their low modulus of elasticity, most of the effects of errors are absorbed at the surface of the gear teeth and have little effect on the strength of the gears. On the other hand, the deformation under load is relatively so great that the useful load they can safely carry is limited by the tendency of the corner of the metal mating gear tooth to score the flank of the non-metallic pinion tooth because of its displacement from its theoretical position by this deformation. This scoring can be minimized by making the tooth of the non-metallic pinion all-addendum, while the tooth of the metal driving gear is all-dedendum. In such a case, about double the loads can be safely carried as compared with the safe loads on gear teeth of conventional design. The 20° stub tooth system appears to be the most effective form for use with spur gears made of these non-metallic materials, for the transmission of power. If $S$ = safe static strength of material, lb. per sq. in., $s$ = safe operating strength of material, lb. per sq. in., $p$ = circular pitch, in., $y$ = tooth-form factor (Table 14), $V$ = pitch-line velocity, ft. per min., $W$ = safe working tooth load, lb., $F$ = face width, in.,

$$s = [\{150/(200 + V)\} + 0.25]S, \quad \text{and} \quad W = spFy.$$

The safe static strength of the phenolic laminated materials is generally taken as 6000 lb. per sq. in. Values of the safe working stresses of these materials at different pitch-line velocities, $V$, are tabulated in Table 18. These values apply to gears of conventional design.

**Table 18.—Safe Working Stress for Phenolic Laminated Materials**

| $V$ | $s$ | $V$ | $s$ | $V$ | $s$ |
|---|---|---|---|---|---|
| 600 | 2625 | 1800 | 1950 | 4000 | 1714 |
| 700 | 2500 | 2000 | 1909 | 4500 | 1691 |
| 800 | 2400 | 2200 | 1875 | 5000 | 1673 |
| 900 | 2318 | 2400 | 1846 | 5500 | 1658 |
| 1000 | 2250 | 2600 | 1821 | 6000 | 1645 |
| 1200 | 2143 | 2800 | 1800 | 6500 | 1634 |
| 1400 | 2063 | 3000 | 1781 | 7000 | 1622 |
| 1600 | 2000 | 3500 | 1743 | 7500 | 1617 |

The most effective use of these materials is for high-speed duty. At low speeds, where the starting torque may be high or where the load may fluctuate widely, or when high shock loads may be encountered, the non-metallic materials do not always prove satisfactory. Hardened steel appears to be the best material to run with the phenolic laminated materials. A good second choice is cast iron or semi-steel. The use of soft steel, brass or bronze is questionable because of the tendency to excessive abrasion.

## 10. EFFICIENCY OF SPUR GEARS

In 1886, Wilfred Lewis reported the results of an extensive series of tests of the efficiency of the common types of gearing, namely, worm, spiral, and spur. In these tests, which were largely confined to worm gearing, the attempt was made to measure the efficiency of spur gears, but the apparatus was not sensitive enough for the purpose, and the errors thus introduced were relatively large. The results obtained indicated a range in efficiency of from 86 to 99% for a pair of large, carefully-cut spur gears under average conditions.

Later tests indicate that the efficiency of a pair of cut gears under average conditions is in the neighborhood of 99%, and this value proves quite satisfactory for use when computing the temperature rise of operation, etc. Recent tests indicate that the efficiency is highest with the finer pitch gears. Also, that it varies with different materials, etc. As the friction loss is so small, however, the use of the value of 1% as the power loss at the tooth-mesh proves generally satisfactory for most practical purposes.

## 11. HELICAL GEARS

Helical gears are gears whose teeth are twisted uniformly throughout the length or face of the gear. The angle, at the pitch line of the helix, of the tooth with the center-line of the gear is called the helix angle. This angle represents the angle of departure of the helix from a spur gear. The complement of this angle, or the angle between the helix and any plane of rotation of the gear, is called the lead angle. The lead of the helix is the axial distance the helix advances to make one complete revolution about the pitch cylinder.

The same basic rack forms used for spur gears are often used on helical gears. In addition, many special forms and proportions are used to meet specific conditions.

Although the conjugate action between the tooth profiles in the plane of rotation is identical to that on spur gears, there is in addition a tendency for continuous action along the helices of mating teeth. In order to obtain the full advantage of this helical action, the advance of the tooth across the face of the gear must be equal to the angular distance of one tooth-interval or more. The greater the number of teeth in helical contact across the face, the smoother will be the action because of the averaging effect of the multiple-tooth contact on the errors, etc.

Gears of this type are made with helix angles up to 45°. The larger the helix angle, the greater the number of teeth in helical contact across the face of any given gear. On the other hand, the larger the helix angle, the greater the end-thrust of the gears. The final solution in any particular case is a compromise between these conflicting factors, depending upon the nature of the drive, the facilities available for resisting end-thrust, etc. The end-thrust on the bearings may be avoided by combining two pairs of helical gears, one gear with a left-hand lead of helix and the other gear on the same shaft with a right-hand lead of helix. When the two gears are made integral from a single blank, they are called herringbone gears.

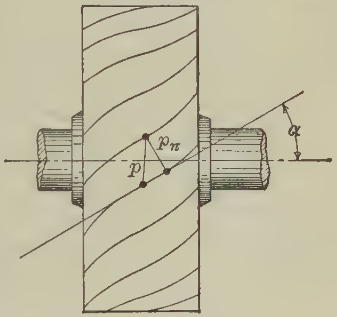

FIG. 14

**MILLED HELICAL GEARS.**—Helical gears may be produced by milling on a milling machine with the same formed milling cutters as are used for spur gears. The general formulas required for such helical gears are shown in Table 19.

**Table 19.—General Formulas for Helical Gearing**

(Adapted from Brown & Sharpe Mfg. Co., Providence, R. I.)

| To Find | Symbol | Formulas. See Fig. 14 |
|---|---|---|
| Number of teeth in gears * | $N_a, N_b$ | $N = DP_n \cos \alpha$. |
| Center distance | $C$ | $C = \frac{1}{2}(D_a + D_b) = (N_a + N_b) \div 2P_n \cos \alpha$. |
| Circular pitch (circumferential) | $p$ | $p = 3.1416\, D/N = p_n/\cos \alpha$. |
| Normal circular pitch | $p_n$ | $p_n = p \cos \alpha$. |
| Diametral pitch | $P$ | |
| Normal diametral pitch | $P_n$ | $P_n = 3.1416/p_n \cdot$(pitch of cutter). |
| Pitch diameter | $D$ | $D = pN \div 3.1416 = p_n N/3.1416 \cos \alpha = Nd \div (N + 2 \cos \alpha)$. |
| Outside diameter | $d$ | $d = D + 2a = D + (2/P_n)$. |
| Angle of teeth with axis | $\alpha$ | $\cos \alpha = N \div (P_n D)$. |
| Thickness of tooth | $t$ | $t = p_n/2$. |
| Addendum or module | $a$ | $a = p_n/3.1416 = 1/P_n = \frac{1}{2}(d - D)$. |
| Whole depth of tooth | $H$ | $H = 2a + 0.1\, t$. |
| No. of teeth marked on cutter when teeth are to be cut in milling machine † | $T$ | $T = (N/\cos^3 \alpha) + (p_n \times D_c \times \tan^2 \alpha)$ ‡ |
| Lead (exact) of spiral on pitch surface | $L_1$ | $L_1 = Np/\tan \alpha = N \times 3.1416/P \tan \alpha$. |
| Lead (approx.) of spiral on pitch surface | $L_2$ | $L_2 = 10WG_2 \div SG_1$. § |

* The subscripts $a$ and $b$ appearing in notations refer to gears $a$ and $b$.

† Cutters of composite system. See Table 4, p. 14–08.

‡ $D_c$ = diameter of milling cutter to middle of tooth profile.

§ $W$ = number of teeth on gear on worm, $G_1$, $G_2$, teeth in first and second gears on stud, respectively, $S$ = teeth in gear on screw, on spiral heads of Brown & Sharpe milling machines. The formula may not apply to spiral heads of different construction.

**HOBBED HELICAL GEARS.**—Helical gears may be hobbed, using the same hobs as are used for hobbing spur gears. There is no theoretical reason why a right-hand helical gear should not be hobbed with a left-hand hob. The use of a hob of opposite hand to that of the helical gear being cut requires the least travel of the hob into and out of the gear blank at the beginning and end of the cut. Hence if the production is large, and separate hobs are to be used for each gear of the pair, cutting time is saved by using hobs of opposite hand to the helix of the gear. For general purposes on small quantities, however, the same hob can be used for both right- and left-hand helical gears. The general formulas for these helical gears are as follows: Let $N_a$, $N_b$ = number of teeth in gears;

$C$ = center distance, in.; $p$ = circular pitch in plane of rotation, in.; $p_n$ = normal circular pitch, in.; $P$ = diametral pitch in plane of rotation; $P_n$ = diametral pitch, normal; $D$ = pitch diameter, in.; $d$ = outside diameter, in.; $\alpha$ = helix angle; $a$ = addendum, in.; $L$ = lead of helix, in. Then

$$D = N/P_n \cos\alpha; \quad d = D + 2a; \quad C = (N_a + N_b)/P_n \cos\alpha;$$
$$L = Np/\tan\alpha; \quad p = p_n/\cos\alpha; \quad P = P_n/\cos\alpha.$$

**LOAD CARRYING ABILITY OF HELICAL GEARS.**—All other factors being equal, helical gears appear to operate more smoothly and quietly than spur gears. This would indicate that the intensity of the dynamic load is less on helical gears than on spur gears. The greater possibility of multiple-tooth contact across the face of a helical gear, as compared with the contact conditions on a spur gear, undoubtedly has a large influence. The exact nature of the distribution of the load among the several teeth in simultaneous contact is still unsolved. With perfect gears and no deformation under load, it would be shared equally by all teeth in contact. With errors present, the deformation on the highest tooth must be equal to the error before another tooth can share the load. The formulas and values below appear to check conservatively with a large quantity of service data, and have been used successfully for several years.

**STRENGTH OF HELICAL GEAR TEETH.**—As with all other types of gears, the load-carrying ability of helical gears may be limited by the heat of operation, the beam strength of the teeth or the surface endurance limits of the materials. The dynamic load is the maximum momentary load set up by the operating or dynamic conditions. Where $W$ = transmitted load at pitch line, lb.; $V$ = pitch-line velocity, ft. per min.; $W_d$ = dynamic tooth load, lb.; $F$ = width of active face, in.; $C$ = deformation constant from Table 11; $\alpha$ = helix angle of gears,

$$W_d = [0.05V(W + FC\cos^2\alpha)\cos\alpha/(0.05V + \sqrt{W + FC\cos^2\alpha})] + W.$$

Helical and herringbone gears often are made with much wider faces than those commonly used on spur gears. With such wide-face gears, the load across the face is not uniform because of the torsional deformation. The following approximation will give some measure of the intensity of this load concentration at one end of the face: When $W_m$ = maximum unit dynamic load, lb. per in. of face, and $D$ = pitch diameter of pinion, in.,

$$W_m = \frac{W_d}{F}\left(1 + \frac{F}{4D}\right).$$

For herringbone gears, the value of $F$ in the above equation should be taken as the width of face of either half of the herringbone pinion because of the centering influence of the herringbone design. If the torsional deflection were uniform across the face, then the load would vary uniformly across the face, the load being greatest where the torsional deflection is least. In such a case, the average unit load would exist at the middle of the face, with a maximum and minimum unit load at opposite ends, each varying the same amount from the average. This probably is a condition that never exists because the varying load will introduce a varying torsional deflection angle. For the sake of simplicity, the foregoing approximation is based on uniform torsional deflection across the face.

**BEAM STRENGTH OF HELICAL GEAR TEETH.**—The same equations and tables as are used for calculating the beam strength of spur gears may also be used for helical gears. See pp. 14–14 to 14–17. The load conditions on helical gears are different in many respects from those on spur gears, but any error thus introduced will be on the safe side. The normal basic rack form would be used in selecting the value of $y$ from Table 14. The beam strength should be larger than the dynamic load, thus including a suitable margin of safety.

**LIMIT LOAD FOR WEAR.**—Let $D$ = pitch diameter of pinion, in.; $F$ = width of active face, in.; $K$ = load-stress factor (same as for spur gears, Table 17), $Q$ = ratio factor (same as for spur gears), $\alpha$ = helix angle, $W_w$ = limiting load for wear, lb. Then

$$W_w = DFKQ/\cos^2\alpha.$$

The limiting load for wear should not be less than the dynamic load for continuous service without appreciable wear. The types of physical wear on helical gears are identical to those on spur gears.

**NON-METALLIC HELICAL GEARS.**—The information given for spur gears of non-metallic materials may also be used for helical gears.

**EFFICIENCY OF HELICAL GEARS.**—There is some question as to the efficiency of helical gears as compared with herringbone and spur gears. Some authorities claim that herringbone gears are more efficient than spur gears, and also that herringbone gears are more efficient than helical gears because of the elimination of the end-thrust on the bearings. The amount of power loss due to end-thrust depends largely upon the efficiency of the thrust bearings. As the power loss is so small, however, the use of the value of 1% as the power loss at the tooth-mesh proves generally satisfactory for most practical purposes. For single helical gears, the power loss due to end-thrust on the bearings should be considered separately.

## 12. BEVEL GEARING

Bevel Gearing is used for connecting intersecting shafts. The teeth of bevel-gears are formed about the frustums of cones whose common vertex is the point where the shafts would meet if prolonged. These cones are called pitch-cones, since they roll upon each other and on them the teeth are pitched. When bevel-gears are the same size and the shafts are at angle of 90°, they are called *miter gears.* A *crown gear* is one in which the sides of the pitch-cone make an angle of 180° with each other. That is, the pitch-cone has no altitude and is therefore a flat surface.

**TEETH OF BEVEL-WHEELS.** (Rankine's Machinery and Millwork.)—The teeth of a bevel-wheel have acting surfaces of the conical kind, generated by the motion of a line traversing the apex of the conical pitch-surface, while a point in it is carried round the traces of the teeth upon a spherical surface described about that apex.

The operations of drawing the traces of the teeth of bevel-wheels exactly, whether by involutes or by rolling curves, are in every respect analogous to those for drawing the traces of the teeth of spur-wheels; except that in the case of bevel-wheels all those operations are to be performed on the surface of a sphere described about the apex, instead of on a plane, substituting poles for centers and great circles for straight lines. In consideration of the practical difficulty, especially in the case of large wheels, of obtaining an accurate spherical surface, and of drawing upon it when obtained, the following approximate method, proposed originally by Tredgold, is generally used:

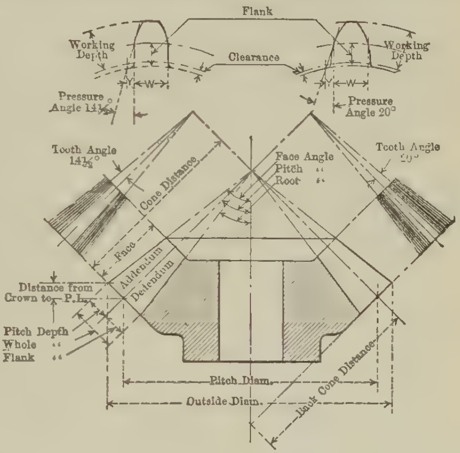

FIG. 15.—Bevel Gear Notation

Let *O*, Fig. 16, be the common apex of the pitch-cones, *OBI*, *OB′I*, of a pair of bevel-wheels; *OC*, *OC′*, the axes of those cones; *OI* their line of contact. Perpendicular to *OI* draw *AIA′*, cutting the axes in *A*, *A′*; make the outer rims of the patterns and of the wheels portions of the cones *ABI*, *A′B′I*, of which the narrow zones occupied by the teeth will be sufficiently near for practical purposes to a spherical surface described about *O*. As the cones *ABI*, *A′B′I* cut the pitch-cones at right angles in the outer pitch-circles *IB*, *IB′*, they may be called the normal cones. To find the traces of the teeth upon the normal cones, draw on a flat surface circular arcs, *ID*, *ID′*, with the radii *AI*, *A′I*; these arcs will be the development of arcs of the pitch-circles *IB*, *IB′* when the conical surfaces *ABI*, *A′B′I* are spread out flat. Describe the traces of teeth for the developed arcs as for a pair of spur-wheels, then wrap the developed arcs on the normal cones, so as to make them coincide with the pitch-circles, and trace the teeth on the conical surfaces.

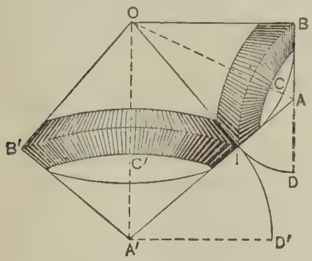

FIG. 16

For formulas and instructions for designing bevel-gears, see Practical Treatise on Gearing, and Formulas in Gearing, published by Brown & Sharpe Mfg. Co.; and Teeth of Gears, by George B. Grant. See also Rankine's Machinery and Millwork, Reuleaux's Constructor, Unwin's Elements of Machine Design, and Logue's American Machinist's Gear Book.

**FORMULAS FOR BEVEL AND ANGULAR GEARS.**—The Gleason Works, Rochester, N. Y., has published a series of rules for calculating the proportions of the various parts of bevel and angular gears. From these rules the formulas in Table 20 have been derived.

**RECOMMENDED PRACTICE FOR BEVEL GEARING.**—(Amer. Gear Mfrs. Assoc., Oct., 1919.) **Length of Face.**—The maximum length of the face of bevel gears

Table 20.—Formulas for Calculating Bevel and Angular Gears

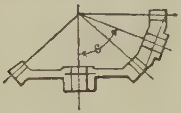
FIG. 17

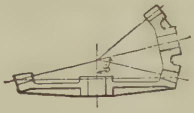
FIG. 18

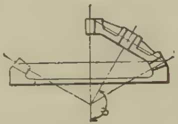
FIG. 19

| To Find | Symbol | Formulas |
|---|---|---|
| | Bevel Gears | |
| Diametral pitch | $P$ | $P = N \div D = 3.1416 \div p$. |
| Circular pitch | $p$ | $p = 3.1416 \div P = 3.1416\,D \div N$. |
| Pitch diameter | $D$ | $D = N \div P = Np \div 3.1416 = Np \times 0.3183$. |
| Pitch depth (addendum) | $a$ | $a = 1 \div P = p \times 0.3183$. |
| Working depth | $W$ | $W = 2a$; $W = a_{\text{gear}} + a_{\text{pinion}}$ (special). |
| Full depth | $H$ | $\begin{cases} H = 2a + 0.05\,p \text{ (standard)}; \\ H = a_{\text{gear}} + a_{\text{pinion}} + 0.05\,p. \end{cases}$ |
| Ratio | $V$ | $V = N \div n$. |
| No. of teeth in gear | $N$ | $\left.\begin{matrix} N \\ n \end{matrix}\right\} = PD = 3.1416\,D \div p$. |
| No. of teeth in pinion | $n$ | |
| Diameter increment | $M$ | $M = 2a \times \cos\alpha$ or $2a \times \cos\beta$. |
| Outside diameter | $d$ | $d = D + M$. |
| Pitch angle, gear | $\alpha$ | $\tan\alpha = N \div n$. |
| Pitch angle, pinion | $\beta$ | $\tan\beta = n \div N = 90 - \alpha$. |
| Cone distance at pitch-line | $C$ | $C = \frac{1}{2} D \div \sin\alpha$. |
| Cone distance at root angle | $c$ | $c = C \div \cos\phi$. |
| Back cone distance | $K$ | $K = C \tan\alpha$. |
| Inside back cone distance | $k$ | $k = C - F \tan\alpha$. |
| Face | $F$ | |
| Addendum angle | $\theta$ | $\tan\theta = a \div C$. |
| Block or dedendum angle | $\phi$ | $\tan\phi = L \div C$. |
| Flank depth | $L$ | |
| Face angle | $\psi$ | $\psi = \alpha + \theta$. |
| Root angle | $\rho$ | $\rho = \alpha - \phi$. |
| Distance crown to pitch-line | $X$ | $X = a \sin\alpha$. |
| Pitch at inside | .. | Pitch at outside : Pitch at inside :: $C : C - F$. |
| Circular thickness of teeth | $t$ | (Standard) $t = \frac{1}{2} p$. |
| Circular thickness, gear tooth | $T_g$ | (Special) see Fig. 20. |
| Circular thickness, pinion tooth | $T_p$ | $T_p = p - T_g$. |
| Chordal thickness of standard and special teeth | $S$ | See Fig. 21. |
| Corrected pitch depth | $A$ | See Fig. 22. |
| Tooth angle | $\lambda$ | $\tan\lambda = (\frac{1}{2} t + L \tan\gamma^4) \div C$. |
| Pressure angle | $\gamma$ | |
| Radius 14 1/2° base circle | $R$ | $R =$ back cone radius $\times \cos 14\frac{1}{2}°$. |
| Radius 20° base circle | $r$ | $r =$ back cone radius $\times \cos 20°$. |
| | Angular Gears | |
| CASE I. See Fig. 17<br>Pitch angle of gear,<br>[Angle of shafts ($Z$) < 90°] | $\alpha$ | $\tan\alpha = \dfrac{\sin Z}{(n/N) + \cos Z}$ |
| CASE II.* See Fig. 18<br>Pitch angle of gear,<br>[Angle of shafts ($Z$) > 90°] | $\alpha$ | $\tan\alpha = \dfrac{\sin(180° - Z)}{(n/N) - \cos(180° - Z)}$ |
| CASE III.* See Fig. 19<br>Pitch angle of gear,<br>[Angle of shafts ($Z$) > 90°] | $\alpha$ | $\tan\alpha = \dfrac{\sin(180° - Z)}{\cos(180° - Z) - (n/N)}$ |
| CASES I, II and III.<br>Pitch angle of pinion | $\beta$ | $\beta = Z - \alpha$. |

*To determine whether a given example belongs to Case II or Case III. If $N \sin(Z - 90°) < n$, use Case II; if $N \sin(Z - 90°) > n$, use Case III.

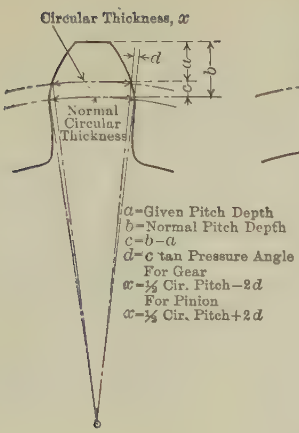

FIG. 20. Circular Thickness, Special Depths

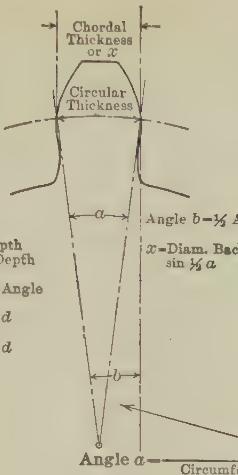

FIG. 21. Chordal Thickness, Standard and Special

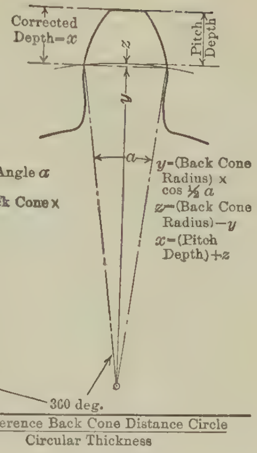
Corrected Depth=$x$
$z$
Pitch Depth
$y$
$a$
$y$=(Back Cone Radius) × cos ½ $a$
$z$=(Back Cone Radius)−$y$
$x$=(Pitch Depth)+$z$
360 deg.
Circumference Back Cone Distance Circle
Circular Thickness

FIG. 22. Corrected Pitch Depth, Standard and Special

should not be over 1/3 the cone distance for gears up to 3 in. pitch diameter and not over 1/4 the cone distance for gears from 3 to 20 in. pitch diameter, assuming that the pitch always is in proper proportion to the size of the gears. A safe rule is to make the face from 1 1/2 to 2 1/2 times the circular pitch.

**Length of Bearing.**—The minimum length of bearing along the face is to be at least 1/2 the length of the face when the gears are held in correct alignment. Gear users should test the strength of the gear housings as follows: Block the driven gear shaft and then apply a lever to the shaft of the driving gear and measure the displacement of both gears when a dead load is applied through the lever. This measurement can be made by inserting shims on the driving side of both ends of the teeth before and after the load is applied.

**Shape of Teeth for Bevel Gears.**—Bevel gears with generated involute teeth of standard addendum, having a pressure angle of 14 1/2 deg. may be used according to the following rule:

| Ratio | 1 to 1 | 1 1/2 to 1 | 2 to 1 | 3 to 1 and over |
|---|---|---|---|---|
| Minimum No. of Teeth | 14 | 18 | 19 | 21 |

The recommended practice rules as given are confined mainly to gears up to 20 in. pitch diameter as distinguished from larger gears. They also apply to average machine design as distinguished from gears for automobiles.

**SPIRAL BEVEL GEARS,** extensively used for driving the rear axles of automobiles, have teeth that are curved on the arc of a circle if produced by the Gleason spiral bevel-gear generator, or they are helical if produced on a generating-gear planer.

The following terms are used in referring to spiral bevel gears: *Angle of spiral* defines the angle that the line *AB* (Fig. 23), drawn tangent to the tooth at the center of the gear face, makes with the element *OA* of the pitch-cone. The *direction of spiral* refers to the direction of the spiral pinion; thus, a left-hand spiral is left-hand on the pinion and right-hand on the gear. *Lead* is the distance that the spiral advances within the face of the gear.

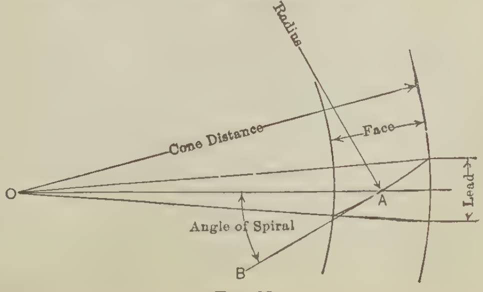

FIG. 23

For automobile practice the Gleason Works recommends gears of

4 to 5 diametral pitch and an angle of about 30°. It says that for a finer pitch, an angle of 25° to 30° will be sufficient. Although gears have been cut as low as 20° and as high as 35°, it is not advisable to go above 35°.

To obtain a sufficient lead and still keep the angle of spiral around 30°, the face may be lengthened, or, if the load is not too great, a finer pitch can be used. Either of these methods increases the amount of lead with relation to the pitch.

The teeth of spiral bevel gears wear no more than those of straight-tooth bevel gears. Due to their curvature, the teeth of spirals engage gradually and their operation is quiet. Spiral bevel pinions also permit of greater endwise adjustment than do straight-tooth bevels, and they are better adapted to high gear ratios. The load-carrying capacity of spiral-tooth bevel gears is about the same as straight-tooth bevel gears.

**END THRUST OF SPIRAL BEVEL GEARS.**—The chief disadvantage in using spiral bevel gears is the necessity of taking care of an additional end thrust on the bearings, which also may reduce the efficiency of the spiral bevel gear to slightly below that of the straight-tooth bevel gear. If the gears reverse, the thrust must be provided for in both directions. The amount of the thrust depends on several varying conditions—horsepower transmitted, pitch diameter and pitch angle of the pinion, revolutions per minute of the pinion, pressure angle and the angle of the spiral.

Where the transmitted tooth load (torque) is known, the Gleason Works gives the following approximate method for figuring the thrust: Thrust of pinion = transmitted tooth load (torque) × $C$, where $C$ is a constant, whose value is given below.

The values given for $C$ are average results from ratios between 3 to 1 and 7 to 1, and are for a pinion turning clockwise on the forward drive when viewed from the motor.

VALUES OF $C$.

| Angle of spiral................ | 20° | 25° | 30° | 35° |
|---|---|---|---|---|
| Right-hand spiral—forward..... | $C = -0.29$ | $-0.44$ | $-0.56$ | $-0.69$ |
| Right-hand spiral—reverse...... | $C = +0.41$ | $+0.59$ | $+0.72$ | $+0.85$ |
| Left-hand spiral—forward....... | $C = +0.41$ | $+0.59$ | $+0.72$ | $+0.85$ |
| Left-hand spiral—reverse....... | $C = -0.29$ | $-0.44$ | $-0.56$ | $-0.69$ |

Values with negative sign indicate pull in a direction toward the cone center; those with positive sign indicate a thrust away from the gear. The value used for transmitted tooth load or torque is to be calculated for the full pitch diameter of the pinion, not the mean diameter. The values of $C$ as given are corrected for this purpose.

EXAMPLE.—Assume 25 horsepower transmitted, 500 r.p.m. of pinion of 15 teeth, 5 diametral pitch, 3 in. pitch diameter, and 30° angle of spiral.
Transmitted tooth load $= (25 \times 33{,}000) \div \pi \times {}^3/_{12} \times 500 = 2100$ lb.
Thrust $= -0.56 \times 2100 = -1176$ lb. for right-hand spiral forward drive, and left-hand spiral reverse.
$= +0.72 \times 2100 = +1512$ lb. for left-hand spiral forward drive, and right-hand spiral reverse.

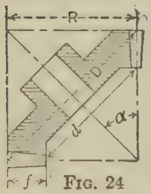

FIG. 24

**LOAD CARRYING ABILITY OF BEVEL GEARS.**—In some respects, the dynamic conditions on bevel gears are similar to those on spur gears. Because of the conical form, however, bevel gears have also many individual characteristics of their own. An axial displacement of either member of a pair from its correct position will change the contact conditions materially. Also, as the pinion usually is overhung from its bearing, bending of the pinion shaft will tend to change the contact. Bevel gears, when first set up and operated under load, seldom have complete contact across their entire face, so that the exact load conditions always are somewhat uncertain. The formulas below appear to give a reasonably close measure of the average conditions.

**STRENGTH OF BEVEL GEAR TEETH.**—The dynamic load is the maximum momentary load set up by the operating or dynamic conditions. When $W$ = transmitted load at pitch line, calculated from the pitch line velocity at the large end, lb., $V$ = pitch line velocity at large end, ft. per min., $W_d$ = dynamic tooth load, lb., $F$ = width of face, in., $C$ = deformation constant from Table 11, then,

For straight bevel gears, $W_d = [0.05\,V(W + FC)/(0.05\,V + \sqrt{W + FC})] + W$;
For spiral bevel gears, where $\alpha$ = angle of spiral,

$$W_d = [0.05\,V(W + FC\cos^2\alpha)\cos\alpha/(0.05\,V + \sqrt{W + FC\cos^2\alpha})] + W.$$

**BEAM STRENGTH OF TEETH.**—For bevel wheels Wilfred Lewis gives the following: Referring to Fig. 24, $D$ = large diameter of bevel; $d$ = small diameter of bevel; $p$ = pitch at large diameter; $n$ = actual number of teeth; $F$ = face of bevel; $N$ = formative number of teeth = $n \times$ secant $\alpha$, or the number corresponding to radius $R$; $y$ = factor depending upon shape of teeth and formative number $N$; $W$ = safe load on teeth, assumed to be applied at the large end of the bevel-gear on the pitch-line $S$ = flexural endurance limit, lb. per sq. in.

$$W = SpFy\,\frac{D^3 - d^3}{3D^2(D - d)}\;;\quad \text{or, more simply,}\quad W = SpFy\,\frac{d}{D}\;;$$

which gives almost identical results when $d$ is not less than $^2/_3\,D$, as is the case in good practice. It must be borne in mind, however, that in the case of machines which consume power intermittently, such as punching and shearing machines, the gearing should be designed with reference to the maximum load $W$, which can be brought upon the teeth at any time, and not upon the average horsepower transmitted.

Table 15 also can be used for the flexural endurance limit stresses of materials for bevel gears. The beam strength of the tooth should be enough larger than the dynamic load to give a suitable margin of safety, as in the case with spur gears.

**LIMIT LOADS FOR WEAR.**—Let $W_w$ = limiting load for wear, lb., $D$ = pitch diameter of pinion at large end, in., $F$ = face width of gear, in., $K$ = load-stress factor, Table 17, $Q$ = ratio factor, $\alpha$ = pitch angle, gear, $\beta$ = pitch angle, pinion, $N$ = number of teeth in gear, $n$ = number of teeth in pinion. Then $Q = 2N \cos\beta/(n \cos\alpha + N \cos\beta)$. For straight bevel gears, $W_w = DFKQ/\cos\beta$

For spiral bevel gears, where $\Delta$ = spiral angle,

$$W_w = DFKQ/\cos\beta \cos^2\Delta.$$

The types of physical wear on bevel gears are identical to those on spur gears.

**EFFICIENCY OF BEVEL GEARS.**—Tests on bevel gears indicate that the efficiency of well-cut and properly lubricated bevel gears ranges from about 95 to 99%. The power losses measured in these tests included the losses caused by the end thrusts, so that the design of the thrust bearings plays an important part. It is probable that the losses at the tooth-mesh alone are practically the same as for spur gears.

## 13. SCREW GEARING

Screw gearing includes those types of gears used to drive non-intersecting and non-parallel shafts, where the teeth of one or both members of the pair are of general screw form, and where the driving takes place by means of a screwing action, the sliding of the contacting tooth surfaces against each other being the predominating action. Screw gearing is of three general types: 1. Spiral gear drives; 2. Worm gear drives; 3. Hourglass or globoid worm drives. There is a fundamental difference in the nature of the geometrical construction, nature of the action and character of pitch surfaces between the three types.

**SPIRAL GEAR DRIVES.**—Helical gears used to drive non-parallel shafts commonly are known as spiral gears. In this case, both members of the pair have a uniform helical lead of tooth along their axes; each member has two pitch surfaces and also two circular pitches. One pitch surface is a cylinder; the other is a plane. The circular pitch on its pitch cylinder is the same as the circular pitch of the mating gear on its pitch plane. The circular pitch on its pitch plane equals the projection of its axial pitch along a line representing the direction of the plane of rotation of the mating gear. Thus when the two shafts are at right-angles to each other, the circular pitch on the pitch plane is equal to its axial pitch. Such pitch surfaces are shown in Fig. 25. The nature of the contact between such gears is point contact.

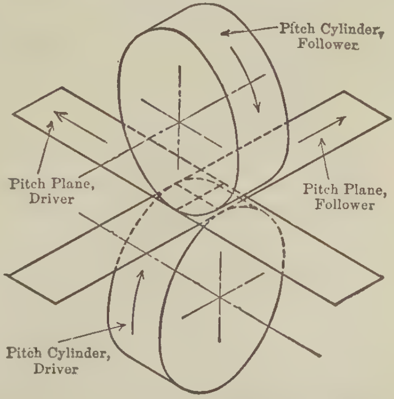

FIG. 25. Pitch Surfaces of Spiral Gears, Pitch Planes Coinciding

**WORM GEAR DRIVES.**—A common conception of a worm gear drive is that of a screw-thread driving a circular nut. A better mental image is obtained by picturing such a drive as the development of a spiral gear drive where one member has been made to envelop the other in order to obtain line contact instead of point contact. Either member may be made to envelop the other. For expediency, the more common practice has been to make the larger member envelop the smaller. Until recently, the use of a smaller member enveloping the larger one has been exceptional, and such a construction has been erroneously called an hour-glass or globoid worm drive. A worm gear drive has one member with a uniform helical lead of tooth along its axis, while the mating member is made to envelop it. The enveloping member retains its cylindrical pitch surface, but loses its pitch plane with the loss of its uniform helical lead of tooth along its axis. The helicoidal member retains its pitch plane but loses its pitch cylinder because the mating pitch plane has been eliminated. Thus the pitch surfaces of a worm drive consist of a pitch cylinder on the enveloping member and a pitch plane on the helicoidal member, and there is but one common circular pitch for both members. There can be no circular pitch in the plane of rotation of the helicoidal member unless there is a mating pitch plane with a constant axial pitch on the contacting member. The contact between the teeth of a worm gear drive is line contact.

**HOUR-GLASS OR GLOBOID WORM DRIVES.**—In an hour-glass or globoid worm drive, neither member of the pair has a uniform helical lead of tooth along its axis. One or both members may be made to envelop the other. The contact is line contact, and the pitch surfaces are indeterminate. No fixed pitch point exists. Each different design must be analyzed individually to establish the nature of the action and position of the contact. The motion is primarily that of a barrel cam operating against a series of cam followers free to rotate about an axis.

## 14. SPIRAL GEAR DRIVES

For light loads within their load-carrying capacity, spiral gears are the most satisfactory type of screw gearing. When made with reasonable accuracy, they transmit motion smoothly and quietly. This is the result of several unique characteristics of such drives. Theoretically, it is impossible to spoil the nature of the action by mislocation or misalignment. As ordinarily designed, the pitch planes of mating gears coincide but travel in different directions (see Fig. 25), depending on the direction of the planes of rotation of the two helical gears. If the axes of the gears are moved farther apart, the sizes of the pitch cylinders will remain unchanged, the corresponding pitch planes will remain tangent to their respective pitch cylinders, and will not coincide. (See Fig. 26.) The conjugate action between the tooth profiles, however, would remain theoretically correct. The gears could be so designed that the axes would be shifted to bring the entire tooth profiles between the two pitch planes, the only result being a relatively small reduction in the duration of contact.

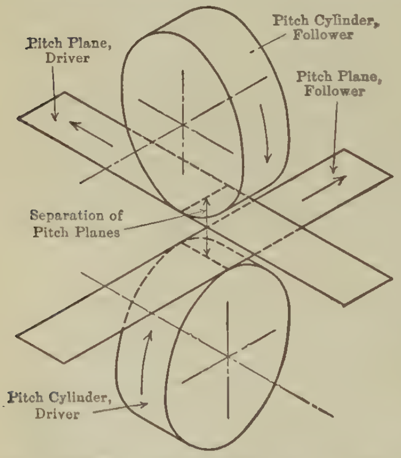

Fig. 26. Pitch Surface of Spiral Gears, Pitch Planes Separated

A similar shifting of the pitch surfaces will occur if the angular setting of the axes of the gears be changed. The circular pitches on the pitch planes then would change, as these pitches depend on the direction of the plane of rotation of the mating gear in relation to their axial position, and this also would cause the size of the mating pitch cylinder to change accordingly. The results of these characteristics are that errors in the position or alignment of involute helical gears used as a spiral gear drive have no effect on the correctness of tooth action.

Generally, the helix angles on a pair of spiral gears are of the same hand. If the angle between shafts is small, however, the helix angles may be of opposite hand. Except when both helix angles are the same, there is no direct solution for the helix angles and pitch diameters of spiral gears, unless the center distance may be varied to suit the particular combination of helix angles which are selected arbitrarily. The final result in all other cases must be found by trial. When spiral gears have helix angles of the same hand, the one with the greater helix, or lesser lead, angle always should be the driver. Except with gears of the same helix angle, the diameters of spiral gears are not directly proportional to the numbers of teeth. The speed ratio is established by the numbers of teeth and not from the diameters. Also the terms *driver* and *follower* are commonly used for spiral gears in place of the terms *pinion* and *gear*.

Let $C$ = center-distance, in., $P_n$ = normal diametral pitch (or diametral pitch of basic rack), $n$ = number of teeth in driver, $N$ = number of teeth in follower, $\alpha$ = helix angle of driver, $\beta$ = helix angle of follower, $\gamma$ = angle of axes, $d$ = pitch diameter of driver, in., $D$ = pitch diameter of follower, in. Then $d = n/P_n \cos\alpha$, and $D = N/P_n \cos\beta$.

When the two pitch planes coincide, $2C = d + D$.

When both helix angles are of the same hand, $\gamma = \alpha + \beta$.

When the helix angles are of opposite hand, $\gamma$ = difference between $\alpha$ and $\beta$.

When $L_1$ = lead of driver, in., and $L_2$ = lead of follower, in., $L_1 = 3.1416\, d/\tan\alpha$; $L_2 = 3.1416\, D/\tan\beta$.

The foregoing equations give all of the basic relationships required to solve any spiral gear problem. As noted before, in many cases the mathematical solution is indeterminate and must be solved by trial. The following simple graphical solution has been devised by J. K. Olsen, chief-draftsman of the Stewart-Warner Speedometer Corp.:

Referring to Fig. 27, the number of teeth and normal diametral pitch are selected arbitrarily. Circles are drawn representing the pitch diameters of the corresponding spur gears at the required center-distance. Two intersecting straight lines are drawn on tracing paper, the angle between them corresponding to the angle between the shafts. This tracing is then placed over the two drawn circles and adjusted until both lines are tangent to their respective circles. Their intersection is on the common center line of the two circles. The respective distances from the centers of the driver and follower to the intersection of the two lines on the common center line are the pitch radii of driver and follower. The angle between the vertical center line of each circle and the line tangent to that circle is the helix angle of that member. The values of the helix angle and pitch radii would be measured from the layout, and should be checked by the above equations. If the sum of the two pitch diameters so calculated is less than twice the specified center-distance by not more than the depth of the tooth, no further refinement is necessary. The leads should be calculated from the formulas, using the measured values of the helix angles and the calculated values of the pitch diameters. The outside diameters of the gear blanks then would be so chosen that their sum equals twice the center-distance plus twice the working depth of the tooth of the gear system used. The teeth would be cut to the standard depth in these blanks. The exact position of the tooth form in relation to the pitch cylinders is of little importance.

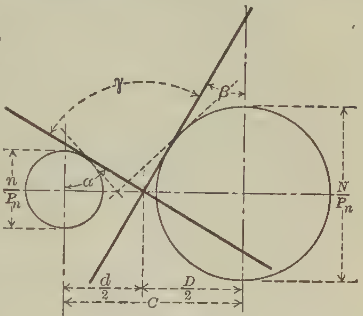

FIG. 27

Fig. 27 applies to the conditions when the two helix angles are of the same hand. In most cases, two points will be found on the common center line which meet the foregoing requirements. The point which gives the larger pitch radius and helix angle to the driver should be used. Sometimes the initial selection of tooth-numbers and pitch results either in a condition where the intersection of the two lines on the tracing paper cannot touch the common center line, or the diameters so obtained are not satisfactory. In the first case, a new selection should be made which will reduce the size of the original circles. In the second case, a new selection should be made in which the size of the circles will be increased.

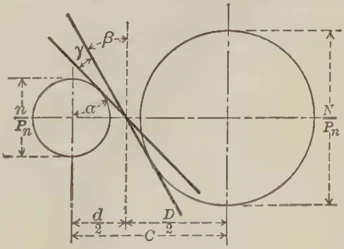

FIG. 28

When the angle between the shafts is small, and the helix angles are to be of opposite hand, the procedure is as indicated in Fig. 28. The procedure is the same as before, except that one line must be tangent to one circle *above* the common center line, while the other line must be tangent to the second circle *below* the common center line.

**TOOTH FORMS FOR SPIRAL GEARS.**—Either the 14 1/2-deg. composite system for milled gears, or the 14 1/2-deg. full depth involute system for generated gears, give the most satisfactory tooth forms for spiral gear drives. The selection of the proper milling cutter, etc., for milled spiral gears is identical to that for helical gears as shown in Table 19.

**LOAD CARRYING ABILITY OF SPIRAL GEARS.**—The nature of the contact between spiral gears is point contact. Because of this point contact, relatively light tooth loads set up very high compressive stresses in the material at the point of contact. With spiral gears, excessive wear due to pitting or abrasion is usually the most critical condition to be avoided, as the teeth will wear excessively under much smaller loads than those required to break the teeth.

Practically no reliable information is available as to the safe loads for wear on spiral gears. The limit load for wear on such gears is very uncertain. If the gears are operated in their actual working position under a light load for a period of time until a thin polished line or band appears on the teeth where contact occurs, they can then carry appreciably greater loads without excessive wear than they could if assembled and operated under full load without the preliminary polishing run. Again, if the gears are carefully run in under increasing loads until a polished band of appreciable width appears on the surfaces of the gear teeth where contact takes place, they can then carry very much higher loads without excessive wear than they can carry after only a short polishing run. In fact, the longer a pair of spiral gears run without abrasive wear on the tooth surfaces, the greater will be the loads that they can carry without excessive wear. It should be apparent, therefore, that load factors given for spiral gears depend on the care with which they have been run in after their installation. With proper care, a load that would cause excessive wear on the gears when first assembled may often be increased to several times the original limit load, because of the influence of careful running in, without resulting in appreciable wear. On the other hand, when excessive abrasive wear has once started, it is almost impossible to stop it without shifting the position of the gears and thus bringing a new part of the tooth surface into contact.

The following formulas and load factors have proved to be reasonably reliable in practice. These are based on initial running in as indicated. The relationship of the dynamic load to the transmitted load is still an open question. In general, because of the screwing action of spiral gear teeth, their operation is much smoother than that of spur or bevel gears. In the absence of definite information, the following relationship may be used: Let $W$ = transmitted tooth load, lb., $W_d$ = dynamic tooth load, lb., $V$ = pitch line velocity of follower in plane of rotation, ft. per min., $d$ = pitch diameter of driver, in., $D$ = pitch diameter of follower, in., $K$ = load factor, Table 21, $Q$ = ratio factor. Then,

$$W_d = \{(1200 + V)/1200\}\, W;\quad Q = \{(2 \times D)/(d + D)\}^2$$

$$W_w = d^2KQ, \text{ where } W_w = \text{limit load for wear, lb.}$$

**Table 21.—Values of *K* for Spiral Gears**

| Materials in Gears | After Short Polishing Run | After Careful Polishing Run |
|---|---|---|
| Steel and steel (250 Brinell) | 2 | 5 |
| Steel (250 Brinell) and bronze | 4 | 12 |
| Steel (500 Brinell) and bronze | 5 | 20 |
| Steel (500 Brinell) and cast iron | 6 | 20 |
| Steel (500 Brinell) and steel (500 Brinell) | 7 | 15 |
| Cast iron and cast iron | 8 | 20 |
| Non-metallic and steel or cast iron | 10 | 25 |

In general, to avoid sparking, bronze or non-metallic materials or hardened steel should be used when the pitch line velocity of either member in their plane of rotation is much in excess of 1000 ft. per min. Bronze never should be used in combination with the non-metallic materials for spiral gears, because of the excessive abrasion or lapping that takes place on the bronze member. In Table 21, semi-steel may be substituted for cast iron when desired.

## 15. WORM GEAR DRIVES

A worm gear drive consists of two members: 1. A helicoidal member with a uniform axial lead; 2. A member partially enveloping the helicoidal member. That is, a worm gear drive is a development of the spiral gear drive, where one member has been made to envelop the other to obtain line, instead of point, contact between the mating tooth surfaces. In most cases, worm drives are used to drive shafts at 90° to each other. They can be substituted, however, for any spiral gear drive, regardless of the angle between the shafts.

The pitch surfaces consist of a pitch plane on the helicoidal member and a pitch cylinder on the enveloping member. The conjugate tooth action best can be studied as a special case of a rack and a gear. The same conjugate tooth action exists when the helicoidal member is moved, without revolving, in an axial direction as exists when the helicoidal member revolves. The rack form, however, is not constant across the face of the enveloping member, but is of changing form. (See Fig. 29.) The amount of change in form depends upon the thread angle in the axial section, the lead of the helicoid and its diameter, and the distance of the particular section from the axial plane. Because of these conditions, higher thread angles are required for effective tooth action as the lead angle is increased. Narrower faces for the enveloping member are sometimes desirable for the higher lead angles, particularly if the thread angle in the axial section is relatively small. Thus a 14 1/2-deg. thread angle should not be used for lead angles greater than about 15 deg.; a 20-deg. thread angle should not be used for lead angles greater than about 25-deg.; a 25-deg. thread angle should not be used for lead angles greater than about 35 deg.; while a 30-deg. thread angle is desirable for lead angles up to 45 deg.

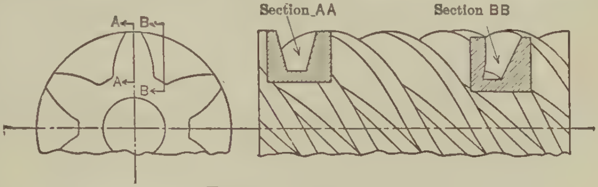

Fig. 29. Worm Gear

Many types of helicoids are used for worms. Where the lead angle is small, the actual forms produced by different methods of cutting the thread are similar, although they are different types of helicoids. As the lead angles increase, however, marked differences become apparent. For example, the axial form of the thread profile produced by a straight-sided lathe tool, tipped to the lead angle, is concave. That produced by a straight-sided milling cutter set to the lead angle generally is convex; although in extreme cases where the diameter of the cutter is small in relation to the depth of the thread and the lead angle is large, the axial form may be of double curvature, convex near the root and concave near the outside or tip of the thread. The larger the diameter of the cutter, the greater the amount of curvature in the axial profile of the thread, because of the side-cutting action of the cutter. If the milling cutter axis is parallel to the axis of the helicoid or worm, a condition that exists when hobbing a thread with an annular cutter, the form produced is that of an involute helicoid which is that of an involute helical gear. This same involute helicoid also is produced when the flat side of a cutter or grinding wheel, set to both the thread angle and lead angle, is used. Fig. 30 shows the differences in axial profile produced on a worm with a lead angle of nearly 45 deg. by different tools of the same form of cutting edge. These tools include a straight-sided threading tool set to the lead angle, a 4-in. diameter milling cutter, a 12-in. diameter grinding wheel, and the involute helicoid which would be produced by the flat side of a milling cutter or grinding wheel.

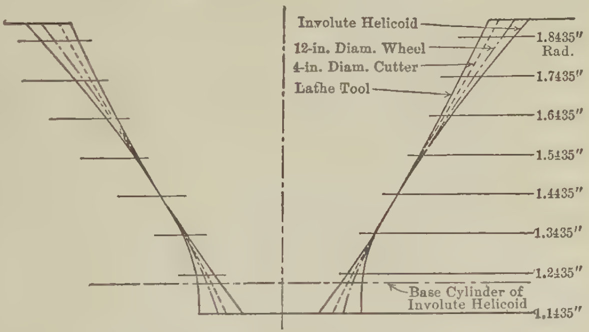

Fig. 30. Axial Sections of Milled and Chased Worms

All of these forms of helicoids have been commonly used as thread forms for worms. As far as theoretical conditions are involved, one type does not appear to have any great inherent advantage over another. The major consideration is that the same form of helicoid as is used for the worm should be reproduced on the cutting tool which is to be used to generate the enveloping member. In general, the best form to use is largely a matter of expediency, controlled by the type of equipment available to produce the helicoidal member. Hence the tool-maker must know the method to be used to produce the helicoidal member in order to make correctly the cutting tool which is to be used to generate the enveloping member.

For many practical reasons, the involute helicoid would appear to be the best form to use where the design is not restricted by existing equipment. With this helicoid, the same hob or generating cutter which is used to generate the enveloping member also may be used to generate the worm or helicoidal member. Again, with this helicoid, the helicoidal members would be involute helical gears, and the same hob or other generating cutter could be used to generate all worms of the same normal diametral pitch of all tooth-numbers. Under such conditions, the practice and technique already developed in the making of tools for helical and spur gears could be applied directly to the making of tools for worms and worm gears. Furthermore, this involute helicoidal form, where ground worms are required, is the simplest one to grind and control, the size of the wheel has no influence on the form produced, and several different grinding methods are available.

In the further discussion, the angle of the thread form will refer to the tangent to the thread profile at or near the middle of the working depth, regardless of the type of helicoid involved.

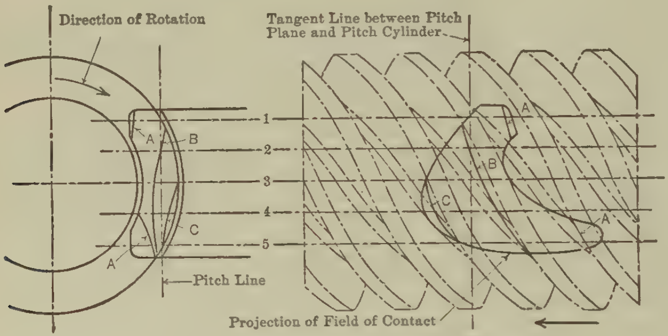
Direction of Rotation
Tangent Line between Pitch Plane and Pitch Cylinder
A
B
C
A
1
2
3
4
5
Pitch Line
A
B
C
A
Projection of Field of Contact

Fig. 31. Contact Lines and Field of Contact

**CONTACT ON WORM GEAR DRIVES.**—The analysis of worm drives is a very complex operation. In the majority of cases, where such analyses have been made, only the conjugate action between parallel sections of the worm, treating them as rack sections, meshing with the enveloping member or gear, have been considered. In addition, it is necessary to have some measure of the nature and position of the actual contact lines between mating teeth, duration and field of contact between them, in order to determine even approximately the intensity of the stresses and probable load-carrying capacities of such drives. Fortunately all types of helicoids follow certain general laws so that workable approximations are possible in all except the extremely critical conditions.

Considerable uncertainty exists as to the most effective position of the pitch plane in relation to the tooth on the worm. The conventional design places it at the middle of the working depth of the tooth or thread on the worm. The practice in some plants is to place it nearer the root of the thread. On all types of helicoids, the tendency of the contact lines is to converge towards the pitch line on the leaving side of the worm. (See Fig. 31.) This shows a 6-thread worm with straight sides of thread in an axial section, meshing with a 40-tooth gear. The heavy outline on the plan view of the worm shows the projection of the field of contact. In the relative position of the two members as shown, contact exists on three threads of the worm as shown by heavy lines *A*, *B*, and *C*.

Using this same worm, when the pitch line is dropped to the outside diameter of the gear, by simply closing in the center distance the amount that the pitch plane is moved, the contact conditions are as shown in Fig. 32. Here the contact lines tend more to sweep from the bottom to the top of the worm thread. This change in position of the contact lines causes much lower stresses to be set up in the materials by the same applied load, as compared with the conditions in Fig. 31. Because of the relative curvature conditions at the contact lines the drive, Fig. 32, can safely carry about double the loads that the drive, Fig. 31, can carry, even with a portion of the gear face undercut as shown. With lower lead angles, the change is less with a change in the position of the pitch plane.

In the design of a worm drive, the center distance can be varied a limited amount, and the pitch plane may be placed anywhere between the middle of the working depth of the thread on the worm and the outside diameter of the gear to suit other conditions, such as a prefixed center distance. On the other hand, the lower the pitch plane is placed on the worm thread, the greater will be its load-carrying capacity and the smoother the action.

**DESIGN OF WORMS.**—Considerable difference of opinion exists as to whether the form of the thread in the axial section or that in the so-called "normal section" should be specified. As a matter of fact, there is no "normal section" as generally implied on any helicoid. As noted before, the essential information is best specified by stating the shape, type and size of the tool to be used to finish the worm. The Am. Gear Mfrs. Assoc. recommends that the thread form to be regarded as standard will be the form produced by a straight-sided milling cutter having a diameter not less than the outside diameter, nor greater than $1\frac{1}{4}$ times the outside diameter, of the worm; also, that the following linear pitches be regarded as standard: $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$, $\frac{3}{4}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{3}{4}$, 2 in.

Since the pitch surface of the worm or helicoidal member is a plane, and since the pitch on such a plane is constant at all distances from the axis, the term *pitch diameter of a worm* is misleading. Considering it as on a screw-thread, where the same term is used to define that diameter where the thread thickness is equal to the width of space, or where both distances are equal to one-half the pitch, it might best be called the nominal pitch diameter, or better, the effective diameter.

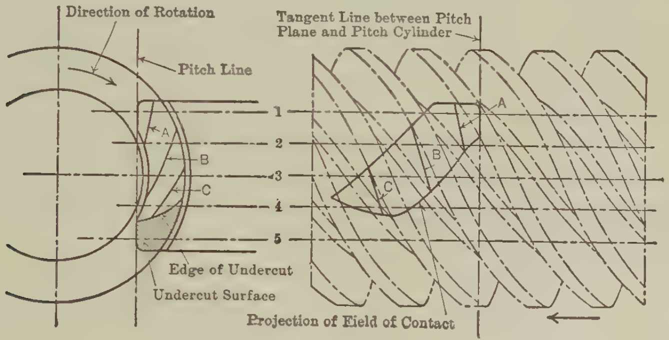
Direction of Rotation
Tangent Line between Pitch Plane and Pitch Cylinder
Pitch Line
A
B
C
1
2
3
4
5
A
B
C
Edge of Undercut
Undercut Surface
Projection of Field of Contact

FIG. 32. Contact Lines and Field of Contact

The Am. Gear Mfrs. Assoc. also recommends that the top edge of the worm thread should always be rounded to avoid cutting, and that the amount of this rounding be made equivalent to 0.05 linear pitch. When the pitch plane of the worm is dropped to about its root diameter or lower, and the worm is the driving member, the conjugate tooth action at this edge always is recess action and under such conditions this rounding is not necessary.

The Am. Gear Mfrs. Assoc. recommends the use of a $14\frac{1}{2}$-deg. and a 20-deg. thread angle for worms from single to quadruple threads. As noted before, the lead angle and not the number of threads is the major factor that should determine the selection of the thread angle. To these two angles could be added a 25-deg. and a 30-deg. thread angle to cover all lead angles up to 45 deg. or higher. If a single thread angle is to be used, the 30-deg. thread angle can be effectively used on the lower lead angles, and several manufacturers of worm gears have adopted this angle as standard for all worms, with very satisfactory results.

**MILLED AND CHASED WORMS.**—For milled and chased worms, where definite axial pitches are maintained, a choice between two procedures is necessary: 1. If fixed tooth heights are used for the same pitch, regardless of the lead angle, a different cutter will be required for each different lead angle. 2. If the same milling cutter or chasing tool is to be used for all worms of the same axial pitch, regardless of the lead angle, a different tooth height must be calculated for each lead angle. In general, as regards the standardization of gears, greatest benefits and economies will be secured by standardizing the tools used to produce them, together with the reduction of variety as far as possible. The use of a single cutter for all worms of a given pitch evidently would be the most economical procedure. In such a case, let $p$ = axial pitch of worm, in., $\alpha$ = half included angle of cutter, $\beta$ = lead angle of worm, $L$ = lead of worm, in., $f$ = width of flat at tip of cutter, in., $c$ = clearance, in., $D$ = nominal pitch diameter of worm, in., $a$ = nominal

addendum, in., $W$ = working depth of thread, in., $H$ = whole depth of thread, in., $d$ = outside diameter of worm, in., $r$ = root diameter of worm, in.
Then, when $c = a/6$,

$$W = \{3/(7 \times \tan\alpha)\}(p\cos\beta - 2f);\ a = W/2 = \{3/(14\tan\alpha)\}(p\cos\beta - 2f);$$

$$H = W + c = 13\,W/12.$$

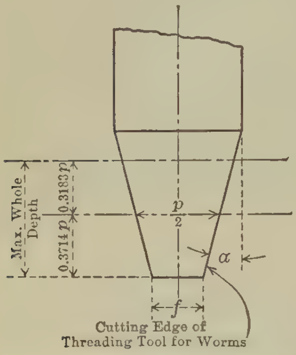

FIG. 33. Forms of Threading Cutters

Fig. 33 and Table 22 show satisfactory forms for threading cutters of the several thread angles.

For worms which are to be made integral with the shaft, for lead angles less than 45 deg., and where a shank hob is used to generate the gear, the formula, $D = 2p + 0.5$, may be used. The Am. Gear Mfrs. Assoc. recommends for worms which are to be mounted on the shaft, $D = 2.4\,p + 1.1$. In both cases, $d = D + W$; $\tan\beta = L/3.1416\,D$; $r = d - 2H$.

**HOBBED WORMS.**—With the increasing use of multiple-threaded worms, where the number of threads on the worm may be 24 or more, the hobbing of worms is growing in favor. In this case, there is no reason why the standard spur and helical gear hobs should not be used to produce such worms, where lead angles are low. For higher lead angles, involute gear hobs with a pressure angle of 30 deg. would prove effective. The form of the basic rack could be the same as shown for the form of the 30-deg. threading tool in Fig. 33. The sizes of such worms would be computed exactly the same as for helical gears, and the generating tool for the worm gear should be made to suit the worm thread form.

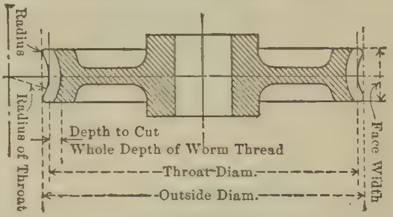

FIG. 34. Worm Gear

**DESIGN OF WORM GEARS.**—The pitch diameter of the worm gear depends on the number of teeth in the gear and the axial pitch of the worm. Let $D_g$ = pitch diameter of worm gear, in.; $N$ = number of teeth in gear; $p$ = circular pitch (axial pitch of worm), in. Then $D_g = pN/3.1416$. All other dimensions of the worm gear depend primarily upon the size of the worm and the center distance. Referring to Fig. 34, when $D$ = nominal pitch diameter of worm, in.; $r$ = root diameter of worm, in.; $c$ = clearance, in.; $W$ = working depth of worm thread, in.; $C$ = center distance, in.,

$$\text{Throat diameter} = 2C - (r + 2c);$$

$$\text{Throat radius} = (r/2) + c;$$

$$\text{Outside diameter} = \text{throat diameter} + (W/2).$$

Face width varies from 0.70 $D$ to 0.50 $D$, depending on the rubbing speed of the worm. With increasing rubbing speeds, the face diameter should be reduced. The pitch diameter of the worm gear should lie between the diameter equal to $(2C - D)$ and the outside diameter of the worm gear. As noted before, the lower the pitch plane is placed on the worm thread, the greater is the load-carrying capacity and the smoothness of the action.

Table 22.—Dimensions of Threading Cutters (See Fig. 33).

| Thread Angle, α | 14 1/2° | 20° | 25° | 30° |
|---|---|---|---|---|
| Tan α | 0.25862 | 0.36397 | 0.46631 | 0.57735 |
| Maximum whole depth | 0.6897$p$ | 0.6897$p$ | 0.6897$p$ | 0.6897$p$ |
| Maximum lead angle, β | 15° | 25° | 35° | 45° |
| Width of flat, $f$ | 0.3079$p$ | 0.2297$p$ | 0.1537$p$ | 0.0712$p$ |
| $2f$ | 0.6158$p$ | 0.4594$p$ | 0.3074$p$ | 0.1424$p$ |
| $3/(14\tan\alpha)$ | 0.82857 | 0.58875 | 0.45953 | 0.37115 |

In general, the pitches commonly used on worm drives are much coarser than are required. The multiple threaded and finer pitch worms are more efficient and run more smoothly than coarser pitch worms. To obtain the most effective worm drive, the pitch should be as fine as the strength of the gear teeth permit. For effective tooth action and full contact across the face of the worm gear, the sum of the number of teeth in the worm and worm gear should be not less than about forty.

**HOBS FOR WORM GEARS.**—The hob for a worm gear should be substantially a duplicate of the mating worm, but two precautions must be observed: 1. The nominal pitch diameter of the hob never must be less than that of the mating worm. If the hob is smaller, contact will be concentrated on the edges of the face of the worm gear, with attendant cutting or scoring and rapid wear. The amount which the hob can be made oversize depends largely on the lead angle of the worm. With low lead angles, it can be made appreciably larger, but with the high lead angles, it can be made only a very small amount oversize, unless the axis of the hob is to be twisted slightly in relation to the normal position of the axis of the worm. Such twisting will throw the initial contact towards the center of the worm face. 2. The length of the hob never should be less than the length of the mating worm in order to fully generate the contacting surfaces. This applies particularly to cases where relatively short worms are used. In general, a hob whose length is equal to 6 or 7 times the axial pitch of the worm will be sufficiently long for all purposes. If shorter hobs are used, the worm should be made somewhat shorter than the hob. If a differential feed of the hob across the periphery of the worm gear is used, shorter hobs are satisfactory, and such hobs often are made as tapered hobs, thus combining the roughing and finishing operation. In such cases, the length of the full diameter of the hob plus the distance of differential feed should be slightly greater than the length of the mating worm.

**LOAD-CARRYING CAPACITY OF WORM DRIVES.**—As with all other types of gearing, there are three limiting factors, any one or more of which may limit the capacity of the drive. These factors are: heat of operation, beam strength of teeth, and surface endurance limits of materials. With worm drives, heat of operation and the surface endurance limits of the materials are the principal limiting factors.

On worm drives, rubbing velocity and pitch line velocity must be considered. For simplicity, consider rubbing velocity to be equal to peripheral velocity at the nominal pitch diameter of the worm, although it varies at different points and may be greater or less than that used, depending upon the nature of the contact. Whence,

$$\text{Rubbing velocity in ft. per min.} = 0.2618\,D \times \text{r.p.m.},$$

where $D$ = nominal pitch diameter of worm, in., r.p.m. = rev. per min. of worm.

$$V = (L \times \text{r.p.m.})/12,$$

where $V$ = pitch line velocity of worm gear, ft. per min., and $L$ = lead of worm, in.

The actual intensity of the dynamic load on worm gear teeth is an open question. In the absence of more definite information, the following relation may be used: $W$ = transmitted tooth load, lb., $W_d$ = dynamic tooth load, lb. Then

$$W_d = W\{(1200 + V)/1200\} \text{ for metal gears}$$
$$= W\{(200 + V)/(200 + 0.25\,V)\}, \text{ for phenolic laminated material.}$$

**BEAM STRENGTH OF WORM GEAR TEETH.**—The Lewis equation may be used to determine the beam strength of worm gear teeth. The safe static strength should be not less than the dynamic load. Let $W_s$ = safe static tooth load, lb., $p$ = axial pitch of worm, in., $F$ = face width of worm gear, in., $S$ = flexural endurance limit of material, lb. per sq. in., $y$ = tooth-form factor; then

$$W_s = SpFy.$$

For steady loads $W_s$ should be about 1.25 $W_d$; for shock loads, $W_s$ should be about 1.50 $W_d$.

With the changing forms and heights of the worm gear teeth across the face, an exact tooth-form factor is a difficult value to obtain. For all practical purposes, where the sum of the number of teeth in the worm and worm gear is not less than 40, a constant value of $y$ for each thread angle can be used, as follows:

| Thread angle = | $14\,^1/_2$ deg. | 20 deg. | 25 deg. | 30 deg. |
|---|---|---|---|---|
| $y$ = | 0.100 | 0.125 | 0.150 | 0.175 |

A hobbed worm gear never is generated completely. Its final finishing is obtained by its actual operation in position with its mating worm. For this reason, the materials used for worm gears should be such that they will conform to the worm without galling or excessive abrasion. Furthermore, the material of the worm should be smooth and hard enough to retain its form through such running-in or burnishing. Therefore, a steel worm to transmit power should have a Brinell hardness of not less than 250, and be as much harder as practicable.

Cast iron and semi-steel, manganese-, phosphor-, and some nickel-bronzes, and several of the aluminum alloys have proved to be satisfactory materials for worm gears, while the phenolic laminated materials prove very effective for the smaller gears and high-speed applications. Cast iron or semi-steel should not be used for rubbing velocities greater than about 1000 ft. per min. because of the danger of sparking. High-grade bronzes and aluminum alloys are essential for satisfactory results. A poorly alloyed material is practically worthless for a worm gear material. Values for $S$ are given in Table 23.

**Table 23.—Values of $S$ for Worm Gears**

| Material | $S$ | Material | $S$ |
|---|---|---|---|
| Gray iron | 12,000 | Manganese-bronze | 17,000 |
| Semi-steel | 18,000 | Phosphor-bronze | 24,000 |
| Phenolic laminated material | 6,000 | | |

Values for nickel-bronze and for the various alloys of aluminum suitable for worm gear materials should be obtained from the manufacturers of the materials. See also pp. 4–19 to 4–33. It should be appreciated that increased physical strength obtained at the expense of lesser ductility of the material is of questionable advantage in materials to be used for worm gears. Also, the physical properties and structure of these materials depend more on the care and skill with which they have been alloyed, poured and heat treated than upon the chemical analysis alone. Alloys of the same chemical analysis may have strength and wear values varying as much as two to one because of the difference in making and the difference in the resulting structure.

**LIMIT LOADS FOR WEAR.**—Let $W_w$ = limiting load for wear, lb.; $D_g$ = pitch diameter of worm gear, in.; $F$ = face width of worm gear, in.; $K$ = load-stress value, Table 24; then,

$$W_w = D_g FK.$$

The values of $W_w$ should not be less than the dynamic load.

**Table 24.—Values of $K$ for Worm Gears**

Running with steel worms of 250 Brinell or higher

| Material in Worm Gear | $K$ | Material in Worm Gear | $K$ |
|---|---|---|---|
| Cast iron or semi-steel | 50 | Phosphor-bronze | 100 |
| Manganese-bronze | 80 | Phenolic laminated material | 125 |

**EFFICIENCY OF WORM GEAR DRIVES** depends on several factors, *viz.*, the lead angle of the worm, the nature of the contact, intensity of the load, speed of rubbing, etc. When $E$ = efficiency; $\beta$ = lead angle of worm; $f$ = coefficient of friction,

$$E = \{\tan \beta(1 - f \tan \beta)\}/(\tan \beta + f).$$

This applies to the efficiency at the tooth mesh only. The overall efficiency of a complete drive also must include the power losses of the thrust and radial bearings of both members of the drive.

The coefficient of friction will vary with the rubbing speed and with the load, and follows closely the conditions existing on plain cylindrical bearings, except that the unit loads on worm gear drives are much greater than those on plain bearings. The values in Table 25 represent average conditions. Table 26 shows the pronounced influence of the lead angle of the worm on the efficiency of the drive. These values are calculated from the foregoing equation, using the values of $f$ given in Table 25.

**Table 25.—Values of the Coefficient of Friction $f$ in Worm Drives**

| Rubbing Speed, ft. per min. | $f$ | Rubbing Speed, ft. per min. | $f$ | Rubbing Speed, ft. per min. | $f$ |
|---|---|---|---|---|---|
| 5 | 0.146 | 150 | 0.045 | 1000 | 0.041 |
| 10 | .116 | 200 | .042 | 1250 | .046 |
| 20 | .090 | 300 | .040 | 1500 | .050 |
| 40 | .070 | 400 | .036 | 1750 | .054 |
| 80 | .054 | 500 | .033 | 2000 | .058 |
| 100 | .050 | 750 | .036 | 2500 | .065 |

A worm with a lead angle of 30 deg. is practically as efficient as one with 45 deg. Lead angles less than 10 deg. are very inefficient.

The heat of operation of worm gear drives would be estimated in the same manner as shown for spur gears, by determining values for the power loss in ft.-lb. per min., and the exposed area of the case in sq. ft.

**Table 26.—Relation of Lead Angle, Speed, and Efficiency of Worm Gear Drives**

| Rubbing Velocity, ft. per min. | Lead Angle of Worm 5° | 10° | 15° | 25° | 35° | 45° |
|---|---|---|---|---|---|---|
| | Efficiency | | | | | |
| 5 | 0.37 | 0.53 | 0.62 | 0.71 | 0.74 | 0.75 |
| 10 | .42 | .59 | .68 | .76 | .79 | .79 |
| 20 | .49 | .65 | .73 | .80 | .83 | .83 |
| 40 | .55 | .71 | .78 | .84 | .86 | .87 |
| 80 | .61 | .76 | .82 | .87 | .89 | .90 |
| 100 | .63 | .77 | .83 | .88 | .90 | .90 |
| 150 | .66 | .79 | .85 | .89 | .91 | .91 |
| 200 | .67 | .80 | .85 | .90 | .91 | .92 |
| 300 | .68 | .81 | .86 | .90 | .92 | .92 |
| 400 | 0.71 | 0.82 | 0.87 | 0.91 | 0.93 | 0.93 |
| 500 | .72 | .84 | .88 | .92 | .93 | .94 |
| 750 | .71 | .82 | .87 | .91 | .93 | .93 |
| 1000 | .68 | .80 | .86 | .90 | .92 | .92 |
| 1250 | .65 | .79 | .84 | .89 | .91 | .91 |
| 1500 | .63 | .77 | .83 | .88 | .90 | .90 |
| 1750 | .61 | .76 | .82 | .87 | .89 | .90 |
| 2000 | .60 | .74 | .81 | .86 | .88 | .89 |
| 2500 | .57 | .72 | .79 | .85 | .87 | .88 |

**HOUR-GLASS WORM DRIVES.**—Hour-glass or globoid worm drives are those where neither member of the pair has a uniform helical lead along its axis. The earliest type of such a drive is the Hindley worm. Many other drives of this type since have been designed. Each different drive of this type has so many individual features that they require individual analyses to determine the exact nature of the contact, etc. Also their production requires special manufacturing facilities. In all cases, however, the maximum theoretical contact that can be obtained is line contact.

## 16. MATERIALS OF GEARS

**MATERIAL FOR BRASS AND BRONZE GEARS.**—(Recommended Practice of the Amer. Gear Mfrs. Assoc., 1933.)—**Spur and Bevel Gears.**—Hard cast bronze, S.A.E. Specification No. 62 is recommended. The following are the limits of composition:

Copper, 86 to 89%; tin, 11 to 9%; zinc, 3 to 1%; lead, max., 0.20%; iron, max., 0.06%. Castings from this bronze should have the following minimum physical properties: Ultimate strength, 30,000 lb. per sq. in.; yield point, 15,000 lb. per sq. in.; elongation in 2 in., 14%.

**Bronze Worm Gears.**—S.A.E. phosphor bronze, No. 65 and S.A.E. leaded gun metal, No. 63, may be used. The limits of composition and physical properties are: Copper, 88 to 90%; tin, 12 to 10%; phosphorus, 0.1 to 0.3%; lead, zinc and impurities, max., 0.5%. Ultimate strength, 35,000 lb. per sq. in.; yield point, 20,000 lb. per sq. in.; elongation in 2 in., 10%.

**Table 27.—Composition of Steel for Gears**

(Recommended Practice of Am. Gear Mfrs. Assoc., 1933)

| Treatment or Number | C | Mn | P max. | | S max. | Ni | Cr | Vn | | Mo |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Forged or Rolled Carbon Steel | | | | | | | |
| Case Hardened..... | 0.15–0.25 | 0.40–0.70 | 0.045 | | 0.055 | .......... | .......... | ........ | | ........ |
| Untreated.. | 0.25–0.50 | 0.50–0.80 | 0.045 | | 0.055 | .......... | .......... | ........ | | ........ |
| Hardened or untreated. | 0.40–0.50 | 0.40–0.70 | 0.045 | | 0.055 | .......... | .......... | ........ | | ........ |
| | | | Forged and Rolled Alloy Steel | | | | | | | |
| 2315 | 0.10–0.20 | 0.30–0.60 | 0.04 | | 0.05 | 3.25–3.75 | .......... | * | † | ........ |
| 2350 | 0.45–0.55 | 0.50–0.80 | 0.04 | | 0.05 | 3.25–3.75 | .......... | .... | .... | ........ |
| 2512 | 0.17 max. | 0.30–0.60 | 0.04 | | 0.05 | 4.75–5.25 | .......... | .... | .... | ........ |
| 3115 | 0.10–0.20 | 0.30–0.60 | 0.04 | | 0.05 | 1.00–1.50 | 0.45–0.75 | .... | .... | ........ |
| 3215 | 0.10–0.20 | 0.30–0.60 | 0.04 | | 0.045 | 1.50–2.00 | 0.90–1.25 | .... | .... | ........ |
| 3250 | 0.45–0.55 | 0.30–0.60 | 0.04 | | 0.045 | 1.50–2.00 | 0.90–1.25 | .... | .... | ........ |
| 3312 | 0.17 max. | 0.30–0.60 | 0.04 | | 0.045 | 3.25–3.75 | 1.25–1.75 | .... | .... | ........ |
| 3340 | 0.35–0.45 | 0.30–0.60 | 0.04 | | 0.045 | 3.25–3.75 | 1.25–1.75 | .... | .... | ........ |
| 6120 | 0.15–0.25 | 0.30–0.60 | 0.04 | | 0.045 | .......... | 0.80–1.10 | 0.15 | 0.18 | ........ |
| 6150 | 0.45–0.55 | 0.50–0.80 | 0.04 | | 0.045 | .......... | 0.80–1.10 | 0.15 | 0.18 | ........ |
| 4615 | 0.10–0.20 | 0.30–0.60 | 0.04 | | 0.050 | 1.50–2.00 | .......... | .... | .... | 0.20–0.30 |
| | | | Steel Castings | | | | | | | |
| | | | A‡ | B§ | | | | | | |
| Case Hardened..... | 0.15–0.25 | 0.40–0.60 | 0.06 | 0.05 | 0.06 | .......... | .......... | ........ | | ........ |
| Untreated or Hardened. | 0.30–0.40 | 0.40–0.60 | 0.06 | 0.05 | 0.06 | .......... | .......... | ........ | | ........ |

* Minimum. † Desired. ‡ A = Acid. § B = Basic.

The following are the limits of composition of the gun metal: Copper, 86 to 89%; tin, 11 to 9%; lead, 2.5 to 1%; phosphorus, max., 0.25%; zinc and impurities, max., 0.50%. The expected physical characteristics of good castings are: Ultimate strength, 30,000 lb. per sq. in.; yield point, 12,000 lb. per sq. in.; elongation in 2 in., 10%. These alloys, especially No. 65, are adapted to chilling for hardness and refinement of grain. No. 65 is preferred for worms of great hardness and fine accuracy. No. 63 is preferred for unhardened worms.

**Bronze Bushings for Gears.**—S.A.E. No. 64 is recommended, of the following analysis: Copper, 78.5 to 81.5%; tin, 9 to 11%; lead, 9 to 11%; phosphorus, 0.05 to 0.25%; zinc (max.), 0.75%; other impurities (max.), 0.25%. Minimum physical characteristics of castings are: Ultimate strength, 25,000 lb. per sq. in.; yield point, 12,000 lb. per sq. in.; elongation in 2 in., 8%.

**Brass Flanges for Composition Pinions.**—A.S.T.M. B 30–32T, and S.A.E. No. 40 are recommended. Composition is as follows: Copper, 83 to 86%; tin, 4.5 to 5.5%; lead, 4.5 to 5.5%; zinc, 4.5 to 5.5%; iron (max.), 0.35%; antimony (max.), 0.25%; aluminum, none. Minimum physical characteristics of the castings are: Ultimate strength, 27,000 lb. per sq. in.; yield point, 12,000 lb. per sq. in.; elongation in 2 in., 16%.

**Table 28.—Limits and Tolerances for Holes in Gears**

(Recommended Practice of Am. Gear Mfrs. Assoc., 1933)

| Diam., in. | Precision Gears, Aircraft, Printing Machines, etc. | | | Automobiles, Machine Tools, etc. | | | Standard Jobbing Gears | | |
|---|---|---|---|---|---|---|---|---|---|
| | + Limit, "Not Go" | − Limit, "Go" | Tolerance | + Limit, "Not Go" | − Limit, "Go" | Tolerance | + Limit, "Not Go" | − Limit, "Go" | Tolerance |
| 0–1/2 | 0.000 | 0.00025 | 0.00025 | 0.00025 | 0.00025 | 0.0005 | 0.0005 | 0.0005 | 0.001 |
| 1/2–1 | 0.000 | 0.0005 | 0.0005 | 0.0005 | 0.0005 | 0.001 | 0.00075 | 0.00075 | 0.0015 |
| 1–2 | 0.000 | 0.00075 | 0.00075 | 0.00075 | 0.00075 | 0.0015 | 0.001 | 0.001 | 0.002 |
| 2–3 | 0.00025 | 0.00075 | 0.001 | 0.001 | 0.001 | 0.002 | 0.00125 | 0.00125 | 0.0025 |
| 3–4 | 0.0005 | 0.00075 | 0.00125 | 0.00125 | 0.001 | 0.00225 | 0.0015 | 0.0015 | 0.003 |
| 4–5 | 0.0005 | 0.001 | 0.0015 | 0.0013 | 0.0012 | 0.0025 | 0.00175 | 0.00175 | 0.0035 |
| 5–6 | 0.0005 | 0.001 | 0.0015 | 0.00175 | 0.00125 | 0.0003 | 0.002 | 0.002 | 0.004 |
| 8–9 | 0.00075 | 0.001 | 0.00175 | 0.002 | 0.002 | 0.004 | 0.003 | 0.003 | 0.006 |
| 11–12 | 0.001 | 0.001 | 0.002 | 0.0025 | 0.0025 | 0.005 | 0.004 | 0.004 | 0.008 |

# Section 15

# CONTROL MECHANISMS

## GOVERNING, REGULATION AND CONTROL

By W. Trinks

## BRAKES

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# CONTROL MECHANISMS

## GOVERNING, REGULATION, AND CONTROL

By W. Trinks

**References.**—Trinks, Governors and the Governing of Prime Movers, Van Nostrand, 1919; Stein, Regelung und Ausgleich in Dampfanlagen, Julius Springer, 1926; Wünsch, Regler für Druck und Menge, Oldenburg, 1930; Tolle, Die Regelung der Kraftmaschinen, Julius Springer, 1921; Behar, Fundamentals of Instrumentation, Instruments Publishing Company, 1932; Smith, Automatic Regulators, Their Theory and Application, *Trans.* A.S.M.E., PRO58-4, p. 291–303, 1936; Mitereff, Principles underlying Rational Solution of Automatic-Control Problems, *Trans.* A.S.M.E., FSP57-9, p. 159–163, 1935.

This section deals with fundamentals only; a complete treatise would fill a good-sized book.

### 1. PRINCIPLES

The words *governing*, *regulation*, and *control* have the same general meaning. Each means adjusting the supply (of power, heat, fluid, etc.) to the demand, or *vice versa*, thereby keeping constant one factor (speed, temperature, pressure, etc.) which is called the controlled quantity. Although the word *controlling* should be applicable to keeping any quantity constant, usage has made it impossible to adopt this term as a standard. We still say: Turbine speed is governed, water level in a boiler is regulated, temperature in a furnace is controlled.

All controlled equipment has self-regulating properties which limit the variation of that quantity which should remain constant, but the variations within the self-regulating range are, as a rule, too great to be allowed in practice. For example, an impulse turbine may reach nearly twice its normal speed if all of the load is removed. Any automatic-control installation consists of: 1. The controlled system; 2. The automatic-control apparatus proper. Through the controlled system there is a flow (of fluid or energy). Normally, *a*, inflow or supply and, *b*, outflow or demand are equal, and a factor or quantity, *c*, which is a function of the flowing medium, is kept practically constant (controlled, governed, regulated). Between inflow and outflow there is, in general, *d*, a storage capacity (tank, flywheel) which keeps down the suddenness of variations of controlled quantity *c*, if either inflow *a* or outflow *b* is changed. The automatic control apparatus consists of a measuring instrument, *e*, (also called primary element), which measures or at least detects (responds to) variations of the controlled factor *c*; an operating element, *f*, which adjusts in some cases the inflow and in others the outflow, if a disturbance occurs in either; and finally an operative connection, *g*, between the measuring (detecting) instrument *e* and the regulating or operating element *f*. In some cases, two of the elements are combined into one.

**GENERAL PRINCIPLE.**—A force is derived from the controlled quantity or variable, which latter may be speed, temperature, pressure, rate of flow, voltage, etc., and is balanced by an external, known force, *i.e.*, weight, spring, fluid pressure, etc., called the balancing force. Any change in the controlled quantity unbalances the measuring system; the unbalanced force, if sufficient to overcome friction, acts to re-balance the supply (inflow) and demand (outflow), and to bring the controlled quantity back to its normal value. The apparent defect that the supposedly constant quantity must change before the control apparatus can act, is almost overcome by making the necessary change exceedingly small, and by the advantage that governing is accomplished no matter whether the disturbance arises in the inflow (supply) or outflow (demand). Mathematically, the laws of governing, regulation, and control are locked up in differential equations; see the references cited.

### Definitions

**Direct-acting Governor, Self-operating Controller.**—If after a disturbance the unbalanced force moves, directly, the mechanism which adjusts the flow of power, fluid, heat, etc., the governor (on an engine or turbine) is called a direct-acting governor, and the controller (for pressure, temperature, etc.) is called a self-operating controller, or self-operating regulator.

**Relay Control.**—If the unbalanced force releases and controls a larger auxiliary force, which, in turn, adjusts the mechanism to restore equilibrium between inflow (supply) and outflow (demand), the controller is said to be of the *relay* type, or is called a *pilot controller.* The amplified force may be derived from a solenoid, an electric motor, from fluid pressure, or from motion of a prime mover.

**The Application.**—Control equipment now is manufactured in regular production, and is applied to many different controlled systems. Any one of these systems, consisting of inflow (supply), storage (capacity), and outflow (demand), is generally called the *application* by the manufacturers of automatic control apparatus.

**Modes of Control.**—After a disturbance, that is to say after a deviation of the controlled quantity from normal, the quantity may settle down to new equilibrium and remain quiescent until the next disturbance occurs. This is known as stable, steady or non-oscillatory control. For certain applications, as turbine speed governing, stability of regulation without oscillations is positively required. If the controlled variable oscillates continually about the normal value, the control is of the hunting type. For some industrial applications, as furnace temperature control, hunting within narrow limits is considered permissible. Control may be of the *on-and-off* or intermittent type, or it may be *floating,* in which case the regulating process (adaptation of supply to demand) is continuous. The on-and-off type necessarily hunts, but the floating type also may hunt.

**Regulation** has several meanings. In speed control, according to the A.S.M.E. Power Test Code, it is the ratio [difference between the maximum (ma) and minimum (mi) values of the controlled quantity] ÷ [arithmetical mean of these extreme values], or

$$(\text{ma} - \text{mi})/\{{}^1/_2\,(\text{ma} + \text{mi})\}$$

for stable or steady regulation and for the full scale of the measuring element. This ratio also is known as *static fluctuation, speed variation* or *speed droop.* In temperature control, as well as in close control of pressures near the atmosphere, division by the absolute temperature or pressure is inconvenient. In such cases the absolute change of the controlled quantity which is necessary to move the final or operating element from one extreme to the other sometimes is called *regulation*; more often it is known as the *throttling range*, or better, the *operating range*. These terms apply to slow changes of supply or demand. In some cases, *regulation* means that difference between the maximum and minimum values of the controlled quantity, either absolute, or expressed in terms of the average value, which occurs when the controller functions after a severe disturbance. If this difference be great, the *regulation is poor*. In this sense, regulation is synonymous with *dynamic fluctuation*, which latter always exceeds the static fluctuation (throttling range, regulating range). In Fig. 1, which is a chart drawn by a tachograph and which shows the change of speed of a steam engine after a sudden change to full load from no load, the static fluctuation (speed droop) is proportional to the vertical distance from 1 to 3, while the dynamic fluctuation is proportional to the vertical distance from 1 to 2.

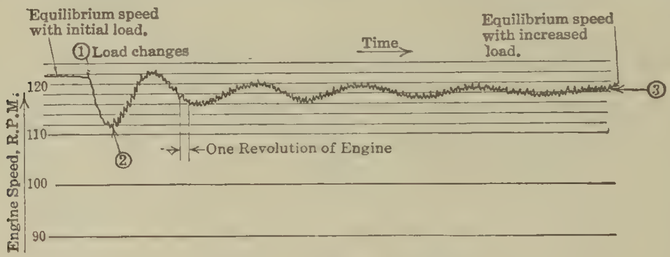

FIG. 1. Tachograph Record

**Sensitivity** and **Sensitiveness** often are used indiscriminately and, unfortunately, express several different concepts. Sensitiveness, as used in speed governing, is inversely proportional to the minimum change in speed which will cause the governor to move, and depends on the friction and lost motion in the governor or linkage. The term also, in accordance with older practice, is used to mean the travel of the governor for 1% change in speed. In the automatic control of quantities other than speed, the terms *ultimate sensitivity, dead range* (dead space, dead zone), or *sticking* (stiction, hysteresis) are used and are synonymous with sensitivity. The ultimate sensitivity of an instrument is inversely proportional to, and is measured by, the amount which the measured quantity may vary without moving the measuring element; this is indicated in Fig. 2, which illustrates the different meanings of the terms sensitivity or sensitiveness. A low ultimate sensitivity usually is caused by friction, but part of this effect is, in certain cases, ascribable to lost motion in the apparatus.

Sensitivity of a controller generally refers to the extent of movement of the pointer lever, or other position-changing element of the measuring instrument, required to cause the valve or other operating element to travel a definite but more or less arbitrary extent. Sensitivity in this sense is said to be high when the movement of the sensitive or primary element required to cause a given valve movement is small; that is, when the characteristic curves in Fig. 2 have steep slope. The *unit sensitivity* of the automatic controller sometimes refers to the value of the change in the measured quantity which is required to cause full valve travel. In that sense it is synonymous with the throttling range.

**Set Point** (Setting Point, Normality).—The value at which the controlled quantity is to be maintained often is indicated by the position of a pointer on the controlling instrument. This is called the set point, and represents the value at which the controller attempts to find equilibrium after each disturbance or load change. In simple devices, as reducing valves or direct-control governors, set pointers as a rule are not provided, but the adjusting screw or equivalent is the means by which the set point is changed.

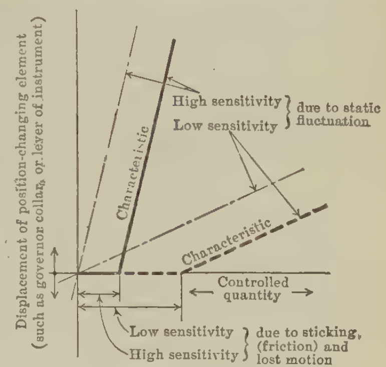

Fig. 2. Variations in Sensitivity

**Control Point** is the particular value of the controlled quantity at which the controller comes to equilibrium after a disturbance. In simple controllers, having a static fluctuation or droop, the control point will, at times, differ from the set point, depending on the load. In more complex types of instruments, the controller maintains the control point at the set point regardless of the load, except immediately after a disturbance.

**Time Lags**, *i.e.*, lapse of time between cause and effect, occur in both the controller and the controlled system (the application).

Lag in the controller is due to mechanical or thermal inertia, to capacity (storage), to lost motion, friction, damping, or to limitations of the supply of actuating energy. It exists in both the measuring and the operating element; a further lag exists in the connection between these elements, but its effects usually are negligible.

In the controlled system (the application), different time lags may exist separately or jointly. There may be a specific elapsed time, due to difference in location of the measuring and the operating elements, before the measured quantity is affected by the change in the supply or demand; this delay may be called the *distance lag*. There is also a delay due to the capacity of the controlled system; it is referred to as the *capacity lag*. The *impulse transfer lag* in the measuring instrument is the time required for transmitting a change in the immediate vicinity of the sensitive element to that element. The sum of these three lags is the *application lag*. In a few cases, the lag is increased still more by the difficulty that the measuring element (for example, a thermocouple) cannot be located at the place where the controlled quantity is to be kept constant, but ahead of it in the flow system. A *beyond-control* lag thereby is introduced, which likewise is part of the application lag.

Any lag or delay in the controlled system which reduces the suddenness of the change in the controlled quantity helps regulation, while any lag which results in lapse of time between change in controlled quantity and adjusting action of the controller makes accurate regulation more difficult.

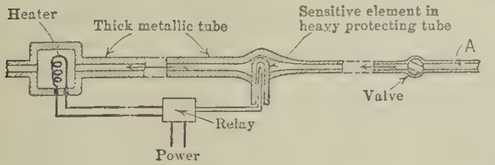

Fig. 3.

Fig. 3 illustrates the various lags in the controlled system. If the valve at the right is opened or closed, the temperature of the medium flowing past the heater is changed. 1. Time elapses while the fluid with changed temperature flows from the heater to the sensitive element; 2. Time elapses while the fluid exchanges heat with the surrounding metallic tube; 3. Time also elapses before the changed temperature of the fluid near the sensitive element can penetrate the protecting tube; 4. If the controlled quantity is to be kept constant at some distance to the right, *e.g.*, at point *A*, Fig. 3, of the measuring instrument, time elapses while the controlled quantity reaches its normal value at the location of the measuring instrument and at the desired location. In speed control, energy which has flowed beyond the throttle persists for a while in changing the speed of the engine or turbine. The second time lag mentioned is beneficial, the other three are harmful to regulation.

**Speed of Response** (promptness) as contrasted with sluggishness, is a measure of the rate at which the measuring instrument follows a sudden variation of the measured (and controlled) quantity. In addition to sticking, lost motion, inertia (mass action), damping, and impulse transfer lag make a primary element sluggish. Speed of response can be measured by the traversing time. The latter is the time during which either a self-operating controller, or the primary element of a relay control, travels from one end of the control range to the other, when actuated and accelerated only by the external balancing force. Example: The centrifugal weights of a non-rotating speed governor are pulled apart to the outer stop, and released. The time required to reach the inner stop is the traversing time. For the average spring-loaded speed governor, the traversing time ranges from 0.04 to 0.06 second; for weight-loaded governors, from 0.10 to 0.20 second. In pressure control, if the bell of the apparatus shown in Fig. 4 is raised to the top of its travel, is released, and falls (due to its own weight) to the bottom of the regulating travel, the time of falling is the traversing time. In many cases of automatic control, changes are so slow that speed of response is no object. This also applies to speed control.

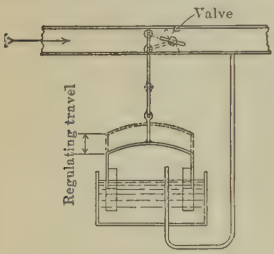

FIG. 4.

**Speed of Controlling** is measured by the time required for the equilibrium-restoring element (control unit) to move from one extreme position to the other, through the throttling range, under the action of the wide-open primary element. In self-operating controls, it is inversely proportional to the traversing time. In relay controls it is inversely proportional to the traversing time of the relay alone.

**Strength** is that force, exerted at the control point, which the balancing force of the instrument exerts when the external balancing force alone, unhindered by friction, acts. Knowledge of the strength is important for self-operating control equipment, such as direct-control governors and reducing valves, because the ratio (frictional resisting force of control mechanism) ÷ (strength) determines the relative or percentage variation of the regulated quantity which must be incurred for overcoming the frictional resistance of the control mechanism. In relay control, strength is generally of less importance.

**Regulating Force** is that fractional part of the strength which is available for overcoming resistance in the control mechanism or in the relay mechanism. In centrifugal speed governors, the regulating force for $P$ percent change of speed equals $2P$ percent of the strength, if friction in the governor is eliminated. In pressure governors, for $P$ percent change of regulated quantity, the regulating force is $P$ percent of the strength; in temperature control with radiation pyrometers, it is $4P$ percent. Since only small changes of the regulated quantity are allowed in practice, the regulating force never can be more than a small fraction of the strength.

**Work Capacity** of the primary element is of importance for self-operating controls only. It is the product of (average strength × distance through which it moves for the useful range of the control instrument). Only a very small fraction of the work capacity is available for doing work in overcoming resistance offered by the equilibrium-restoring mechanism. The work capacity of a relay is the product of (average static force × maximum travel of relay).

**Temporary Stability.**—Control with an absolutely constant value of the regulated quantity (except with a short deviation following a disturbance) is possible, if a change of controlled quantity, as a function of the load, is permitted during a disturbance and is, later on, automatically wiped out either by leaky gag-pots floating between springs or by a *double response*. (See below.)

**Asymptotic Regulation** in controllers with temporary stability is that type in which the rate of change of controlled quantity decreases as the control point is approached.

**Isoposic*** **Regulation** is that type in which the control mechanism comes to rest with the same value of the quantity to be controlled for all positions of the control mechanism (if the load changes very slowly); or in which the long-time static fluctuation, or droop, is zero. The term describes performance only, which may be obtained by different mechanisms. Regulators for pressure, rate of flow, or temperature, having this characteristic, sometimes are called *non-corresponding*, while speed governors of this type are called *isochronous governors*. The strictly isochronous governor, unaided by tangential inertia or by temporary stability, cannot be used, because by its dynamic interaction with the prime mover, it produces excessive and incessant fluctuations of speed.

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* From Greek "Iso," the same, "Posos," quantity.

## 2. RELATION BETWEEN CONTROL INSTRUMENT AND APPLICATION

In the use of control apparatus, the latter may be built especially to be an integral part of the application, as for instance a shaft governor on a steam engine, of which it is a part. In the great majority of cases, a commercial control apparatus is purchased separately and applied to the controlled system (the application). The commercial controller, in this very common case, must have enough adaptability and adjustments to fit many different systems. The purchased control apparatus has no influence on factors affecting controller performance which are inherent in the application, as: 1. Lag due to storage of fluid, or enegry, beyond the control of the regulating element; for example, steam stored in multi-stage engine when load changes. 2. Impulse transfer lag to the measuring instrument, or primary element; for example, the thickness and material of the protecting tube of a thermocouple frequently are prescribed by the owner of the controlled system. 3. Suddenness and amount of disturbance; for example, power demand suddenly removed by kicking open of a circuit breaker.

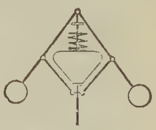


Fig. 5. Simple Speed Governor

If great time-lags (p. 15–05) are coupled with very sudden or very great disturbances, the very best control apparatus cannot produce close regulation. Lack of understanding of this situation has caused much quarreling between purchaser and seller of automatic control equipment, and keeps prices of control equipment high, because the cost of difficult installations, which require many trips of service men in the attempt to make them work satisfactorily, must in the long run be distributed over all, and even the simplest, control equipment and installations. Sufficiently early contemplation of control equipment in the design of systems would overcome many of these troubles.

## 3. TYPES OF CONTROL EQUIPMENT

The multiplicity of types is so great that only the most important ones can be mentioned here.

**SELF-OPERATING CONTROL APPARATUS.**—This type is exemplified by engine governors, by float control of water level in tanks, water closets, or sometimes in steam boilers, and by reducing valves. Of necessity, regulation is continuous or floating. For stability of regulation, without hunting, and also for parallel operation, a static fluctuation or regulation of low sensitivity is required, at least during a disturbance. Its use is limited to those cases in which a considerable force can be derived from the controlled quantity, and in which the frictional resistance of the equilibrium-restoring operating mechanism either is small or can be made small by vibration. Small mass of parts in measuring instruments is desirable, to prevent hunting. Self-operating controllers are subject to upsetting influences from unbalanced valves or valve gears; in particular, butterfly valves. Examples of self-operating regulators are shown in Figs. 5 to 10. Fig. 5 shows diagrammatically a simple form of speed governor. Fig. 6 shows a highly-developed spring-loaded speed governor of the Jahns type (Massey Machine Co.); its properties are given in Table 1. Fig. 7 shows a simple form of spring-

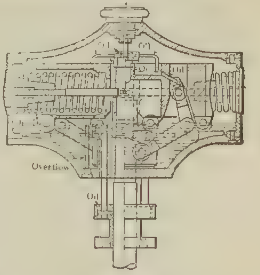


Fig. 6. Massey Governor

**Table 1.—Data of Spring-loaded Governors** (Fig. 6)
(Massey Machine Co., Watertown, N. Y.)

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| MID-STROKE OF GOVERNOR | | | | | | | |
| Rev. per minute (standard) | 490 | 400 | 325 | 275 | 225 | 190 | 160 |
| Highest possible r.p.m. | 560 | 460 | 380 | 315 | 255 | 210 | 175 |
| Lowest possible r.p.m. | 260 | 220 | 180 | 145 | 120 | 100 | 70 |
| Stroke, in. | 63/64 | 1 3/8 | 1 3/4 | 2 3/8 | 3 1/8 | 4 | 5 |
| AT STANDARD SPEED | | | | | | | |
| Strength, lb. | 132 | 240 | 440 | 990 | 1800 | 2800 | 5000 |
| Total work capacity, in.-lb. | 130 | 330 | 770 | 2350 | 5625 | 11,200 | 25,000 |
| Weight, boxed, lb. | 75 | 143 | 240 | 548 | 944 | 1,530 | 2,400 |
| Overall diameter, in. | 10 1/4 | 13 11/16 | 17 3/8 | 23 | 28 3/4 | 35 | 44 |

loaded governor (Pickering) for throttling control; the sizes in which this governor is made are shown in Table 2. In Fig. 8 appears a spring-loaded pressure regulator of the reducing valve type (Foster), data concerning which are given in Table 3. Fig. 9 shows a feed-water regulator for controlling the level of the water in a boiler; the feed valve is opened or closed by contraction or expansion of the inclined metallic tube, caused by the change

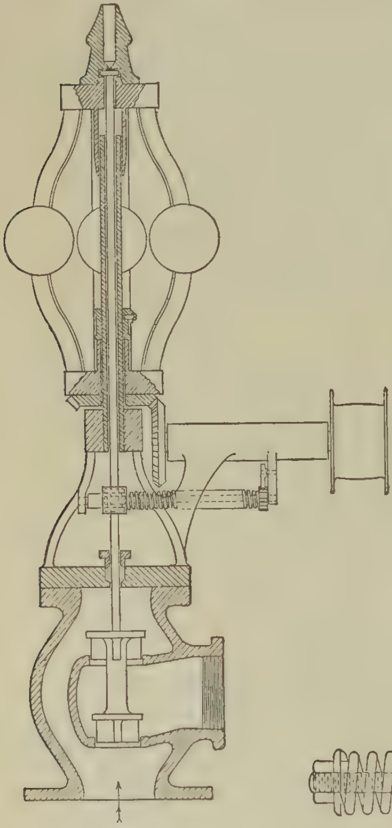


Fig. 7. Pickering Governor

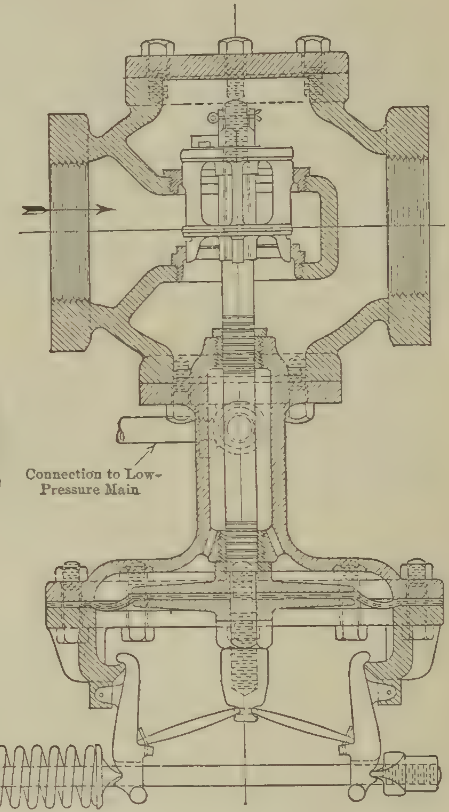


Fig. 8. Pressure-reducing Valve

**Table 2.—Sizes and Properties of Pickering Governor** (Fig. 7)

| Size of Governor (Size of Inlet Pipe, in.) | Extreme Height, in. | Greatest Expansion of Balls, in. | Speed of Governor, r.p.m. | Diam. of Pulley, in. | Width of Belt, in. | Size of Governor (Size of Inlet Pipe, in.) | Extreme Height, in. | Greatest Expansion of Balls, in. | Speed of Governor, r.p.m. | Diam. of Pulley, in. | Width of Belt, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | 14 | 5 | 500 | 2 | 3/4 | 4 | 42 | 12 | 320 | 5 | 2 1/2 |
| 3/4 | 14 | 5 | 500 | 2 | 3/4 | 4 1/2 | 42 | 12 | 320 | 5 | 2 1/2 |
| 1 | 18 | 6 1/2 | 350 | 2 1/2 | 1 1/4 | 5 | 48 | 15 | 275 | 5 | 2 1/2 |
| 1 1/4 | 19 | 6 1/2 | 350 | 2 1/2 | 1 1/4 | 6 | 52 | 16 1/2 | 275 | 6 | 3 |
| 1 1/2 | 25 | 8 | 340 | 3 1/2 | 1 1/2 | 7 | 54 | 16 1/2 | 275 | 7 | 3 |
| 2 | 26 | 8 | 340 | 3 1/2 | 1 1/2 | 8 | 56 | 18 | 260 | 7 | 3 |
| 2 1/2 | 31 1/2 | 9 | 275 | 4 | 2 | 9 | 59 | 20 | 260 | 8 | 3 |
| 3 | 35 | 11 | 340 | 4 | 2 | 10 | 62 | 20 | 225 | 8 | 3 |
| 3 1/2 | 36 | 11 | 340 | 4 | 2 | ....... | ....... | ....... | ....... | ....... | ....... |

Table 3.—Sizes and Properties of Pressure-reducing Valve (Fig. 8)

| Size, in. | Overall Height, in. | Face to Face of Outlets or Flanges, in. | Initial Pressure lb. per sq. in. Gage | Delivery Pressure, lb. per sq. in. Gage, Adjustable From | To | Maximum Variation in Delivery Pressure, percent | Capacity, lb. Saturated Steam per hr. at 250 lb. per sq. in. Initial Pressure |
|---|---|---|---|---|---|---|---|
| 3/4 | 14 3/8 | 4 1/4 | 250 max. | 0 | 15 max. | ± 5% | 1,420 |
| 1 | 14 3/8 | 4 5/8 | " | " | " | " | 1,760 |
| 1 1/4 | 15 1/8 | 5 1/8 | " | " | " | " | 2,850 |
| 1 1/2 | 16 3/8 | 5 3/8 | " | " | " | " | 4,700 |
| 2 | 17 5/8 | 6 1/2 | " | " | " | " | 7,050 |
| 2 1/2 | 20 5/8 | 9 | " | " | " | " | 12,850 |
| 3 | 21 1/4 | 10 | " | " | " | " | 21,900 |
| 3 1/2 | 22 | 11 | " | " | " | " | 33,300 |
| 4 | 24 | 12 | " | " | " | " | 42,800 |
| 4 1/2 | 24 1/4 | 14 | " | " | " | " | ...... |
| 5 | 24 3/4 | 15 | " | " | " | " | 71,200 |
| 6 | 26 3/4 | 17 | " | " | " | " | 92,000 |
| 7 | 32 3/4 | 18 1/4 | " | " | " | " | 133,000 |
| 8 | 32 3/4 | 20 3/4 | " | " | " | " | 181,000 |
| 10 | 34 1/2 | 23 1/2 | " | " | " | " | 298,000 |
| 12 | 37 1/2 | 27 1/4 | " | " | " | " | 451,000 |

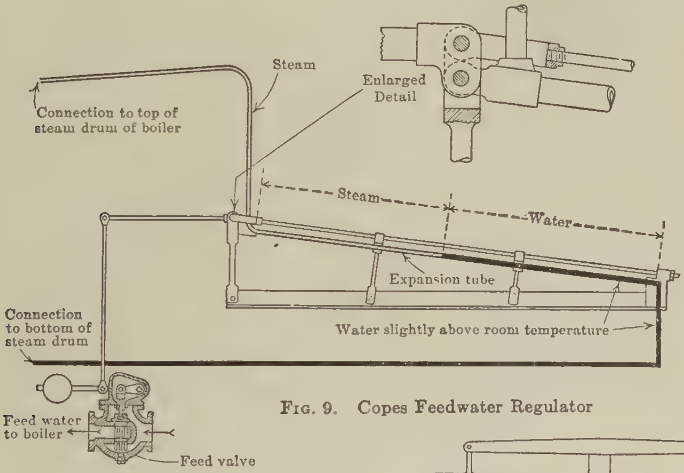

Steam
Enlarged Detail
Connection to top of steam drum of boiler
Steam
Water
Expansion tube
Water slightly above room temperature
Connection to bottom of steam drum
Feed water to boiler
Feed valve

Fig. 9. Copes Feedwater Regulator

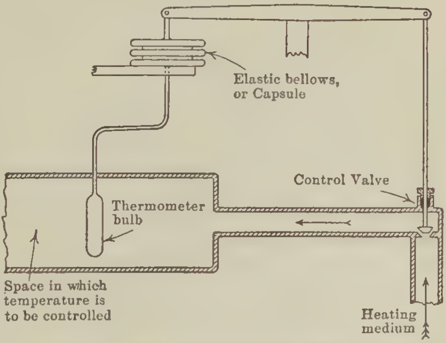

Elastic bellows, or Capsule
Control Valve
Thermometer bulb
Space in which temperature is to be controlled
Heating medium

Fig. 10. Direct-acting Temperature Control

in relative lengths of the parts exposed to steam or to cooler water, respectively, as the water level rises or falls. Fig. 10 shows a self-operating temperature control device.

**RELAY TYPES.**—Most commercial automatic control equipment is of the relay type. The advantages are: Measuring instrument is small, and it is not subject to upsetting forces due to variable resistance of equilibrium-restoring mechanism. It has the disadvantage of complication, and of cost in small controlled systems. For example, the control for a small, simple heat-treating furnace sometimes costs more than the furnace itself. Relay controls are of the following types:

**1. Proportional (corresponding) Control** with coupling-back (stabilizer, rigid return, precision action, vernier valve action, valve positioner).—For examples, see Figs. 12 to 14. The coupling-back causes the movement of the relay to be proportional to the movement of the sensitive element. The action of this type of control is identical with that of the self-operating type, but with amplification of the regulating force of the primary element. In consequence it has a static fluctuation, or droop. In practice, two or three amplifications in series occasionally are employed. A compact double relay in which a floating element is at once the primary relay piston and the secondary relay valve is illustrated in Fig. 11. Coupling-back is accomplished by a rigid return of the pilot valve as in the speed governor of Fig. 12; or by a stabilizing spring, as in the speed governor of Fig. 13, or in the pressure control of Fig. 14; or by still other means. Amplification may include conversion from one type of energy into another type, for instance conversion from electrical energy into fluid-pressure energy.

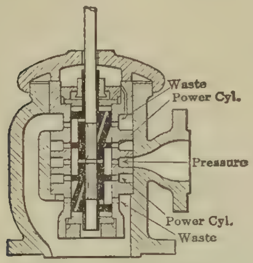


Fig. 11. Double Relay

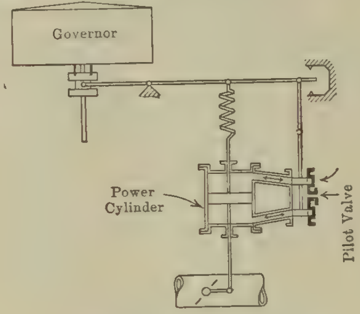


Fig. 13. Relay Speed Governor with Stabilizing Spring

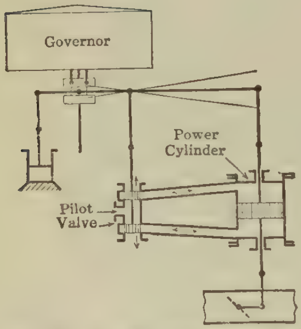


Fig. 12. Relay Speed Governor with Rigid Return of Pilot Valve

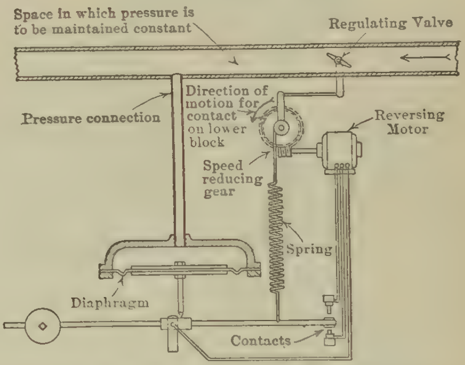

Space in which pressure is to be maintained constant
Regulating Valve
Pressure connection
Direction of motion for contact on lower block
Reversing Motor
Speed reducing gear
Spring
Diaphragm
Contacts

Fig. 14. Relay Pressure Control with Stabilizing Spring

**2. Continuous Control** without coupling-back. In this type, the relay equipment, once set in motion, is stopped only by a reversal of motion of the primary sensitive element. This is a simple type of control which is used where the load changes are relatively gradual and where the application lag is small. In general, it allows the control point to shift about the set point, and thus causes hunting unless the controlled system has inherent stability, as for instance in pressure control. It seldom is used in speed governing, but is common in pressure control. See Fig. 15.

**3. Isoposic Control with Relay.**—Proportional (corresponding) control with slowly yielding coupling-back, or slowly yielding return. This type is identical in action with No. 1 during a disturbance, which means that it has a static fluctuation during a disturbance. However, the interposition of a slowly-yielding return element, such as a spring-pressed oil pot, removes the static fluctuation and returns the controlled quantity to its original value. The mechanism which returns the controlled variable to normal value has many names; it is referred to as a follow-up, a reset, a stabilizer, a double response, a compensator of the second order. Control of this type is extremely useful in cases where there is a large application lag, because it eliminates hunting, while permitting practically constant value of the controlled quantity regardless of the load, except immediately after a disturbance. A speed governor of this type is shown in Fig. 16, and two pressure con-

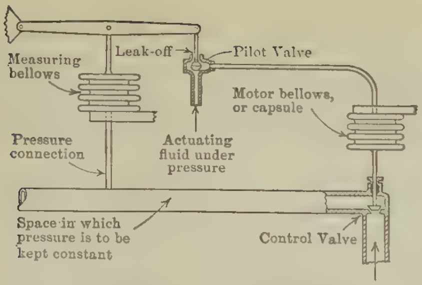

Measuring bellows
Leak-off
Pilot Valve
Motor bellows, or capsule
Pressure connection
Actuating fluid under pressure
Space in which pressure is to be kept constant
Control Valve

Fig. 15. Relay Pressure Control without Return

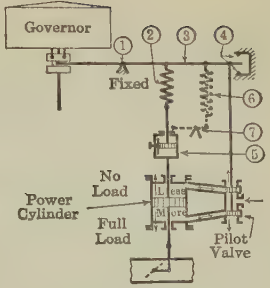


Fig. 16. Relay Speed Governor with Yielding Return

trols in Figs. 17 and 18; in the former, the slowly yielding return element is an oil gag-pot; in Fig. 18, it is a double bellows with a capillary resistance-tube. In the latter type, adjustment of damping action is obtained by variation of effective length of lever arm on which double bellows act.

**4. Anticipator Control.**—The name is misleading, but was originated to indicate that when the controlled quantity was returning to normal, the control action would anticipate the complete return to normal. This type of control is the same as No. 3, with the exception that, by any one of various means, a large correcting motion of the valve or operating element is caused to occur in the early stages of a controlling action after a disturbance, followed by a gradual decrease of the valve motion as the instrument approaches equilibrium. A variety of specific designs of instruments fall into this classification. This type of control is useful in cases with great time lag, for enforcing stability of regulation and preventing hunting. The same effect also can be obtained by intermittent control (see p. 15–12).

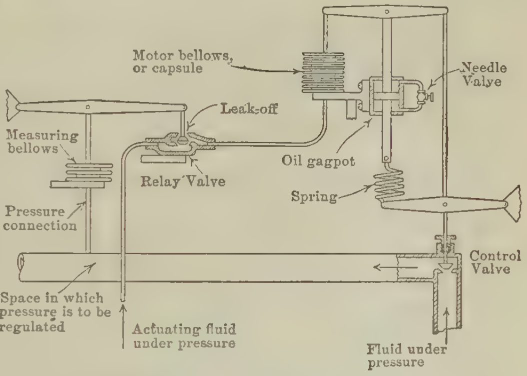


Fig. 17. Relay Pressure Control with Yielding Return, Spring-loaded Oil Gag-pot

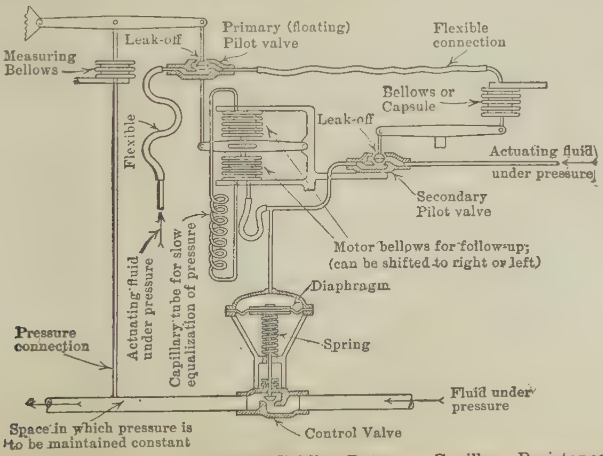

Leak-off
Primary (floating) Pilot valve
Flexible connection
Measuring Bellows
Bellows or Capsule
Leak-off
Flexible
Actuating fluid under pressure
Secondary Pilot valve
Motor bellows for follow-up; (can be shifted to right or left)
Actuating fluid under pressure
Capillary tube for slow equalization of pressure
Diaphragm
Spring
Pressure connection
Fluid under pressure
Space in which pressure is to be maintained constant
Control Valve

Fig. 18. Relay Pressure Control with Yielding Return. Capillary Resistance Tube

Another type of anticipatory control, which is useful for applications having considerable time lag, does actually anticipate the change of controlled quantity at the measuring instrument.

Example.—In control of temperature in residences, the thermostat located in a room may be acted upon not only by the heat of the room air but also by a very small electrical heating-coil partly surrounding the thermostat and energized when the motor- or solenoid-operated burner of the furnace is in operation. When the burner is idle, the thermostat may be cooled by evaporating water.

**5. Interrupted or Intermittent Control.**—Regulating action is interrupted after an adjustable regulating movement and is resumed at intervals, if the control point has not yet returned to normal position. The waiting time between, and the extent of each of the regulating movements, are made adjustable. This type of control was designed to imitate the actions of a cautious human operator.

**6. Two-point and Three-point Control.**—This type of control also is called two-position and three-position control. Control systems using electric contacts are frequently of the on-or-off type, or of the high-or-low type. They are continually hunting, unless the motion of the motor is periodically interrupted, as above indicated, or two motors with differential action are used. The motion of electric motors cannot be damped without complication.

In the three-point control, when the controlled quantity is at the correct value or within a narrow range on either side of it, the control is at rest. Unless the band on either side of the correct value is made rather broad, this control likewise is of the hunting type. The amplitude and frequency of the hunting depend on the character of the controller as well as on that of the applications. Numerous innovations have been introduced in controllers in order to minimize it.

## 4. REQUIREMENTS FOR GOOD CONTROL (REGULATION)

**DIRECT-ACTING, SELF-OPERATING CONTROL.**—The combined measuring instrument and motor must be nearly free from friction, or must be freed from friction by vibration. In most cases, damping by liquid friction is needed. Mass of the instrument, and the distance through which the mass travels, should be small. The strength of the regulating instrument must be very great compared to the frictional resistance of the equilibrium-restoring mechanism. The latter must not exert a greatly variable force on the measuring instrument. Damping, as well as static fluctuation, should be adjustable. All time lags should be reduced to a minimum. Regulation (static fluctuation, stability) is required, at least temporarily, to prevent hunting.

**RELAY GOVERNING.**—The measuring instrument must be practically free from friction and have small inertia. The sensitivity of the control, that is, the motion of the regulating mechanism for unit change of controlled quantity, must be adjustable. The speed of adjusting of the regulating mechanism (controller time) must be adjustable in certain types. It should be quick at the start, slowing down as the new position is approached. A reset, follow-up, compensator, or double response, all these terms having the same general meaning, is desirable in most cases. In speed control, the relay must move quickly.

The relay mechanism must not exert so much reaction against the measuring instrument as to unbalance the latter. If danger of reaction exists, for instance, from a large pilot valve, a double or floating relay is used. See Fig. 11.

## 5. FORCES USED IN GOVERNING, REGULATION, AND CONTROL

**SPEED CONTROL.**—Centrifugal force (radial inertia) of weights pivotally mounted and revolving around a shaft is most commonly used. Oil pressure generated by a centrifugal pump, and air pressure generated by a fan or turbo-blower, also have been used. Tangential inertia (tangential acceleration) was tried on account of its anticipatory character, but is a failure unless coupled with centrifugal force, when it is useful for shaft governors.

**PRESSURE CONTROL.**—Pressure exerted against a floating bell, or a diaphragm membrane, or liquid in a U-tube manometer, furnishes the force for low pressures, while the Bourdon tube is useful for high pressures. A piston in a cylinder has too much friction, unless rotated, in which case it can be, and often is, used for high pressures. Change in rate of flow, either of the supply or demand, also has been used for pressure control. However it is unsatisfactory unless used in connection with control by force derived from controlled pressure.

**RATE-OF-FLOW CONTROL.**—This is reduced to pressure control by making use of the pressure drop through an orifice, nozzle, or Venturi tube; the only difference being that pressure acts on both sides of a diaphragm, instead of on only one side; or, instead, two floating bells are balanced against each other. A Pitot tube can be used in place of the orifice. In addition to constant pressure drop, if constant rate of flow is to be maintained, the viscosity of liquids and the density of gases must be kept constant. The former may call for auxiliary temperature control. The latter may require both temperature control and pressure control.

**CONTROL OF LEVEL OF LIQUID.**—The buoyancy of a float is the most commonly used force. In control of water level in boilers, difference of temperature between steam and water has been utilized. See Fig. 9.

**CONTROL OF TEMPERATURE.**—For temperatures below red heat, temperature changes often are converted into pressure changes by means of gases or vapors enclosed in bellows. See Fig. 10. Gases in bulbs connected to Bourdon tubes also are used. Difference in expansion of solids, exemplified by curved bimetallic strips, etc., also produces a force. Electric current, started by an electric resistance thermometer, flowing through a solenoid produces a force which is used for regulating purposes. For temperatures above red heat, thermocouples and radiation pyrometers are used. A thermoelectric current moves the pointer (control pointer) of a galvanometer, or potentiometer. The pointer is depressed at regular intervals and closes circuits on "high" or "low" contacts, whereby relays are set in action to start motors which move the equilibrium-restoring mechanism. Radiation pyrometers are used in the same way. They also are used to send current through a solenoid which deflects a regulating jet of oil. Instead of a pointer, a light beam may be used to actuate a photoelectric cell.

**CONTROL OF MIXTURE RATIO OF FLOWING FLUIDS.**—This is reduced to pressure control by letting each fluid produce a differential pressure, the same as in control of quantity of fluid flowing, and balancing the two pressure differences against each other. The densities of the two fluids must be controlled in order to avoid variations in the mixture ratio.

**CONTROL OF HUMIDITY.**—The hygroscopic properties of cross-grained wood or of human hair commonly are used in the primary element. In more elaborate instruments, wet and dry bulb temperature difference is used to effect the regulation by means of either vapor-filled thermometers or electric resistance thermometers or thermocouples.

**COMBUSTION CONTROL** generally consists of simultaneous control of furnace pressure and of fuel-to-air ratio. For pressure control see above. For clean gaseous fuels and for liquid fuels, control of fuel-air ratio is reduced to that of controlling the mixture ratio of flowing fluids, for which see above. If oil fuel is used, its temperature must be kept constant, to avoid changes in viscosity, because such changes upset the reading of the pressure differential. No force has been derived satisfactorily from the flow of raw producer gas, on account of clogging of instruments, in particular the measuring orifice, by tar.

Combustion control in the power plant involves control of powdered coal or of stoker feed, neither of which can be measured accurately. The consequence is that indirect methods are used. Since combustion control in the power plant is highly special, it cannot be described here.

**POTENTIAL (VOLTAGE) CONTROL OF ELECTRICAL GENERATORS.**—A force proportional to the voltage is derived from a current passing through a constant-resistance coil. In alternating-current generators, rectifiers sometimes are used to convert alternating current to direct current. The force, acting by rectilinear motion or by producing torque, changes the field resistance in direct-current generators, and either generator-field or exciter-field resistance in alternating-current generators. Stability easily is obtainable through magnetic damping. The use of electron tubes also gives powerful methods of control.

**PROGRAM REGULATION.**—Instead of the controlled quantity being held approximately constant, it may be made to vary automatically with time, according to a predetermined program. For example, the temperature of an annealing furnace may be made to increase gradually over a long period of time and then to decrease slowly, allowing the charge to cool at a rate in accordance with the metallurgical requirements. The general principle utilized in program regulation is the controlled shifting of the set point or control point by a cam actuated by a clock, or by some other form of constant-speed device, usually electric.

Not only one quantity but all the important quantities of a process may be made to vary automatically according to a program laid out in advance, by mounting several cams, controlling pressure, temperature, rate of flow, etc., respectively, on a shaft turned at constant slow speed under control of the clock. The program itself may be varied by changing the cams and the speed of the shaft.

**CYCLE CONTROLLERS** are similar to program controllers with the exception that they do not hold any quantity constant but simply open and close valves in a pre-determined sequence. An example is automatic reversal of blast-furnace stoves.

## 6. CONTROL TROUBLES AND THEIR REMEDIES

Regulation troubles may be due to causes having to do with the metering device, with the motor device or operating element, or with the connection between the two. Furthermore, these causes originally may have existed in, or even may be inherent in, the design of the regulator, or they may arise only after the device has been in operation for some time. In the latter case they usually are caused by wear, increased friction, clogging, or leakage.

**HUNTING.**—Troubles include hunting. In speed governing, hunting also is called racing, pumping, or surging. If the control acted correctly at first, and hunting develops only after the installation has been in operation for some time, the primary element first should be checked to see that it responds. If it operates correctly one or more of the following items may be the cause: Insufficient oil in the gag-pot; oil in gag-pot too light; adjustment of oil pot not tight enough; friction in governor or in instrument mechanism or in operating element or stuffing box. To ascertain whether or not friction is responsible for the hunting, move the governor collar or the controller by hand alternately in either direction with equal force. If hunting disappears while the controller is being oscillated, friction is responsible. To ascertain whether or not friction in the relay mechanism is too great for the relay force, try to move the mechanism by hand or by a lever in the direction of least resistance, and keep it up for some time, helping the relay along. If hunting becomes noticeably less, the friction is in the relay mechanism. Hunting also may be caused by leaky pilot valves and worn-out oil pumps; by too low a pressure of air supply or oil supply to the control; by dirt in the air- or oil-pilot valve; in electrically-operated devices, by too low a supply voltage, or dirty contacts. Lost motion in the relay mechanism has a similar effect. A weak spring or weight, keeping the joints in contact always in one and the same direction, can be used to eliminate lost motion. In devices using diaphragms, trouble may be caused by the latter wearing out or cracking.

In any control, trouble may be caused by overload, *i.e.*, demand for the controlled quantity beyond the range in which the control was designed to operate, or by any considerable change in the application or method of use.

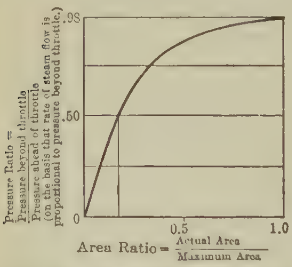


FIG. 19. Characteristic Curve of Throttling Valve

In speed governing, hunting or racing also may be caused in releasing-gear (Corliss) engines by dulling or rounding of the knife edges of the catch blocks, or by improper adjustment of the vacuum pots. Slipping governor belts, too, often are responsible. Speed adjustment of variable-strength governors often produces racing. If the racing starts with the first operation, it may have the same causes, but there are additional causes which, once removed, do not return. If racing occurs near one certain position of the governor or valve gear, the stability of the governor in that position is too small either from: 1. Wrong design of governor; 2. Because the power control mechanism does not vary the supply of energy uniformly with the travel of the governor; 3. Because forces from the power control mechanism react upon the governor. Case 1 can be determined by running the governor on a test block. All three cases usually yield to the application of springs which act over a small part of the governor travel only, and increase stability over that part. See Fig. 5, in which the spring enters into action near the no-load position. Case 2 frequently is encountered, because, near the closed position of a throttle valve, a small increase in area changes the flow through the valve much more than it does if the valve is almost wide open. See Fig. 19. This feature, which has been a prolific trouble maker in governing and regulation, can be counteracted by providing a linkage or cam arrangement between the governor, or other primary element, and the throttle, or else by a valve design in which a given increment of valve lift uncovers fewer square inches near the closed position than it does near the open position of the valve.

In addition to the cited causes, hunting at any load may be caused by too small a static fluctuation of the primary element. This can be remedied by using a greater part of the instrument travel, or by adding a stability-producing spring to the apparatus.

# BRAKES *

**FRICTION BRAKES** are used: 1. To slow or stop a machine by converting its energy of motion into heat; 2. To control the speed of a descending load. Brakes may be classified as block brakes, band brakes, axial brakes, and load brakes.

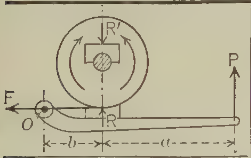
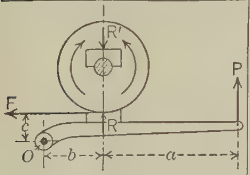
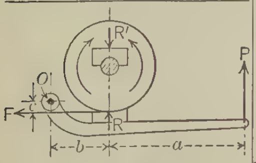
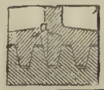
The following points must be considered in the design of brakes: 1. Each part must have sufficient mechanical strength safely to withstand the maximum load that will come upon it during operation. 2. The pressure per unit area of surface must not be taken so high that excessive wear will result. 3. The coefficient of friction should be carefully considered; its value depends on the materials in contact, on the presence or absence of lubricating matter, and on the rate of slippage. With metal on metal, with increased velocity of slip, the coefficient of friction decreases from its value as the coefficient of friction of rest; the reverse, however, is true of leather. 4. The heat generated by the rubbing surfaces must be so dissipated as not to injure the brake.

## 1. BLOCK BRAKES

**RESISTANCE OF BLOCK BRAKES.**—The simplest form of brake is the block brake, commonly used for railway car wheels, and in some forms for hoisting engines. In the car wheel brake the resistance is not only that of sliding friction, but the resistance of cutting or grinding away the surface of the metals in contact. If $P$ = total pressure acting normal to the sliding surface, lb., $\mu$ = coefficient of friction, $V$ = velocity of sliding, ft. per min., then the energy absorbed by the brake is $PV\mu$ ft.-lb. per min. Values of $\mu$ may be taken from the tables, pp. 8–23 to 8–28. It should be noted that the value of $\mu$ is different for lubricated and unlubricated surfaces. If pressure $P$ is great enough to

**Table 1.—Block Brake Formulas.**

$P$ = force at end of brake lever, lb.; $F$ = tangential force at rim of brake wheel, lb.; $R$ = reaction between brake wheel and block, lb.; $\mu$ = coefficient of friction; $P(a + b) = R\,b$.

| | |
|---|---|
|  | Line of action of $F$ passing through fulcrum $O$<br>For rotation in either direction<br>$F = \mu R = \frac{\mu P(a+b)}{b}$ . . . . . . . [1]<br>$P = \frac{F\,b}{(a+b)}\left\{\frac{1}{\mu}\right\}$ . . . . . . . . [2] |
|  | Line of action of $F$ passing above fulcrum $O$<br>$F = \mu R = \frac{P(a+b)}{b\{(1/\mu) \mp (c/b)\}}$ . . . [3]<br>$P = \frac{F\,b}{(a+b)}\left\{\frac{1}{\mu} \mp \frac{c}{b}\right\}$ . . . . . [4]<br>The minus sign is used for clockwise, and the plus sign for counterclockwise rotation. |
|  | Line of action of $F$ passing below fulcrum $O$<br>$F = \mu R = \frac{P(a+b)}{b\{(1/\mu) \pm (c/b)\}}$ . . . . [5]<br>$P = \frac{F\,b}{(a+b)}\left\{\frac{1}{\mu} \pm \frac{c}{b}\right\}$ . . . . . [6]<br>The plus sign is used for clockwise, and the minus sign for counterclockwise rotation. |
|  | For grooved brake wheel and block,<br>Substitute $\frac{\mu}{\sin\alpha + \mu\cos\alpha}$ for $\mu$ in the above equations<br>$\alpha = 1/2$ included angle of groove faces. |

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* Staff revision.

squeeze out the lubricant, the value of $\mu$ increases, and the surfaces will cut and wear with an increase of temperature.

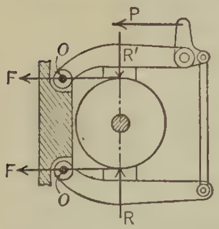


FIG. 1. Double Block Brake

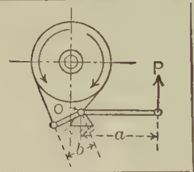
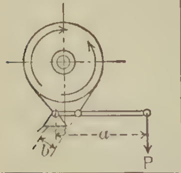
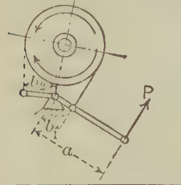
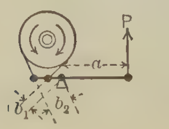
**Block Brake Formulas.**—Various arrangements of block brakes are possible. Formulas for different arrangements of single block brakes are given in Table 1. If the faces of the sliding surfaces are grooved, the values given by formulas [1] to [6] should be corrected as shown in the table. In the single block brake, the reaction $R$ of the brake shoe is opposed by an equal and opposite reaction $R'$ at the bearing. The braking effect resulting from the friction of the journal can be neglected in designing the brake, but should be considered in the design of the bearing. The single-block brake tends to bend the shaft. In the double-block brake, Fig. 1, reactions $R$ of the two blocks are equal and opposite, and tendency to bend the shaft is eliminated. The braking force is equal to twice the value given by equation [2]. If the line of action of $F$ passes either inside or outside of fulcrums $O$, the braking load is not the same at the two blocks, and a tendency to bend the shaft exists. The total braking load is the sum of the loads given by equations [3] or [5] as the case may be.

## 2. BAND BRAKES

A band brake consists of a band or strap, usually of steel, sometimes of leather wrapped around the major portion of the circumference of the brake wheel, so connected to levers that it may be tightened on the wheel. In another form, used in some automobile brakes, the band is expanded against the internal circumference of a hollow brake drum. The

**Table 2.—Formulas for Band Brakes**

$P$ = force at end of brake lever, lb.; $T_1$ = tension in tight side of band, lb.; $T_2$ = tension in slack side of band, lb.; $F = T_1 - T_2$ = tangential force at rim of brake wheel, lb.; $\mu$ = coefficient of friction between band and wheel; $e$ = base of Naperian system of logarithms = 2.17828; $\alpha$ = angle of contact of band with wheel, radians (1 radian = 57.296 deg.).

| | |
|---|---|
|  | $T_1 = F\dfrac{e^{\mu\alpha}}{e^{\mu\alpha}-1}$; $T_2 = F\dfrac{1}{e^{\mu\alpha}-1}$. . . . . . [7]<br>For clockwise rotation, $P = \dfrac{Fb}{a}\left\{\dfrac{e^{\mu\alpha}}{e^{\mu\alpha}-1}\right\}$ . . . . . . . . [8]<br>For counter-clockwise rotation, $P = \dfrac{Fb}{a}\left\{\dfrac{1}{e^{\mu\alpha}-1}\right\}$. . . . . [9] |
|  | For clockwise rotation, $P = \dfrac{Fb}{a}\left\{\dfrac{1}{e^{\mu\alpha}-1}\right\}$ . . . . . . . . [10]<br>For counter-clockwise rotation, $P = \dfrac{Fb}{a}\left\{\dfrac{e^{\mu\alpha}}{e^{\mu\alpha}-1}\right\}$ . . . . [11] |
|  | For clockwise rotation, $P = \dfrac{F}{a}\left\{\dfrac{b_2 e^{\mu\alpha} - b_1}{e^{\mu\alpha}-1}\right\}$ . . . . . . . [12]<br>For counter-clockwise rotation, $P = \dfrac{F}{a}\left\{\dfrac{b_2 - b_1 e^{\mu\alpha}}{e^{\mu\alpha}-1}\right\}$. . . . . [13]<br>Unless $b_2 > b_1 e^{\mu\alpha}$ the band will automatically grip the wheel and cause rupture of the band or some other part. |
|  | For clockwise rotation, $P = \dfrac{F}{a}\left\{\dfrac{b_2 e^{\mu\alpha} + b_1}{e^{\mu\alpha}-1}\right\}$ . . . . . . . [14]<br>For counter-clockwise rotation, $P = \dfrac{F}{a}\left\{\dfrac{b_1 e^{\mu\alpha} + b_2}{e^{\mu\alpha}-1}\right\}$ . . . . [15]<br>If $b_1 = b_2 = b$, for both cases, $P = \dfrac{Fb}{a}\left\{\dfrac{e^{\mu\alpha}+1}{e^{\mu\alpha}-1}\right\}$. . . . . [16] |

brake wheel may be of cast iron or steel, or it may be faced with leather or wood. The band itself may be lined with a special brake lining, as in automotive practice, possessing a high coefficient of friction and high temperature resistance. See Kent's Mechanical Engineers' Handbook—Power, p. 14–73, forming Vol. 2 of this series.

**RESISTANCE OF BAND BRAKES.**—The theory of band brakes is the same as that of the friction of belts. See p. 24–14. The fundamental equations are:

$$F = T_1 - T_2 \qquad [17]$$

and

$$T_1/T_2 = e^{\mu\alpha} = 10^{0.00758\mu\theta} \qquad [18]$$

where $F$ = total friction of the band on the wheel, $T_1$, $T_2$ = respectively, tensions in tight and slack sides of the band; $e$ = base of Naperian system of logarithms = 2.71828; $\mu$ = coefficient of friction; $\alpha$ = arc of contact of band with the brake wheel, radians; $\theta$ = arc of contact, degrees. Values of $e^{\mu\alpha}$ are given in Table 3. Table 2 shows various arrangements of band brakes and the formulas applicable thereto.

**Table 3.—Values of $e^{\mu\alpha}$**

| Proportion of Circumference | Radians $\alpha$ | Deg. $\theta$ | Values of $\mu$ | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0.1 | 0.2 | 0.25 | 0.3 | 0.35 | 0.4 | 0.45 | 0.5 | 0.6 | 0.7 | 0.8 |
| | | | Values of $e^{\mu\alpha}$ | | | | | | | | | | |
| 0.1 | 0.6283 | 36 | 1.065 | 1.134 | 1.170 | 1.207 | 1.246 | 1.286 | 1.327 | 1.369 | 1.458 | 1.552 | 1.653 |
| .2 | 1.2566 | 72 | 1.134 | 1.286 | 1.369 | 1.458 | 1.552 | 1.653 | 1.760 | 1.874 | 2.126 | 2.410 | 2.733 |
| .25 | 1.5708 | 90 | 1.170 | 1.369 | 1.481 | 1.602 | 1.733 | 1.874 | 2.028 | 2.193 | 2.566 | 3.003 | 3.514 |
| .3 | 1.8850 | 108 | 1.207 | 1.458 | 1.602 | 1.760 | 1.934 | 2.126 | 2.336 | 2.566 | 3.099 | 3.742 | 4.518 |
| .4 | 2.5133 | 144 | 1.286 | 1.653 | 1.874 | 2.126 | 2.410 | 2.733 | 3.099 | 3.514 | 4.518 | 5.809 | 7.468 |
| .425 | 2.6704 | 153 | 1.306 | 1.706 | 1.950 | 2.228 | 2.546 | 2.910 | 3.326 | 3.801 | 4.964 | 6.484 | 8.468 |
| .45 | 2.8274 | 162 | 1.327 | 1.760 | 2.028 | 2.336 | 2.690 | 3.099 | 3.569 | 4.111 | 5.455 | 7.237 | 9.602 |
| .475 | 2.9845 | 171 | 1.348 | 1.816 | 2.109 | 2.448 | 2.842 | 3.300 | 3.831 | 4.447 | 5.994 | 8.078 | 10.888 |
| .5 | 3.1416 | 180 | 1.369 | 1.874 | 2.193 | 2.566 | 3.003 | 3.514 | 4.111 | 4.811 | 6.586 | 9.017 | 12.346 |
| .525 | 3.2987 | 189 | 1.391 | 1.934 | 2.281 | 2.690 | 3.173 | 3.742 | 4.412 | 5.204 | 7.237 | 10.067 | 13.999 |
| .55 | 3.4558 | 198 | 1.413 | 1.996 | 2.373 | 2.820 | 3.352 | 3.984 | 4.736 | 5.629 | 7.593 | 11.235 | 15.873 |
| .575 | 3.6128 | 207 | 1.435 | 2.060 | 2.467 | 2.956 | 3.541 | 4.243 | 5.082 | 6.089 | 8.738 | 12.541 | 17.999 |
| .6 | 3.7699 | 216 | 1.458 | 2.126 | 2.566 | 3.099 | 3.742 | 4.518 | 5.455 | 6.586 | 9.602 | 13.999 | 20.409 |
| .65 | 4.0841 | 234 | 1.504 | 2.263 | 2.776 | 3.405 | 4.176 | 5.123 | 6.283 | 7.707 | 11.594 | 17.442 | 26.241 |
| .7 | 4.3982 | 252 | 1.552 | 2.410 | 3.003 | 3.742 | 4.662 | 5.809 | 7.237 | 9.017 | 13.999 | 21.733 | 33.739 |
| .75 | 4.7124 | 270 | 1.602 | 2.566 | 3.248 | 4.111 | 5.204 | 6 586 | 8.336 | 10.551 | 16.903 | 27.078 | 43.379 |
| .8 | 5 0266 | 288 | 1.653 | 2.733 | 3.514 | 4.518 | 5 809 | 7.468 | 9 602 | 12.346 | 20.409 | 33.739 | 55.774 |

## 3. AXIAL BRAKES

Axial brakes are those in which resistance is applied by the axial movement of the braking members. Two general forms are in use, cone brakes and disc brakes.

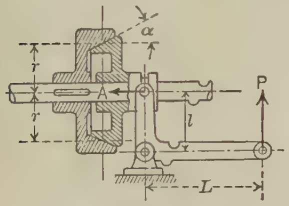


Fig. 2. Cone Brake

**CONE BRAKES.**—Fig. 2 illustrates one form of cone brake. Force $P$ applied to the brake lever $L$ causes axial movement of the brake cone into its mating shoe, thus creating resistance to the rotation of the member to which the shoe is attached. Let $M$ = twisting moment on brake shaft, in.-lb.; $r$ = mean radius of brake wheel, in.; $P$ = pull on lever, lb.; $F = M/r$ = force at the circumference of the brake wheel, lb.; $\alpha$ = half angle of the cones; $\mu$ = coefficient of friction between brake wheel and shoe; $A$ = axial pressure, lb., along the cone shaft forcing the cones together; $N$ = normal pressure on the cone, in order that the braking action shall overcome the turning moment; $L$ = length of lever, in.; $l$ = distance from fulcrum of brake lever to axis of cone. Then

$$F \gtrless 2\mu N \qquad [19]; \qquad A = PL/l \qquad [20]$$

$$N = \frac{A}{2(\sin\alpha + \mu\cos\alpha)} \qquad [21], \quad \text{or} \quad \frac{A}{2\sin\alpha} \qquad [22]$$

$$F \gtrless \frac{A}{\sin\alpha + \mu\cos\alpha} \qquad [23], \quad \text{or} \quad \frac{\mu A}{\sin\alpha} \qquad [24]$$

$$P \lessgtr \frac{Fl\sin\alpha + \mu\cos\alpha}{\mu L} \qquad [25], \quad \text{or} \quad \frac{Fl\sin\alpha}{\mu L} \qquad [26]$$

Angle $\alpha$ usually ranges from 10 to 18 deg. For cast iron on cast iron, $\mu$ can be taken as 0.12; for one surface of wood and the other of cast iron, $\mu$ = 0.25 may be used.

From an experimental investigation of a cone brake by Prof. H. Bonte (*Zeit. V. D. I.*, Dec. 15, 1915), Prof. Leutwiler (Machine Design, p. 419) derives the following conclusions concerning formulas [23] and [24]: 1. When $\alpha$ is 15 deg., the error introduced by using [23] is large, while the agreement between the experimental results and those obtained by using [24] is very close. 2. When $\alpha$ is 30 deg., the experimental results lie between those obtained by [23] and [24]. When plotted, the curve representing the results obtained by [24] lies above, but much closer to the experimental curve than that obtained by [23]. 4. For the angles 45 deg. and 60 deg., the experimental points lie above those obtained by [24], which in turn lie above the points obtained by [23]. 5. Apparently, the coefficient of friction is not constant, as generally assumed, but varies slightly with the pressure. As a result of this experimental investigation, Prof. Bonte recommends that [23], which apparently is incorrect, should not be used in the design of cone brakes or clutches.

**DISC BRAKES** are of the same general construction as cone brakes, excepting that the friction faces are discs instead of cones. Most brakes are of the multiple-disc type, *i.e.* having more than two friction faces in contact. With the same arrangement for applying axial pressure as in Fig. 10, let $n$ = number of friction surfaces in contact; $r$ = mean radius of friction discs; other notation as before. Then

$$A = F/n\mu r = Pl/L \qquad [27]$$

$$P = FL/n\mu lr \qquad [28]$$

$$F = Pn\mu lr/L \qquad [29]$$

See Load Brakes below for formulas dealing with the application of axial pressure by means of a screw instead of a lever.

## 4. LOAD BRAKES

Load brakes to control the speed of descent of load on cranes and hoists may be manually or automatically operated. They are made in several forms.

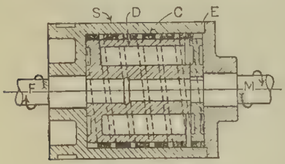


Fig. 3. Coil Brake

**COIL BRAKES.**—Fig. 3 shows a coil brake. Drum $D$ is keyed to the driven or follower shaft $F$ but is free to turn on the motor shaft $M$. Coil $C$ is made of flat rod wound into a helix, ground internally and externally. The external diameter is made slightly larger than the internal diameter of sleeve $S$ so that the coil must be slightly compressed when the brake is assembled. One end of the coil is fastened to flange $E$, which is keyed to shaft $M$; the other end is secured to drum $D$. In hoisting, motor shaft $M$ turns in the direction of the arrow at the right; this action expands the coil against sleeve $S$ and causes drum $D$ and shaft $F$ to revolve with the casing. If the motor stops, holding the load suspended, shaft $F$ and drum $D$ under the action of the load tend to turn in the direction of the arrow $F$; this expands the coil against the sleeve $S$, which is prevented from turning by a ratchet, not shown, on the end. To lower the load, motor $M$ reverses its direction, thus releasing coil after coil from contact with the casing; when sufficient coils have been thus released, shaft $F$ turns under the action of the load, in the direction of the arrow, which has the effect of again expanding the coil sufficiently to hold the load.

The coil brake is, in effect, an internal band brake, and the formulas used in the computation of band brakes may be applied. See p. 15–16. In applying these formulas $T_1/T_2 = K$. The value of $K$ may be obtained by $\log_e K = \mu\, l/r$, where $l$ = length of circular arc of coil in contact with the drum, and $r$ = radius of drum.

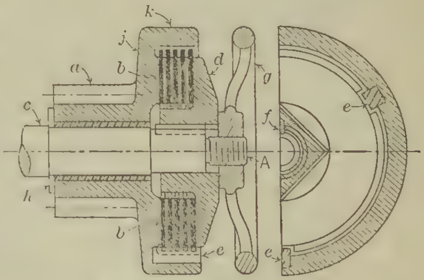


Fig. 4. Weston Brake

**DISC LOAD BRAKE.**—Fig. 4 shows a multiple-disc brake of the Weston type. This brake consists of a pinion $a$, bushed at $h$, and with a flange at $j$ which is extended to form the casing $k$; this casing encloses a series of friction discs which are made to rotate with it by means of the feathered keys $e$. Alternating with these discs are others fitted to the square hub of the flange $d$. The pinion $a$ is free on the shaft but the flange $d$ is keyed to the shaft at $f$. The hand-wheel $g$ enables the operator to control the speed of the load by increasing or decreasing the pressure between the discs. The shaft is held from running backward by a ratchet and pawl not shown.

Formulas [27] to [29] would be modified by use of the hand-wheel and screw as follows: Let $A$ = axial thrust on friction discs; $F$ = frictional tangential force on discs; $M_1$ = frictional moment between flange $d$ and hub of hand-wheel $g$; $n$ = number of frictional faces in sliding contact; $P$ = tangential force at circumference of hand-wheel; $R$ = radius of hand-wheel; $r$ = mean radius of friction discs; $r_1$ = mean radius of screw; $\mu$ = coefficient of friction between discs; $\theta$ = helix angle of screw; $\phi$ = friction angle of screw; all dimensions are taken in inches and all forces in pounds. Then

$$A = \{P R/r_1 (\tan \theta + \phi)\} - M_1 = F/n\mu r \quad . \quad . \quad . \quad . \quad [30]$$

$$F = n \mu r [\{P R/r_1 \tan (\theta + \phi)\} - M_1] \quad . \quad . \quad . \quad . \quad . \quad . \quad [31]$$

$$P \overset{=}{>} \{F r_1 \tan (\theta + \phi)/n \mu r\} + M_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [32]$$

Fig. 5 shows a form of disc load brake for traveling cranes. The pinion $F$ drives the hoisting drum. In hoisting, gear $A$, driven by the motor, is rotated on the screw in a direction that causes it to move axially on the shaft and clamp the friction faces $E$ of the ratchet wheel $B$ between the gear $A$ and the flange $C$ which are fast on the shaft. Ratchet wheel $B$ is prevented from rotating backward under the pull of the load by the pawl $D$. The load thus is sustained so long as pressure is maintained between $A$ and $C$. When the load is lowered, $C$ is held stationary by pawl $D$. The direction of rotation of gear $A$ is reversed, tending to unscrew it and remove the frictional resistance between it and flange $C$. However, the load will rotate the shaft, thus tending to screw the shaft into gear $A$ and so restore the friction. These two tendencies balance each other, and the load descends under control and at a speed determined by the speed of the motor. The force required to raise and lower loads is given by equations [33] and [34], in which $P$ = tangential force acting at mean radius of load, $r_1$ = mean radius of thread; $r_2$ = mean radius of friction faces $E$; $s$ = pitch of thread; $\mu$ = coefficient of friction between friction faces of $A$, $B$, and $C$; $\mu_1$ = coefficient of friction between screw and nut; $W$ = load. All dimensions are in inches, and all loads and forces in pounds.

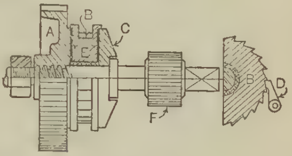


Fig. 5. Load Brake for Traveling Crane

To raise load $$P r_1 = W r_1 \left\{\frac{s + 2 \mu_1 \pi r_1}{2 \pi r_1 - \mu_1 s}\right\} + \mu W r_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad [33]$$

To lower load $$P r_1 = W r_1 \left\{\frac{2 \mu_1 \pi r_1 - s}{2 \pi r_1 + \mu_1 s}\right\} + \mu W r_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad [34]$$

In determining the value of $P r_1$ for raising the load, the acceleration of the load must be considered and added to the actual load being lifted. The acceleration load is equal to $W a/g$, where $a$ = acceleration, ft. per sec. per sec., and $g$ = acceleration due to gravity = 32.2.

MAGNETIC AND ELECTRIC BRAKES.—For braking the load on electric cranes a band brake sometimes is used which is held off the drum by the action of a magnet or solenoid, and is put on by the action of a spring or weight. The solenoid usually consists of a coil of wire connected in series with the motor, and a plunger working inside of the coil. It should be so proportioned that its action is not delayed by residual magnetism when the current is cut off. Too rapid action is prevented by making the end of the solenoid an air dash-pot.

For electrically-driven machinery an electric motor makes a most efficient brake by reversing the direction of the electric current, causing the motor to become a generator supplying current to a rheostat in which it is converted into heat and dissipated. In some cases the electric current generated, instead of being absorbed in a rheostat, is fed into the main electric circuit. In this case the energy of the rotating mass, instead of being wasted in friction or in electrical heating, is converted into electric energy and thus conserved for further use.

## 5. COEFFICIENT OF FRICTION AND HEAT LOSSES

COEFFICIENT OF FRICTION.—The coefficient of friction is highest for static conditions and decreases with increase of velocity. According to Galton (Hess, Mach. Design), the coefficient of friction of cast-iron brake shoes on steel tires varies with the velocity as follows:

| | | | | | | |
|---|---|---|---|---|---|---|
| Velocity, ft. per min. | Beginning | 440 | 880 | 2200 | 4000 | 5250 |
| Coefficient of friction | 0.330 | 0.273 | 0.242 | 0.166 | 0.127 | 0.074 |

Wichert has determined the effect of velocity upon the coefficient of friction for cast-iron brake shoes upon steel-tired car wheels to be approximated by the following formula,

$$\mu = B\{(1 + 0.002\,V) \div (1 + 0.0011\,V)\} \qquad [35]$$

where $B$ = coefficient of friction of rest under the given condition of the surfaces relative to each other; $V$ = velocity, ft. per min.

Brake tests made by the Penna. R.R. indicate that coefficient of friction decreases with hardness of the cast-iron brake shoe, and that the maximum possible coefficient of friction is obtained with a Brinnell hardness of about 190.

The coefficients of friction for materials used for brakes, for velocities ranging from 180 to 3900 ft. per min., and pressures from 7 to 140 lb. per sq. in. are given in Table 4.

**Table 4.—Coefficients of Friction for Brake Materials.**

(From Various Sources)

| Brake Material | Value of $\mu$ on | | Brake Material | Value of $\mu$ on | |
|---|---|---|---|---|---|
| | Cast Iron | Wrought Iron | | Cast Iron | Wrought Iron |
| Beech | 0.29–0.37 | 0.54 | Fiber | 0.10–0.20 | ........ |
| Bronze, lubricated | 0.06–0.10 | .... | Leather, dry | 0.3 | ........ |
| Cast iron, dry | 0.25–0.3 | .... | " , lubricated | 0.23 | ........ |
| " " lubricated | 0.08–0.12 | .... | Oak | 0.30–0.34 | 0.40–0.51 |
| Cork * | 0.35 | .... | Poplar | 0.35–0.40 | 0.49–0.65 |
| Elm | 0.36–0.37 | 0.49–0.60 | Willow | 0.46–0.47 | 0.60–0.63 |

* See also p. 8–23.

**HEATING OF BRAKES.**—The energy absorbed by a brake must be dissipated as heat. Consequently, the brake should be designed with ample radiating surface, ribs or fins being provided if necessary, to promote rapid dissipation of heat. Referring to Fig. 5, the energy absorbed by a disc brake is as follows: Let $E$ = energy absorbed by brake, ft.-lb. per min.; $r_o$, $r_i$ = respectively, inner and outer radii of friction discs, in.; $r$ = mean radius of screw, in.; $n$ = number of friction surfaces in contact; $N$ = revolutions per min.; $p$ = total pressure on friction discs, lb.; $S$ = area of radiating surface, sq. in.; $H$ = total heat to be dissipated, B.t.u. per min.; $T$ = torque, in.-lb., on pinion $F$; $\theta$ = helix angle of screw; $\phi$ = friction angle of screw. Then

$$E = \frac{2}{3}\left(\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right)\frac{\pi}{12}\,n\mu N p \qquad [36]$$

$$p = T/r \tan(\theta + \phi) \qquad [37]$$

$$H = E/(778 \times 2\,S) \qquad [38]$$

The heat dissipation is a function both of the temperature of the brake and of the time that it is at rest betweeen applications, during which heat is being radiated. Let $m$ = number of minutes the brake is at rest; $t_1$, $t_2$ = minimum and maximum temperatures of the brake drum, deg. F.; $c$ = mechanical equivalent of the specific heat of the brake-drum material = (specific heat/778); $k$ = radiation factor; $w$ = weight of brake drum rim, lb.; $S$ = area of radiating surface, sq. in. Then

$$\log(t_2 - t_1) = 0.434\,k\,S\,m/c\,w \qquad [39]$$

and

$$E = c\,w\,(t_2 - t_1) \qquad [40]$$

Values of $k$ range from 0.4 to 0.8 per min. per deg. F. temperature difference above the surrounding air. Values of $t_2$ may range from 200° F. with wood friction blocks, to 500° F. with asbestos friction surfaces.

# Section 16

# VIBRATION AND NOISE

By Arthur L. Kimball

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# VIBRATION AND NOISE

By Arthur L. Kimball

## VIBRATION

**FREE AND FORCED VIBRATIONS. RESONANCE.**—It is important to differentiate between *free* and *forced* vibrations. Free vibration occurs when the driving force is removed, dying out more or less gradually. Forced vibration is maintained by a driving force. The former has a frequency of its own; the latter can have any frequency, depending on that of the driving force. If the frequency of the driving force is in step with the natural or free period of the vibration, the forced vibration will build up to a large amplitude. The vibrating body is then in *resonance* with the exciting force.

Any body which vibrates freely when the exciting force is removed illustrates free vibration, *i.e.*, a weight vibrating on a spring. Free vibrations, as such, are not important in vibration prevention. Forced vibrations give trouble if an exciting stimulus is present. Resonant types, where the exciting force is in step with the free vibration period, are the most difficult. Knowledge of free vibration period thus is important. An example of a forced vibration is an unbalanced rotor which quivers at all speeds, due to unbalanced centrifugal force; the frequency always equals the speed frequency. If speed is in step with the free vibration frequency of the rotor between bearings, vibration will reach a peak value, and illustrate resonance. All critical speed vibrations are resonant. Calculation of critical speeds is the determination of free vibration periods to which the body resonates.

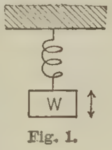


Fig. 1.

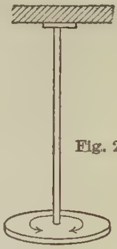


Fig. 2.

## 1. VIBRATION FREQUENCY CALCULATION

The following simple cases of calculation of natural vibration frequency are given for reference. Two basic formulas are:

**Linear vibration** for spring supported weight (Fig. 1), $f = (1/2\pi)(\sqrt{kg/W}$ . . . [1]

**Torsional vibration** for disk on rod (Fig. 2), $f = (1/2\pi)\sqrt{L_1 g/I}$ . . . . . . [2]

where $f$ = frequency, cycles per sec.; $g$ = acceleration of gravity = 386 in. per sec.$^2$; $k$ = stiffness constant of spring = force required to stretch it to unit distance, lb. per in.; $L_1$ = same constant in torsion, or force required to twist spring or rod through 1 radian of arc, ft.-lb. per radian; $W$ = weight, lb.; $I$ = moment of inertia, lb.-in.$^2$ = $WR^2$; $W$ = weight of body, lb., executing torsional vibrations; $R$ = radius of gyration. Formula [1] is used more often than [2], as most problems involve linear, rather than torsional, vibrations.

Substituting the value of $g$ for the inch-pound-second system, [1] and [2] reduce to

**Linear vibrations.** $f = 3.13\sqrt{k/W}$ cycles per sec. . . . . . . . . . [3]

**Torsional vibrations.** $f = 3.13\sqrt{L_1/I}$ cycles per sec. . . . . . . . . . [4]

**GRAVITY DEFLECTION FORMULA.**—For a spring carrying a weight (Fig. 1) causing extension $d$ of the spring, elastic constant $k$ must equal $W/d$ = weight per unit deflection. Substituting this in [3],

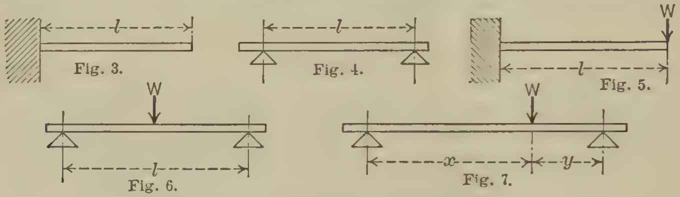
$$f = 3.13/\sqrt{d} \quad . . . . . . . . . . . [5]$$

where $f$ = cycles per sec.; $d$ = in. deflection. Thus if a weight causes an elastic support of relatively small weight to deflect downwards through $d$ in., [5] at once gives the up-and-down frequency. This formula is useful for quick, approximate estimates of certain vibration frequencies. Thus, a shaft carrying a relatively heavy fly-wheel has a deflection of $d$. The frequency and the critical speed (see p. 11-10) can be estimated, without knowing weight of the fly-wheel.

**VIBRATION OF UNIFORM BARS.**—For any vibrating bar of uniform cross-section, the frequency is given by

$$f = (k/l^2)(\sqrt{EI/w}) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [6]$$

where $k$ depends on shape of bar and how it is supported, and $l$ = length, in. For a cantilever bar (Fig. 3), $k = 11$; for a beam supported at the ends (Fig. 4), $k = 31$. Table 1 gives frequencies for circular and rectangular bars. In this table, $I$ = moment of inertia of cross-section, in.$^4$ (not mass moment of inertia); $w$ = weight, lb. per in.; $w_1$ = weight, lb. per cu. in.; $l$ = length, in.; $D$ = diam., in.; $h$ = depth, in., $E$ = modulus of elasticity.



Figs. 3–7. Vibration of Uniform Bars

**VIBRATION OF LOADED UNIFORM BARS.**—Table 2 gives frequencies of loaded bars, where weight of bar is neglected compared with weight it carries. Symbols have same meaning as in Table 1. These formulas are derived from [5] by substituting proper value of $d$ produced by a load $W$ for each case.

**VIBRATION OF CIRCULAR DISC OF UNIFORM THICKNESS SUPPORTED AT THE CENTER.** (Fig. 8.)—Such a plate has exactly the same vibration frequencies as one perfectly free; the formula is

$$f = K(h/r^2)\sqrt{(Eg/\rho)} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [7]$$

where $h$ = thickness of disc, in.; $r$ = radius of disc, in.; $E$ = modulus of elasticity; $g$ = 386 in. per sec.$^2$; $\rho$ = density of disc material, lb. per cu. in.; $K$ = constant depending on the number of nodes $n$ along disc edge; when $n = 4$, $K = 0.25$; $n = 6$, $K = 0.54$. This assumes the absence of ring nodes, which case will be omitted.

## 2. VIBRATION MEASUREMENT

To measure vibration, frequency and amplitude must first be determined; without knowledge of these, quantitative study is impossible. Also, form of path described, *i.e.*, circular, elliptical, etc., and the phase relative to another vibration sometimes may be necessary.

**REED FREQUENCY METER.**—The most generally useful frequency meter is of Frahm tachometer type, Fig. 9. It comprises a series of reeds, fixed on a common base,

**Table 1.—Frequencies for Circular and Rectangular Bars.**

| | Uniform Bar, cycles per sec. | Circular Section, cycles per sec. | Rectangular Section, cycles per sec. |
|---|---|---|---|
| Cantilever (Fig. 3) | $11/l^2\sqrt{EI/w}$ | $(2.7\,D/l^2)(\sqrt{E/w_1})$ | $(3.18\,h/l^2)(\sqrt{E/w_1})$ |
| Beams supported at ends (Fig. 4) | $31/l^2\sqrt{EI/w}$ | $(7.7\,D/l^2)(\sqrt{E/w_1})$ | $(8.96\,h/l^2)(\sqrt{E/w_1})$ |

**Table 2.—Vibration of Loaded Uniform Bars.**

| Type of Loading | Frequencies, cycles per sec. | |
|---|---|---|
| | Circular cross-section | Rectangular cross-section |
| Loaded cantilever (Fig. 5) load on end | $1.2\,D^2\sqrt{E/W\,l^3}$ | $1.56\sqrt{E\,b\,h^3/W\,l^3}$ |
| Loaded beam (Fig. 6) loaded in center of span | $4.8\,D^2\sqrt{E/W\,l^3}$ | $6.24\sqrt{E\,b\,h^3/Wl^3}$ |
| Loaded beam (Fig. 7) load off center of span | $1.2\,D^2\sqrt{\frac{E\,l}{W\,x^2\,y^2}}$ | $1.56\sqrt{\frac{E\,b\,h^3\,l}{W\,x^2\,y^2}}$ |

carrying masses at their tips, and tuned to respond one after another over a range of frequencies. A convenient model about 6 in. long is made by Leeds and Northrup Co., Philadelphia, covering frequencies from 900 to 12,000 per min. by means of three instruments. It is easily portable, readily applied, and shows rapidly-varying frequencies as well as more than one frequency at a time if present. Above 12,000 cycles per min. (200 per sec.), sensitivity is lost due to stiffness of the reeds. For best response the meter should rest solidly against the vibrating body, direction of reed vibration being roughly parallel to direction of motion of exciting vibration. This instrument is an excellent substitute for speed counters and speed tachometers.

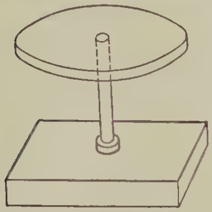


Fig. 8. Vibration of Circular Disc Supported at Center

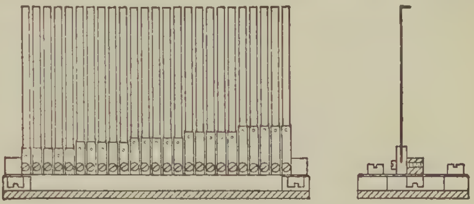


Fig. 9. Frahm Tachometer

**INERTIA AMPLITUDE INDICATOR.**—One form of instrument for amplitude measurement is the inertia type, Fig. 10. This is an ordinary Starrett dial strain gauge reading in mils, set in a hole in a block of metal weighing about 2 lb.; axis of indicator stem passes approximately through center of gravity of the block. The weight is so held that the indicator stem makes contact with the vibrating body, the metal block being kept free. The indicator stem oscillates with the vibrating body, but the weight to which the indicator body is set does not oscillate because of its inertia. If the vibrating body has an amplitude of, say, 3 mils in the direction of motion of the stem a blurred image 3 mils wide appears on the indicator dial scale. The scale numbering has no relation to the amplitude, which changes its position on the scale as the indicator is allowed to approach or recede from the vibrating body. If the indicator is held reasonably steady, the amplitude of the oscillation is easily read. This instrument is portable and easily applied, with the indicator stem pointing in any direction necessary to observe vibration amplitude. It is satisfactory for amplitudes of 10 mils or more for frequencies of ordinary rotating machinery. Precautions in using this instrument are: Avoid chattering of indicator button for frequencies over 100 per sec. with amplitudes of over 2 or 3 mils; for high frequencies the button may not follow the vibration if amplitude is too great. For frequencies below 5 per sec., the weight itself may vibrate and thus decrease indicated amplitude. In special cases, the amplitude indicator without the inertia part can be rigidly attached to a supporting post if a convenient point of attachment *free from vibration* is available.

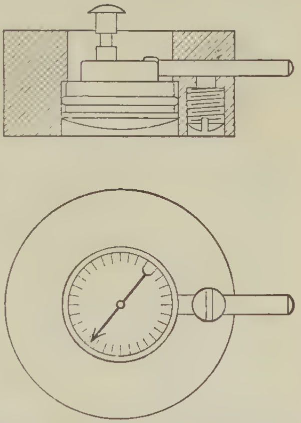


Fig. 10. Inertia Amplitude Indicator

**THE VIBROGRAPH** will make a complete graphical record of a vibration, giving a permanent record of both frequency and amplitude. A good instrument, made by Cambridge Instrument Co., traces the record by means of a fine stylus on moving celluloid film. It will easily record amplitudes under 0.5 mil, of frequencies over 100 per sec. The Geiger vibrograph is a well-known instrument which records with a pen on a moving strip of paper Both instruments are actuated by a spring-suspended mass on the seismograph principle. They are rather elaborate, with levers, etc., and lost motion must be compensated. For a more detailed description, see Vibration Problems in Engineering, S. Timoshenko, p. 327 *et seq.*, D. Van Nostrand Co. 1928.

**TORSIOGRAPH.**—To detect torsional oscillations in a revolving shaft, as a ship propeller shaft or a Diesel engine shaft, governed by a heavy fly-wheel, a Geiger or other torsiograph, should be used. This is an inertia instrument comprising essentially one part rigidly attached to the shaft, taking part in its torsional oscillations, and a fly-wheel, mounted in its own bearings and driven through a flexible spring at the shaft speed.

The fly-wheel maintains substantially constant angular velocity. A multiplying lever mechanism responds to shaft oscillations with respect to the non-oscillating fly-wheel, and records oscillations in a graph (see Timoshenko, *loc. cit.*). There is no satisfactory way of detecting torsional oscillations of a revolving body by such a simple device as used for linear vibrations of a stationary body. The avoidance of torsional oscillations of this character is very important, as many failures have been traced to torsional resonance involving excessive periodic stresses in shafting (see p. 16–11).

**ELECTROMAGNETIC METHODS.**—One type of electrical vibration measuring instrument records by motion of a pin placed against the vibrating body, registering in the same way that a loud speaker records oscillations on a phonograph record. This instrument can be made very sensitive to small amplitudes and high frequencies, as those producing sound radiation. Another instrument for lower frequencies and larger amplitudes, depends on the fluctuation of electrical impedance produced by air gap variation caused by the vibration, which in turn modulates the amplitude of a relatively high-frequency exciting electric current.

Both methods readily record frequency on the oscillograph, for example, but translation of electrical into mechanical amplitudes requires careful calibration over the frequency range, especially for lower frequencies. This usually is due to the spring applying the recording pin to the vibrating body being too stiff to deflect through full vibration amplitude. The author has encountered errors from this cause in this type of instrument. See Vibration Recorder, A. V. Mershon, *Trans.* A.I.E.E., 1926.

## 3. VIBRATION PREVENTION

Of the several methods of attack in vibration reduction, the particular method or methods adopted depend on the individual problem. Briefly these methods are:

**REMOVAL OF EXCITING CAUSE.**—This, when possible, is most satisfactory. Unbalance in rotating machinery is a common source of trouble. In rigid rotors it can be almost completely eliminated by balance weights. Gear tooth frequency noise is best avoided by precision in manufacture.

**TUNING** is applicable only when vibration is distinctly of resonant type. Resonant vibrations usually are excited by a relatively small stimulus, which happens to be in step with a natural vibration frequency. The tuning of steam turbine wheels is an example, also the avoidance of shaft critical speeds through selection of proper diameter at the operating speed.

**ELASTIC SUSPENSION.**—The function of elastic suspension is to *isolate* a powerful forced vibration so that it cannot be transmitted beyond the vibrating piece of apparatus. This is not a friction damping process. Springs do not absorb vibrations through damping, but are often a very effective means of preventing transmission; *e.g.*, the spring suspension of a railroad car isolates from the car body forced vibration imparted to the wheels; the leaf spring contains some friction damping which eliminates resonant bouncing, but has no part in isolation. It can be shown that such friction in springs and shock absorbers somewhat increases forced vibration transmission. The amount of increase is small, even when friction is considerable. Friction is necessary to avoid building up resonant vibrations.

Elastic suspension methods are much used to suspend noisy machinery, etc., but there is still much guesswork in their application. An incorrectly designed elastic suspension may do more harm than good. The design of a good suspension is not always an easy matter and often requires knowledge of vibration principles as well as experience.

**DAMPING,** or reduction of vibration amplitude through action of friction, is a cure for resonant vibrations only. Forced vibrations are apt to result from powerful exciting forces which are not much reduced by application of friction devices. For example, gear tooth frequency hum or whistle is a forced vibration, which is reduced but little by stuffing the wheel with a non-resonant damping substance with internal friction. On the other hand the resonant ring of the wheel due to its natural periods, may be greatly reduced by such treatment. Shock absorbers used in vehicle suspension are typical friction dampers which limit the amplitude of resonant vibrations excited by road irregularities.

### Removal of Exciting Force

**MECHANICAL BALANCE OF MACHINERY.**—It is possible to completely mechanically balance any rigid rotor, whatever be its unbalance or its shape, by only two weights. The problem is to determine the position and the size of these two weights. Balancing is a fine art, and trial-and-error methods by a clever mechanic generally will waste money.

**STATIC AND DYNAMIC BALANCE.**—Mechanical unbalance in a rigid rotor always can be completely represented schematically as in Fig. 11. The weight $A$ takes account of displacement of the center of gravity of the rotor from its axis; weights *B* and *C* represent a couple, depending on how much the rotor is cocked or twisted sideways from its true axis. The three weights can be replaced by two by splitting *A* into two parts, placing one in the plane of *B* and the other in the plane of *C*. The two weights in each of these planes then can be replaced by a single one in each plane by applying the parallelogram of forces (centrifugal forces in this case). See A. L. Kimball, Vibration Prevention in Engineering, pp. 85–94 (John Wiley & Sons, Inc., 1933) and S. Timoshenko, Vibration Problems in Engineering, p. 157 *et seq.* (Van Nostrand). That part of unbalance represented by *A*, Fig. 11, is *static unbalance*; it can be detected by a static or gravity test by placing the rotor on ways and noting its heavy spot. The part represented by weights *B* and *C* is *dynamic unbalance*; it can be detected only by a rotational or dynamic test which develops a centrifugal couple. A rotational test, however, does not separate dynamic from static unbalance, but combines the effect of both; otherwise, mechanical balance would be comparatively simple. For this reason, specially designed machines and special methods, described below, must be used to determine the amount and position of balance weights.

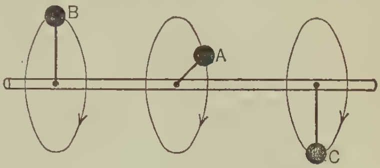


Fig. 11. Mechanical Unbalance

**BALANCING MACHINES.**—For large production, some type of good balancing machine is essential. Many are unsatisfactory. A good machine is the Precision balancing machine built by the Gisholt Mfg. Co. Another widely used German machine built by Karl Schenk, the Lawaczeck-Heymann machine, is fairly satisfactory for rotors of 500 lb. and up, and is less expensive than the former. The Precision machine has proved itself of particular value in balancing automobile crankshafts. E. L. Thearle (General Electric Co.) has produced the best balancing machines in existence for that company's type of work; these have not been sold outside to any extent (1933). Tinius Olsen Co. has produced a very good machine for small rotors.

Balancing is a highly specialized art. Different balancing problems require different machines, designed for the particular conditions to be met. Only large and fairly uniform production justifies the design and purchase of such a machine.

## Balancing of Special Jobs

**SMALL RIGID ROTORS.**—The following method of balancing small rigid rotors gives good results, with a minimum of apparatus. The rotor, *R*, Fig. 12, is placed in a light wood frame *F*, resting in half bearings in the frame, and driven by motor *M* through belt *L*, whose tension is adjusted by screw *S*. The rotor is first pivoted in plane *C*, the pivots in plane *D* being withdrawn. A spring, not shown, is placed under the frame at bearing *B*, of sufficient stiffness to cause resonance at 700 to 800 r.p.m. The rotor being driven through the resonant frequency, the deflection indicator *E* is adjusted with its button resting on top of the frame to register vibration. The arc through which the needle oscillates approximately measures unbalanced moment about fulcrum in plane *C*.

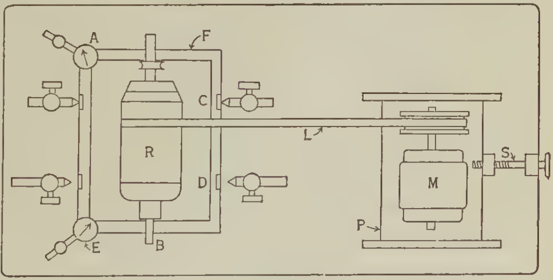
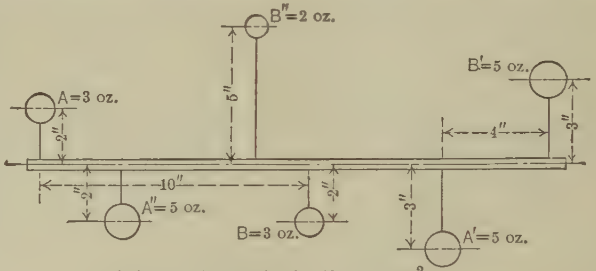


Fig. 12. Balancing Set

To find amount and position of the weight, vibration amplitude is read and a small trial weight successively placed in four positions 90° apart in plane *D*; the amplitude of vibration is read in each case. A curve of rise and fall of amplitude is plotted against angular position of trial weight, from which amount and position of weight can be estimated. Inspection of the curve shows position where weight subtracts most from the unbalance and adds most to it; the former is the correct position of balance weight. Its amount is estimated by comparing change of vibration amplitude produced by the weight in different positions with the original vibration amplitude, assuming vibration amplitude to be a direct measure of weight required. The trial weight for small rotors may be a piece of wax.

Besides having a light frame to insure large amplitudes, the outfit must not vibrate as a whole due to insecure foundation; it must be placed where no floor vibrations are transmitted to it, otherwise indicator readings will be inconsistent. Furthermore, the frame must be sufficiently stiff to resist all tendency to flexure. A good way to isolate the apparatus from such vibrations and prevent detrimental shaking is to place the outfit on a massive iron plate resting on toy rubber balls or an inflated automobile inner tube. The driving motor must be well balanced, and a light belt used to give a constant tension perpendicular to the oscillatory motion of the frame; the pull should not be too great. To balance the other end of the rotor, the process is repeated, using the other fulcrum. Weights so determined should produce complete balance so that the indicators show no oscillatory motion.

**AIRPLANE PROPELLER** unbalance may be made up of unbalanced air reaction and mass unbalance. If one blade exerts slightly greater traction on the air than the other, an unbalanced couple is present which revolves with the propeller, always tending to deflect the forward end of the propeller shaft towards the blade exerting the smaller traction, thus causing the engine to wobble. Since traction on one propeller blade is several hundred pounds, any dissymmetry between blades, either of shape or pitch, produces a relatively large unbalanced couple, and aerodynamic unbalance may far exceed pure mechanical unbalance. Hence, extremely precise mass balance of airplane propellers may be disappointing. Little systematic study has been made of this problem, and quantitative data are lacking; the slighest change in pitch of a blade noticeably affects plane vibration. The problem is complicated by blade deflections during operation which must react on the traction. Propellers of high power planes are subject to severe engine torque pulsations with resulting vibration, but the exciting force comes from the engine.



Unbalanced Couple=3 x 2 x 10=60 oz.-in.$^2$
I. Balancing Couple=5 x 3 x 4 = 60 oz.-in.$^2$
II. Balancing Couple=2 x 5 x 6 = 60 oz.-in.$^2$ or 5 x 2 x 6=60 oz.-in.$^2$

FIG. 13. Methods of Balancing Centrifugal Couple

**RULES FOR DESIGN.**—A good rule for design of rotating apparatus with moving parts, where unbalanced forces are present, is to keep the center of gravity of the complete moving system as nearly as possible on the rotation axis for all positions of the shaft as it revolves. Next, estimate as nearly as possible, the centrifugal couple which remains (weights $B$ and $C$, Fig. 11) and place *anywhere* on the shaft a pair of weights to produce a counteracting couple. Fig. 13 shows two possible ways of balancing a centrifugal couple produced by weights $A$ and $B$ in one plane, namely, by $A'$, $B'$, or by $A''$, $B''$. Note that the pair of weights may be placed anywhere in this plane along the shaft if the couple produced is equal and opposite in magnitude to that of the original weights. The couple in each case is measured by the product of either one of the two weights times its distance from the shaft times its distance along the shaft to the second weight of the pair. The two weights do not have to be equal, but the products (magnitude of a weight × distance to shaft axis) must be equal. The use of unequal weights and axial distances is shown where the pair $A'' + B''$ are counterweights.

**WORKMANSHIP.**—In symmetrical rotors with perfect workmanship and materials, there should be no unbalance. For a $^1/_2$-Hp., 1800 r.p.m. motor, for example, an unbalanced weight of 0.05 oz. will produce appreciable vibration of say 2 mils amplitude. So small an unbalance cannot be avoided by ordinary shop processes, even with the best supervision. It is cheaper to manufacture rapidly and balance later.

**MEASUREMENT OF AMOUNT OF UNBALANCE.**—It is sometimes necessary to form a quick estimate of the amount of unbalance in a machine. To measure this in an electric motor, place it on rubber pads which should compress at least $^1/_8$ in. under the motor at usual speeds. Maximum vibration amplitude in a direction transverse to shaft is an approximate measure of the amount of unbalance. This is best determined by an inertia amplitude indicator, Fig. 10. The suspension must be at least as yielding as specified, otherwise the vibration produced will be either too small or greatly amplified through resonance.

Another form of suspension used by the author to estimate unbalance is to hang the motor by a relatively long spring through straps or a *light* cradle, preferably of wood, to reduce inertia. Weight of the motor should stretch the spring perceptibly. Vibration amplitude is measured as before by amplitude indicator. This method offers less restraint to the vibrating body than mounting on rubber pads.

The object of this test is to find how unbalanced apparatus vibrates when unrestrained by outside forces. The exciting stimulus acts almost entirely against the inertia of the body itself, and oscillations produced are proportional in amplitude to the exciting force.

In single-phase motors, a vibration of considerable intensity may be present, due to a torque pulsation of twice the alternating current frequency, *i.e.*, 120 cycles per sec. for a 60-cycle circuit (see Single-phase Motor Torque Pulsations, Kimball and Alger, *Trans.* A.I.E.E., 1924, p. 730). This frequency is much higher than ordinary rotation speeds, and thus much higher than vibration from unbalance which is always in step with r.p.m. frequency. It shows up well with the above suspensions. It is easily separated because of its higher frequency, and its causing the motor to oscillate about the shaft as a stationary axis. Mechanical balance will not cure this electrical vibration.

Motor hums from electrical causes mostly are eliminated by elastic suspension. See p. 16–16. By these methods it is possible to quickly select the best balanced among a number of similar motors, if means for balancing are not available.

**ELIMINATION OF SELF-INDUCED VIBRATIONS.**—A troublesome type of vibration that occurs in rotating machinery operating above the critical speed is termed *self-induced vibration.* Its frequency is always the same as the critical speed frequency, which may be lower than the r.p.m. frequency of the shaft in the same way that the revolution frequency of the earth around the sun is much lower than its rotation frequency about its axis.

**Whirling Due to Oil Action in Journal Bearings.**—This type of self-induced vibration arises only when the rotation of the machine is more than twice the critical speed frequency. It is produced by wedges of oil driving the journal around in the bearing and exciting a whirl. If a well-balanced machine tends to unstable operation, and vibration appears when the rotation frequency is more than twice the critical speed, this kind of whirl is the probable cause. The best method of elimination is to insure that the critical speed frequency is more than half the r.p.m. frequency of the shaft. Another remedy, in smaller rotors, is to replace journal bearings by ball bearings.

**Whirling Due to Frictional Action within the Rotor.**—Any rotor with self-contained friction, produced by flexure while revolving, will build up a whirl at any rotational speed higher than the critical. Friction of this type may be produced by rings so shrunk on a shaft that working may occur between the bore of the rings and the surface of the shaft. Frictional working also can arise in laminated structures, carried by the shaft, which structures flex and rotate at the same time.

These phenomena usually occur in motors with flexible shafts which operate far above the critical speed, and in such devices as extractor type washing machines, which contain rotating parts operating well above the lower critical speed.

## 4. VIBRATION ELIMINATION THROUGH TUNING

### Shaft Whirling

**CRITICAL SPEEDS.**—When a shaft or rotor, revolving in bearings, speeds up to where centrifugal force tending to whip it sideways just balances the elastic stiffness tending to keep it straight, the slightest mechanical unbalance builds up a whirling motion, which may be of destructive amplitude. The rotor then is revolving at a *critical speed.* It is inadvisable to run any rotor or shaft at or near critical speed for any length of time, regardless of perfection of balance. Critical speeds usually are avoided by designing the rotor shaft sufficiently stiff to hold its operating speed well below its critical speed. This sometimes is impossible for very heavy or very high speed rotors of long span between bearings. The shaft then is made sufficiently flexible to insure operation well above critical speed. This practice is to be avoided if possible, as vibration may develop from other causes in operation above critical speeds, although, so far as unbalance is concerned, operation is stable except close to critical speed.

**CRITICAL SPEED CALCULATION.**—Critical speeds of a rotor are almost identical with natural vibration frequencies which the stationary rotor may assume.

**Uniform Shaft.**—The first critical speed is given by the formula

$$7.7(d/l^2)\sqrt{E/w_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8],$$

where $f$ = critical frequency, cycles per sec.; $d$ = shaft diameter, in.; $l$ = span between bearings, in.; $E$ = modulus of elasticity shaft material; $w_1$ = weight of 1 cu. in. of shaft material, lb. This formula is given in Table 1, for a circular bar vibrating between two end supports. For steel, the formula becomes $f = 7900\ d/l^2$ . . . . . . . . . . [9]

At the first critical speed, the shaft whirls in one loop with nodes at the bearings; at second critical speed, in two loops with a node in the middle, at 4 times the first speed; at third critical speed, in three loops at 9 times the first critical speed, etc. Upper critical speeds, which enter only in high-speed installations, are special problems in themselves.

**Shafts with Concentrated Loads.**—A useful approximate formula derived by Dunkerley from analysis, checked by careful tests on models, for a uniform shaft carrying concentrated loads, Fig. 14, is

$$1/f^2 = 1/f_s^2 + 1/f_1^2 + 1/f_2^2, \text{ etc.}, \qquad [10]$$

where $f$ is critical frequency of the whole system; $f_s$ that of shaft *by itself* as calculated from [8]; $f_1$ that of shaft considered as weightless, with load $W_1$ by itself, given by the last formula in column 1 of Table 2, etc., for the other concentrated loads. It is evident that values for $x$ and $y$ in the latter formula depend on location of $W_1$, $W_2$, etc., on the shaft. Quantities are in., lb. and sec. This formula should not be used for 3-bearing sets.

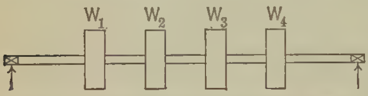


FIG. 14. Shaft with Concentrated Loads

**Shaft of Varying Cross-section and Load.**—When cross-section of shaft is not uniform, and the load is distributed along it, as in most rotors, the problem is not so simple. It can be approximately solved from knowledge of the static load-deflection curve, as vibration frequency calculated on the basis of this deflection curve is only slightly higher than that based on the true deflection curve for actual vibration.

When static deflection curve is known, vibration frequency or critical speed frequency easily is calculated from

$$f = 3.13\sqrt{\frac{W_1 y_1 + W_2 y_2 + W_3 y_3 + \text{etc.}}{W_1 y_1^2 + W_2 y_2^2 + W_3 y_3^2 + \text{etc.}}} \qquad [11]$$

where shaft and rotor are divided into a series of slices of convenient length of weight $W_1$, $W_2$, etc. (including loads), and corresponding deflections $y_1$, $y_2$, etc. See Fig. 15. All quantities are in., lb. and sec. The problem is primarily one of static deflection curve determination, which, in cases similar to Fig. 15, requires using graphical statics. For the solution of such a problem, see Timoshenko, Vibration Problems in Engineering.

**Three-bearing Sets.**—A 3-bearing problem can be solved by formula [11]; both first and second critical speeds should be found, as they may not lie far apart, while the second critical of a rotor with only one span will be at least twice the first. At the first critical speed of a 3-bearing set, one span deflects upward when the other deflects downward; in finding static deflection curve, loads on one span must be taken as acting upwards. Deflection of one span influences that of the other through a bending moment across the middle bearing. The deflection curve determination must take account of this. The second critical speed occurs at a somewhat higher frequency where both spans deflect in the same direction, with a point of inflection in the shaft near the middle bearing. The static deflection curve then is found from loads all acting in one direction, but again, bending moment is transmitted across the middle bearing. The best procedure for this special problem is to refer to more advanced treatments noted in preceding paragraph.

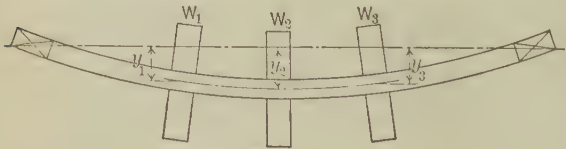


FIG. 15

**BEARING ELASTICITY.**—All critical speeds are actually somewhat lower than calculation indicates, as no bearing is perfectly rigid. Also, up-and-down stiffness of a bearing is apt to be much greater than its sidewise stiffness. Due to this, two critical speeds may be present; one involves an elliptical whirl where the motion is mostly up and down; the other, at a speed a few percent lower is sidewise elliptical motion. The lower frequency results from greater sideways elastic yielding of the bearings. For very heavy rotors, critical speed may be 30% lower than calculated for rigid bearings.

**ELLIPTICAL SHAFTS.**—If a shaft is not circular, but elliptical or flattened on opposite sides, being stiffer in one axial plane than in one 90° away, bad performance must be expected in operation near critical speed. The rotor evidently will vibrate in one plane at a somewhat lower frequency than in the plane for which it is stiffer. In an ideal case, with no damping, operation is unstable between these two frequencies. A model shaft and fly-wheel, with the shaft flattened on two sides vibrates violently near the critical speed, and must be restricted by a guard ring. Rotors which perform badly near the critical speed are open to suspicion of being built stiffer in one plane than another.

**TORSIONAL VIBRATION OF LOADED SHAFTS.**—Critical torsional vibration frequencies as well as transverse criticals may occur in shafts carrying fly-wheels, and subject to periodic torque impulses, such as occur in Diesel engines. Torsional critical speeds are more insidious than transverse, as they are not evidenced in operation by excessive vibration, as are the transverse criticals. Since dissipation is small, operation on a torsional critical speed causes large torsional stresses which often result in fatigue failure.

Loaded shafts only are treated below. Torsional inertia of shaft itself is neglected, as it usually is small. This cannot be done in all cases, but in many problems nearly all torsional inertia is concentrated in certain planes, as in shafts carrying fly-wheels and crank-shafts. See Tulpin, Torsional Vibration (John Wiley & Sons) and Wilson, Practical Solution of Torsional Vibration Problems (Wiley).

**SHAFT CARRYING THREE FLY-WHEELS.**—Fig. 16 shows a shaft carrying 3 fly-wheels, one of which may be the equivalent of a crank-shaft. This system has two critical frequencies, given by the expression

$$f = 3.13\sqrt{(-B \pm \sqrt{B^2 - 4AC})/2A} \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

where $A = (I_1 I_2 I_3)/(k_1 k_2)$; $B = -[\{(I_1 I_2 + I_1 I_3)/k_1\} + \{(I_2 I_3 + I_1 I_3)/k_2\}]$; $C = I_1 + I_2 + I_3$; $I_1, I_2, I_3$ = moments of inertia of the three fly-wheels; $k_1, k_2$ = torsional elastic constants of the two shaft lengths between the fly-wheels, in.-lb. per radian of twist.

For a disc-shaped fly-wheel of weight $W$ lb. and radius $r$ in.,

$$I = (Wr^2/2) \text{ lb.-in.}^2 \quad . \quad . \quad . \quad . \quad . \quad [13]$$

If the shafting which spans distance between two fly-wheels is of uniform diameter, for a given span

$$k = \pi N a^4/2l \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [14]$$

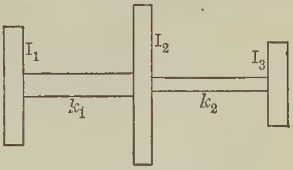


Fig. 16. Shaft Carrying Three Fly-wheels

where $N$ = rigidity or shear modulus of shaft material; $a$ = shaft radius, in.; $l$ = length of span, in. The lower of the two frequencies given by [12] is that for which the inside radical is negative. The first and last fly-wheels vibrate in opposite phase with a node somewhere between. If the system is made up of three identical fly-wheels connected by two identical spans the node is exactly at the middle fly-wheel, which remains stationary while the two end ones vibrate against each other. For the upper of the two frequencies, the end fly-wheels vibrate in phase; the middle one is in phase opposition, with a node somewhere along each connecting shaft length, making two nodes in all. No more resonant frequencies are possible with this system.

**SHAFT WITH ONE FLY-WHEEL AT EACH END** is solved by formulas [12], [13] and [14], where $I_3 = 0$, in which case [12] reduces to the simple form,

$$f = 3.13\sqrt{(I_1 + I_3)k/I_1 I_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15]$$

where $k = k_1 k_2/(k_1 + k_2)$. This system contains one resonant frequency. Torsional elastic constants $k_1$ and $k_2$ are calculated by [14] for shaft sections of uniform diameter as before, where $l_1$ and $l_2$ are lengths of the two shaft sections, and all quantities are in in.-lb.-sec. system. If the shaft is of uniform diameter from $I_1$ to $I_3$, the constant $k$ can be directly found by [14].

**DESIGN CONSIDERATIONS.**—The torsional stiffness of shaft and inertia loads must be such that none of the calculated natural frequencies is in step with any periodic impulse from the driving engine. An engine with several cylinders will produce several torque impulses per revolution. If, for example, 12 torque impulses per second are present at operating speed, there are also present $^{12}/_{2}$, $^{12}/_{3}$, $^{12}/_{4}$, $^{12}/_{6}$, $^{12}/_{12}$ that is, 6, 4, 3, 2 and 1 impulses per second, as well as 12, which may excite resonance. Safety requires the system to be so stiff that all its critical frequencies are above the highest torsional impulse frequency. No resonance then can be built up. This is particularly necessary in the massive systems used with Diesel engines. Friction devices sometimes are used for smaller engines, whereby one or more inertia parts are permitted to turn on the shaft against friction clamps; when torsional oscillations develop, the clamps damp them out. Friction devices of this kind often give trouble.

Most actual cases cannot be treated by the simple analysis given above, because of varying shaft sections or the presence of more than three inertia parts, etc. For more complicated problems in this field, as automobile crank-shaft design, see Timoshenko, Vibration Problems in Engineering; Root, Dynamics of Engine and Crank-shaft Design (John Wiley & Sons). An understanding of general principles, however, should enable a designer intelligently to approach problems of this kind, and to guard against vibration.

**TURBINE WHEEL VIBRATION. Cause.**—The edge or periphery of an elastic disc will transmit a train of flexural waves around it at a characteristic speed determined by mass per unit area and stiffness of the disc. As the boundary of the disc has a finite circumferential length, the wave lengths must have certain fixed values; *e.g.*, the disc edge may carry a continuous train of 2, 4, 6 or 8, etc., waves, whose wave length is $^1/_2$, $^1/_4$, $^1/_6$, $^1/_8$, etc., of the disc circumference. Each wave train has a different velocity, longer wave lengths moving at considerably higher speeds than shorter. In a disc of uniform thickness, supported at the center, wave speed, rev. per sec., can be calculated by

$$\text{rev. per sec.} = [K(h/r^2)\sqrt{Eg/\rho}]/2n \quad . \; . \; . \; . \; . \; . \; . \; [16]$$

where $K$ depends on $n$, the number of nodal points along the disc edge; for values of $K$ and meanings of symbols, see p. 16–04.

Turbine wheels are not of uniform thickness, and must be calculated by special methods. For detailed discussion of turbine wheel vibration and design, see W. Campbell, Protection of Steam Turbine Disc Wheels from Axial Vibration, *Trans.* A.S.M.E., vol. xlvi, p. 31, 1924.

Critical speeds exist for turbine wheels at which operation is hazardous; large wheels have several such speeds. These critical speeds are equal to the several wave train speeds given by [16] except for a correction necessary due to stiffening effect of centrifugal force, which makes wave speeds for a revolving wheel somewhat higher than those given by [16] for a stationary wheel. See Campbell's paper noted above.

When a turbine wheel runs at one of its criticals, the wave train at whose speed the wheel revolves will be stationary in space, because the wave train moves backwards in the wheel at same speed that the wheel turns forward. A small axial pressure on the wave zone of the wheel near its outer edge, *i.e.*, the bucket zone, may sustain and build up such a standing wave train. Thus only the wave train moving backward in the wheel is built up, but the corresponding forward moving wave train is not maintained.

Long waves, corresponding to 4 or 6 radial nodes, build up more easily than wave trains of shorter wave length. Large, relatively thin wheels are more susceptible to wave-building action due to the greater area on which steam acts, and their greater flexibility, permitting small initial disturbances from which wave trains build up to be more easily excited. For the same reason large-amplitude waves build up easily, with resulting fatigue failures of disc material.

**Prevention.**—The two-fold remedy is: 1. Design the wheels to avoid operation at all critical speeds or wave train speeds. For small turbines, this usually can be done; with large wheels it is not practical for the highest speed wave train, unless the wheels are excessively heavy. Next best is to place operating speed half way between that of the 4-node and 6-node wave train. 2. Do not depend too much on avoiding criticals, but keep the buckets and wheels sufficiently stiff, based on past successful operation.

**Precautions.**—It is not safe to depend on calculation alone. Measured frequencies, from which critical speeds can be calculated, may diverge 10% from the best calculations, due to initial stresses in the disc itself, or to cramping of buckets. A hoop tension or compression along the outer zone of the disc will alter frequency materially. For details see Campbell's paper above noted.

**GEAR WHEEL VIBRATIONS. Torsional Oscillations in Gear Trains.**—Fig. 17, illustrates a system of gears and fly-wheels, subject to torsional oscillation. The two frequencies of this system are best calculated by reducing to a system equivalent to Fig. 16, by replacing $I_1$ and $K_1$, Fig. 17, by $I_1'$ and $K_1'$, where $I_1' = n^2 I_1$ and $K_1' = n^2 K_1$, $n$ being gear ratio. The inertia of the pinion has been neglected. If moment of inertia $I_2$ of the gear wheel is small compared with $I_1$ and $I_3$ of the fly-wheels, $I_2$ may be taken as zero. There then is but one resonant frequency, found by

$$f = \sqrt{\{(I_1' + I_3)/I_1'I_3\}\{K_1'K_2/(K_1' + K_2)\}} \quad . \; . \; . \; . \; . \; [17]$$

It is remarkable that in this case the pair of gears gives the same frequency that would obtain if the shafts were fixed end to end with no gears between, regardless of value of gear ratio $n$. This can be checked by substituting $I = I'/n^2$ and $K = K'/n^2$ in [17], and comparing with result obtained when $I'$ and $K'$ are used. The relative vibration *amplitudes*, however, depend directly on $n$; that is, in Fig. 17, when gear train is present, for a given oscillation amplitude of $I_3$, that of $I_1$ is $1/n$ times its value for the shafts direct connected without the intermediate gears. For more than one pair of gears the problem is more complex. See Timoshenko, Vibration Problems in Engineering, p. 154, for a mathematical analysis of this problem.

**Vibration of Gears Due to Impact.**—Practically all gears produce a hum of gear tooth frequency, which in high-speed gearing under load, may be very penetrating. This is primarily a forced vibration, and will be little helped by tuning. The only sure cure is precision of manufacture, together with a design which permits of perfectly constant angular velocity for all tooth positions. Sometimes a resonant ring in gear wheels also is present. This type of noise can be largely eliminated by methods given under head of vibration damping. See p. 16–17.

**VIBRATION OF STRUCTURES.**—All structures and buildings are subject to resonant vibrations of several frequencies; the lower frequencies arise from a swinging of the whole structure; the higher are due to vibrations of component parts. The best way to eliminate such vibrations, when set up by a periodic force of known frequency, is some method of tuning.

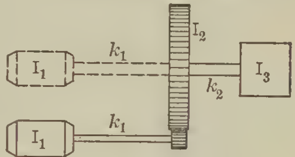


FIG. 17. Torsional Oscillation in Gear Train

**FOUNDATION VIBRATIONS OF MACHINERY.**—Foundations of heavy machinery may be structures of considerable size. Sidewise and up-and-down frequencies in a steam turbine should be roughly estimated in design and compared with the revolution frequency, or, for other types of machinery, with the predominant exciting frequency. Accurate calculations of frequencies of such structures are not worth while, due to their complexity, unless a thorough study is contemplated. Rough estimates based on common-sense application of vibration principles give almost as reliable results as more refined calculations.

The method of procedure is: Reduce the system to a single lumped mass vibrating against single stiffness constants, one for up-and-down motion, one for sidewise motion, etc., and use formula [3]. For example, a structure, Fig. 18, carries rotating machinery of weight $W_1$. The total weight used to calculate up-and-down vibration frequency includes, say, half the structure floor, as only the middle portions vibrate up-and-down to any extent. For sidewise vibration, the total floor weight and part of the columns is included, as the whole floor and column tops move through approximately full amplitude. Constant $K$ is found in each case by estimating the downward force on the platform and the sidewise force, respectively, in pounds required to produce 1 in. of deflection. This requires some ingenuity, but with a little practice it is not difficult for an approximate result. Frequency depends on the square root of mass and stiffness, and error is minimized in the final result.

Even if the principal modes of vibration have frequencies a safe distance from the operating frequencies, individual parts of the structure may resonate, especially when built of steel members. To cure resonance of individual members, stiffen or load them in different ways until vibration disappears. This usually is not difficult. It evidently is impractical to attempt to predetermine frequencies of *all* the individual members of such a structure.

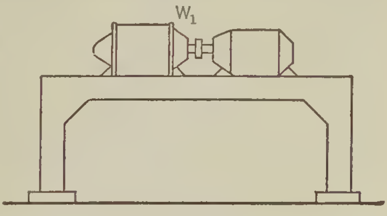


FIG. 18

**BUILDING VIBRATIONS.**—Ordinary buildings have sidewise natural frequencies of 2 or 3 per sec., as is clearly shown during slight earthquakes, or when delicate apparatus is being mounted.

**GROUND VIBRATIONS.**—Heavy machinery causes local vibration of its foundation, and also at times earth waves radiating in all directions to a considerable distance. If the foundation rests on bed rock, such transmission is negligible; with a foundation on clay or sand, and a periodic force of high intensity and low frequency, surface waves can be detected $1/2$ mile or more away in clay or sand beds. Ground waves may be avoided by placing heavy machinery foundations on bed rock, and by avoiding clay or sand flats, as well as unbalanced exciting forces. Bed rock, however, transmits higher frequency, objectionable sound vibrations better than clay.

**RESONANCE IN HELICAL SPRINGS.**—Helical springs subject to periodic compression and extension, as automobile valve springs, may resonate badly, with resulting fatigue failures. Longitudinal resonant frequencies in helical springs fixed at each end are calculated with considerable precision by

$$f = (n/2\,l)\sqrt{k\,g/w} = (9.8\,n/l)\sqrt{k/w} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [18]$$

where $n$ = number of half waves between fixed ends; $l$ = length of spring, in.; $w$ = weight per in. of axial length, lb.; $k$ = elastic constant of spring *per unit length, i.e.*, force, lb., necessary to double the length of spring; $f$ = cycles per sec. Thus the lowest frequency, where $n = 1$, corresponds to $^1/_2$ wave between ends; the next, where $n = 2$, to two half-waves, etc. See Fig. 19. To avoid these resonances, the spring must be so designed that none of these frequencies are in the operating range. If possible, its stiffness should be such that the lowest resonant frequency for which $n = 1$ is above the operating range of frequencies.

**SHIP VIBRATION.**—Hulls of ships, particularly steel ships, are subject to resonant vibrations. A long ship has several resonant frequencies in which flexural vibrations of the ship as a whole occur, with two or more transverse nodal lines. These frequencies can be calculated approximately, and should not be in step with exciting forces. The principal periodic exciting force to be avoided is propeller blade frequency, *i.e.*, rev. per sec. × number of blades. This force comes from a periodic reaction of water from the propeller on parts of the stern, which is difficult completely to eliminate. Such a condition of resonance may be tuned out to a large extent by altering cargo distribution, thus altering natural frequency.

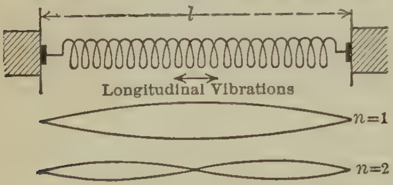


FIG. 19

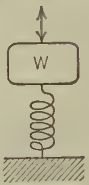


FIG. 20

## 5. VIBRATION REDUCTION BY ISOLATION

### Forced Vibrations

Forced vibrations may be cured by resorting to *isolation*; they do not yield readily to damping and cannot be tuned out. The standard method of isolation is to use an elastic suspension.

**ELASTIC SUSPENSIONS** function through ability to reduce the periodic force reaction of a vibrating body on its support. Fig. 20 shows the simplest possible spring suspension, where a weight $W$, with an up-and-down forced vibration of frequency $f$ imposed on it, is supported by a vertical spring. Transmissibility of such a spring support is given by

$$\text{Transmissibility} = \epsilon = f_0^2/(f^2 - f_0^2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [19]$$

where $f_0$ is natural or resonant up-and-down frequency of spring support as given by [3] and [5]. *Transmissibility* may be defined as the ratio of vibration forces *actually* transmitted to those which *would* be transmitted without spring suspension. Thus, to be of use, $\epsilon$ must be less than unity; experience shows that in most cases to be really effective $\epsilon$ should be as small as $^1/_{30}$ to $^1/_{40}$. This means that $f_0$ should be from $^1/_5\,f$ to $^1/_6\,f$. Fig. 21 shows a graph of [19] as dependent on $f/f_0 = r$.

**PRECAUTIONS IN USE OF ELASTIC SUSPENSIONS.**—All elastic suspensions necessarily have their own resonances. Thus if a suspension is designed to isolate a frequency of $f$ = 120 cycles per sec. with a resonant frequency $f_0$ = 30 cycles per sec., it must be certain that no 30-cycle exciting frequency is present in addition to the 120-cycle. This is especially true for rotating machinery, whose rotation speed is usually in that field. Fig. 21 shows that transmissibility is greater than unity when $f$ approaches $f_0$, and may be very large near resonance. Furthermore, $\epsilon$ is always greater than unity when $f$ is less than $f_0$. Nearly all elastic suspensions have resonance frequencies other than $f_0$ corresponding to the particular mode of vibration for which they are designed. This must be considered in design.

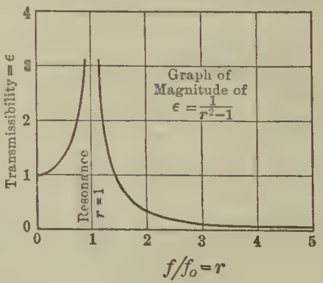


FIG. 21

Elastic suspensions do not damp a vibration by friction, but prevent transmission by power to isolate. Contrary to general belief, damping *increases* transmissibility, although the increase is small with a moderate amount of damping. See Kimball, Vibration Prevention in Engineering, pp. 96–107. A moderate amount of damping is of advantage as it safeguards against excessive vibration near resonance, but not because it improves isolation.

The following facts should be considered in design, including the best material to use. Cut-and-try methods will, in the long run, be costly; elastic pads of too large area in proportion to their thickness do not yield enough under gravity to isolate other than high frequencies. See formula [5] for frequency calculation from gravity deflection $d$.

**ELASTIC SUSPENSION MATERIALS.**—Steel, rubber, cork and gelatin compounds commonly are used for spring suspensions.

**Steel** has the advantage of being very durable. It contains practically no damping, and therefore is subject to possible resonances to other exciting forces than the one being screened.

**Rubber** is more flexible than steel from the design standpoint. See discussion of elastic suspension of motors, p. 16–16. It has some damping. To avoid age-hardening, only the highest grade, produced for this special purpose, should be used. Long life then is possible, even under severe conditions. Rubber should not be loaded more heavily than 50 lb. per sq. in. It is very incompressible, and yields almost entirely through shear. In designing rubber supports, space for elastic flow must be maintained, otherwise they may be too rigid (see Fig. 22). Rubber pads usually should be of small area so they can freely squeeze out sideways under load. For a given thickness the hardness of a rubber pad increases disproportionately under compression with increase of area (see Fig. 23.) Rubber must not be used at temperatures of, say, over 120° F. and should be protected from oil. For a discussion of the properties of rubber, and of its design constants see p. 5–27.

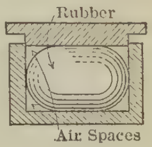


Fig. 22. Rubber Support

**Cork** differs from rubber in that it is directly compressible and space for elastic flow is not needed. See p. 5–25. The stiffness of cork increases directly as its area when used in large pads. Cork pads of comparatively large area thus may be effective in isolation, particularly for higher frequencies in the sound range. Cork has the disadvantage of gradually compressing under load after initial loading, resulting in gradual loss of isolating power for lower frequencies. Despite this fact cork has performed successfully for many years in some installations.

**Gelatin** has elastic properties similar to rubber, but has a lower elastic constant, and thus a lower stiffness. In design, the same considerations enter as for rubber. In its usual form it should not be loaded over 25 lb. per sq. in. It resists oil, but not water. It will fail if heated to 212° F.

**SUSPENSION FREQUENCY CALCULATIONS** for up-and-down frequencies are in all cases made by formula [5], but for rubber, cork and gelatin actual frequency is higher than calculated, due to imperfect elasticity. For some materials, gravity compression $d$ must be greater than required by [5]. For cork, to obtain required suspension frequency, it may be 2 or 3 times as great as for perfectly elastic material like steel.

**REMARKS ON APPLICATIONS.**—General application of elastic suspensions requires considerable knowledge of vibration. Elastic suspensions usually have several modes of vibration. Exciting frequencies, other than the one being isolated, probably will be present, *e.g.*, revolution frequency of a rotating machine operating over a wide range. A suspension must be so designed that all its vibration frequencies avoid all exciting frequencies. An otherwise good elastic suspension may have an objectionable low-frequency quiver. A suspension, flexible in some directions, often must be rigid against belt pull, in which case rubber is better than steel springs. Steel springs sufficiently flexible in one direction may be too flexible in others. Flexible mountings will not successfully operate on a floor that is both light and flexible, *e.g.*, spring suspended machinery on spring supported car bodies, or airplane engines on the relatively light structure. These compound elastic systems with more than one mass and elastic connection, are not readily solved by analysis. Simple cases of suspensions on floors having both elasticity and inertia sometimes can be successfully treated. See Kimball, Vibration Prevention in Engineering, pp. 96–107.

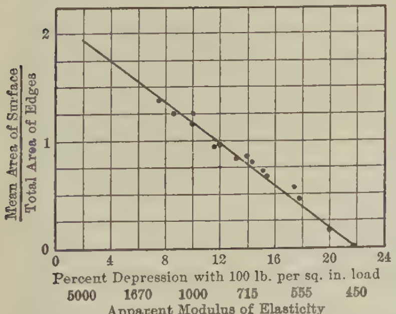


Fig. 23. Apparent Modulus of Compression as a Function of Shape. Tests on ½-in. rubber. Smallest dimension greater than twice the thickness

**FLOOR SUSPENSIONS.**—Some machinery can be isolated by mounting on a comparatively flexible floor frame, elastically supported on the solid floor below. This prevents the transmission of rumbles, grinding noises and hums through the building, to radiate sound from panels, etc. Springs or rubber pad suspensions used at intervals of about 1 ft., depending on the floor load, will perform well, steel being more permanent. The flooring should yield perceptibly, say 0.1 in. under machinery load. A difficulty is the presence of possible side swings and low-frequency quivers, which best are cured by rearranging supports, rather than by calculation. Printing press vibration, for example, has been so isolated. Machinery requiring exact alignment, as an accurate lathe, cannot be placed directly on a flexible floor, but must be set on a rigid, spring-supported platform. Such work is best done by concerns specializing in construction of elastic floorings.

If a cork layer is used instead of a spring-supported flexible floor, low-frequency rumbling noises may be transmitted, if not the higher frequency notes, unless a thick enough layer is used to permit appreciable compression ($^1/_{16}$ to $^1/_8$ in.), under load.

**MOTOR SUSPENSIONS.**—**Single Phase Electric Motors** are subject to torque pulsation of twice the A.C. supply frequency, *i.e.*, 120 cycles per sec. for a 60-cycle source, causing a 120-cycle torsional forced vibration of both stator and rotor about the motor axis. This 120-cycle vibration causes the hum noticed in domestic use of such motors, which a well-designed elastic suspension will completely eliminate. To isolate such 120-cycle vibrations, a suspension of torsion flexibility is required, whose natural frequency $f_0$ about the motor axis is, say, 20 cycles per sec.; since $f = 120$, $f/f_0 = 6$, and from [19], $\epsilon = {}^1/_{35}$, giving fair isolation. This support must have considerable transverse rigidity to resist belt pull, and also to prevent resonating to the speed frequency due to slight mechanical unbalance. Fig. 24 shows an elastic suspension using rubber which is satisfactory.

**Polyphase Electric Motors** with good elastic suspension can be used instead of direct-current motors where noiseless operation is required. While subject to much less vibration than single-phase, no polyphase motor is free from electrical hums which can be transmitted to surrounding structures. The motor may be isolated by mounting on a heavy concrete block supported on a layer of cork, or on cork blocks at isolated points, sufficiently thick to cushion the mass. This construction will be fairly rigid against belt pull. If the motor is placed on the cork footing, without the concrete block, noise probably will be transmitted. Another method is to set the motor on a small area of flexible flooring and avoid low-frequency resonances in the operating range (see Floor Suspensions, above). A third method is to use specially designed elastic bases furnished by the maker. For polyphase motors, the design of such a base is not easy, but correctly designed with rubber pad suspensions, performance is excellent, cost is low, and less space is required than by other methods.

**AUTOMOBILE ENGINE SUSPENSION.**—Flexible supports of comparatively low resonant frequency are being used (1936) on many types of automobile engines with apparent success. Avoidance of suspension resonances is difficult with a wide operating speed range and consequent wide range of exciting frequency. The principal resonant frequency is low and is easily passed through in accelerating the engine. At normal operating speeds, isolation is excellent, and engine vibration is hardly perceptible in the car.

**AIRPLANE VIBRATION.**—A serious difficulty in isolating airplane engine vibration is the relative lightness of the airplane structure, compared with engine weight. A support flexible enough for complete isolation lacks necessary strength. For torsional oscillations a rubber type support is possible, of sufficient torsional flexibility for good isolation and enough transverse rigidity and strength to be safe. This field requires further study.

**ISOLATION FROM VIBRATION.**—The isolation of certain bodies from forced vibrations and waves of outside origin sometimes is necessary. This problem is the reciprocal of that of vibration elimination through tuning (see p. 16-09), and the same principles and formulas apply.

**ISOLATION OF DELICATE APPARATUS** from building tremors requires an idea of the approximate frequencies present, obtained by observing quiver of the surface of a dish of mercury. Building and ground tremors (from vehicles, etc.) contain low-frequency harmonics which can be isolated only by a suspension of frequency of less than one cycle per sec. This follows from transmissibility formula [19], where $f$ is frequency of the building tremor and $f_0$ that of the suspension. Evidently both the up and down oscillating frequency and the sidewise swing of the suspension must be very low. The instrument to be isolated should be hung by long helical springs which extend several inches under the load. See formula [5]. Since a seconds pendulum is about 3 ft. long, suspension springs and wires must be at least several feet long to insure a low enough frequency of sidewise swing.

Several pieces of apparatus may be isolated together on a relatively massive platform hung either from the ceiling or a davit, using springs extending several inches under the load.

Good isolation sometimes can be obtained by placing the apparatus on a heavy metal plate resting on an inflated automobile inner tube. This is not sufficient for very low-frequency disturbances, because to screen from the latter the suspension must yield several inches under load.

All helical springs used in suspensions should be loosely stuffed or wrapped with damping material to prevent local spring resonant oscillations. It is important not to use too much damping as in a spring suspension it *increases* the transmissibility.

**VEHICLE SUSPENSION.**—A successful vehicle suspension should be as flexible or of as low frequency as is consistent with stability. It must have some damping. The lower the frequency the better the isolation from road impulses. Damping is essential in vehicle suspensions, to limit unavoidable resonant bouncing oscillations due to the great range of exciting frequencies produced by various road conditions and speeds.

In railroad cars, the use of built-up leaf springs provides damping by friction between individual leaves. In automobiles, leaf spring friction is not sufficient. So-called shock absorbers are necessary, whose function is to reduce transmissibility at or near suspension resonance points. For other exciting frequencies, transmissibility is slightly increased by such devices.

Automobile suspensions must isolate against more severe vibration excitation than railroad car suspensions. They are of compound character, with one elastic support between wheel and road and another between wheel and car body. Fig. 25 schematically represents such system with damping omitted; $m_1$ represents car body mass; $k_1$, car spring elastic constant; $m_2$, wheel mass; $k_2$, tire elastic constant. Actually the system is more complex, but the scheme of Fig. 25 yields the two principal modes of vibration of ordinary automobile suspensions.

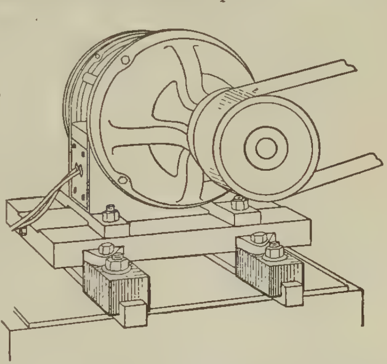


Fig. 24. Elastic Motor Suspension Using Rubber

Due to its compound character, transmissibility is given by an expression which contains both of its two resonant frequencies $f_1$ and $f_2$ as well as exciting frequency $f$ from the road. Where $y_1$ and $y_3$ = amplitudes, this expression is

$$\text{Transmissibility } \epsilon = \frac{y_1}{y_3} = \frac{1}{\{(f^2 - f_1^2)/f_1^2\}\{(f^2 - f_2^2)/f_2^2\}}, \quad . \quad . \quad . \quad [20]$$

The higher resonant frequency $f_1$ is excited by "washboard" roads which causes the wheels to resonate up and down between the road and stops on the vehicle body. This condition arises when $f = f_1$, and without shock absorbers would be excessive. The lower resonance occurs when $f = f_2$, where the whole car body swings up and down in step with long waves in the road. At this frequency shock absorbers again act. This compound system reduces to a simple one, with transmissibility given by [5], when a rigid connection replaces $k_1$. In this case, $f_1$ becomes so large, compared with $f$, that the first factor $(f^2 - f_1^2)/f_1^2$ can be replaced by unity.

This discussion brings out the principal factors in vehicle suspensions. Frequencies $f_1$ and $f_2$ should be as low as possible, so that $\epsilon$ is as small as consistent with stability between resonant points. At the same time, $\epsilon$ is limited at these points by shock absorbers which are not so stiff that $\epsilon$ is appreciably increased between resonant points. For derivation of [20], see Kimball, Vibration Prevention in Engineering, pp. 107–109.

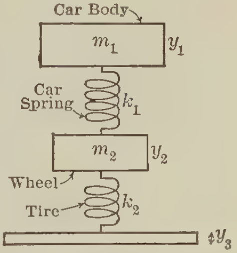


Fig. 25. Diagram of Vehicle Suspension

## 6. VIBRATION DAMPING

Vibration damping is reduction of vibration by friction. Neither avoidance of resonance by tuning nor prevention of transmission by spring suspensions are damping processes. Vibration damping is of maximum effect in curing resonant vibrations where the rate of energy supply exciting vibration is relatively small.

**SLIDING FRICTION DAMPING** exists in built-up leaf springs whose leaves slip against each other, creating friction when the spring oscillates. This friction is important in limiting resonant vibrations of railroad cars on their springs. Sliding friction damping is used in some forms of automobile snubbers.

Torsional oscillations in shafts may be limited by a heavy fly-wheel, attached to the shaft by sliding friction, which slips against oscillations of the shaft. Critical speed vibrations of shafts can be eliminated by spring-supporting bearings in a collar which permits slipping against a friction clamp. Sliding friction dampers often give trouble from variable friction due to variation of pressure of the clamps, to scoring, or to presence of foreign materials like oil.

Sliding friction in the joints is important in limiting vibration of steel structures because of the extremely low internal friction of steel. Turbine wheels with separately-made buckets have far more resistance to vibration than those with buckets integral with the disc.

**IMPACT DAMPING** arises from the impact of loose parts of a structure in an effort to vibrate independently. Such impact has a strong dissipating effect and rapidly damps vibration. A steel gear which otherwise would ring under a hammer blow will sound quite dead if a steel ring is snapped into a slot under the rim on each side. Impact damping is effective in dissipating high-frequency vibrations of small amplitude.

**FLUID DAMPING** differs from sliding friction damping in that in the former, the resisting force rapidly increases with the velocity of motion; in the latter, resisting force is independent of velocity. Dash pots utilizing fluid flow have powerful damping action for large amplitudes, which rapidly falls off with slow motion at small amplitudes.

**SOLID DAMPING.**—All solid materials have more or less internal friction damping resistance to periodic deflection. The most perfect tuning fork loses its vibration amplitude from this cause when all external friction effects, as air resistance, have been removed. Softer elastic materials, as rubber, have considerable internal friction. Steel has so little solid damping that structures built up of steel members owe nearly all of their vibration resistance, as a whole, to sliding and impact friction joints. Individual members and panels, however, may be damped largely through what internal friction exists. Individual members in a steel structure may be very resonant.

Concrete has ten times as much damping as steel, and concrete structures are not particularly resonant. The overall damping in a steel structure does not differ greatly from that of a similar concrete structure, due to dissipation in friction and vibration interference between individual members.

# NOISE

## 1. NOISE IN GENERAL

When vibrations of bodies are transmitted into the air as sound waves of different frequencies, the heterogeneous sound resulting is noise. Noise reduction has (1936) become important in engineering.

**VELOCITY OF SOUND** is a wave phenomenon in the atmosphere. It has constant velocity $V$ of transmission, regardless of its frequency or wave length, of about 1100 ft. per sec.

$$V = \sqrt{1.4\,p\,g/w}, \quad \ldots \ldots \quad [1]$$

where $V$ = velocity, in. per sec.; $p$ = atmospheric pressure, lb. per sq. in.; $g$ = acceleration of gravity, in. per sec. per sec. = 386; $w$ = weight of 1 cu. in. of air. Since $w$ varies as $p$, $V$ is unaltered by pressure change. When temperature increases, however, $w$ alone decreases. Thus while not altered by pressure change, transmission velocity is proportional to the square root of absolute temperature, and varies about 5% in ordinary temperature ranges.

**PERCEPTION OF SOUND.**—The human ear can perceive as tones, noise frequencies from 100 to over 16,000 cycles per sec. In its most sensitive range, 500 to 6000 cycles per sec. the ear is extraordinarily sensitive, the wave amplitude of easily detected sounds being of molecular dimensions. In this range, amplitudes of a fraction of a millimeter are heard as terrific noises. Sounds in this range are easily excited and difficult to eliminate.

**FREQUENCY AND WAVE LENGTH.**—The frequency of a sound is the number of complete waves transmitted per second. High frequencies correspond to short wave lengths and *vice versa*.

$$\text{Wave length} = \lambda = V/f, \quad \ldots \ldots \quad [2]$$

where, if $V$ = in. per sec., and $f$ = cycles per sec., $\lambda$ is in inches. Table 1 compares frequencies and wave lengths in the audible range.

**Table 1.—Sound Frequencies and Wave Lengths**

| Frequency $f$, cycles per sec | 100 | 200 | 500 | 1000 | 2000 | 4000 | 6000 | 10,000 |
|---|---|---|---|---|---|---|---|---|
| Wave length, in | 132 | 66 | 26.4 | 13.2 | 6.6 | 3.3 | 2.2 | 1.3 |

A heterogeneous sound may be made up of wave lengths from this entire range, superimposed on each other, the smaller travelling over the larger. Individual frequencies are called components. In noise work, sound should be considered in terms of its wave length.

**SOUND VELOCITY IN SOLIDS** is much higher than in air. In steel, it is about 17,000 ft. per sec.; in other building materials it is less, but much greater than in air.

**FREQUENCY AND CHARACTER OF SOUND.**—Rumbles and boomings are of long wave length. Ordinary whistling sounds are of intermediate wave length. Tinkling noises and rattling of paper are of higher frequency. Hissing sounds are complex sounds of very high frequency. The whirring of a fan is a very complex noise in which, for small fans, higher frequencies predominate; for large fans and blowers a considerable proportion of low-frequency harmonics are present.

## 2. NOISE MEASUREMENT

**SOUND LEVEL METERS** afford the best means of measuring noise. No short-cut method or cheap type of meter will be satisfactory. The energy associated with sound is very small and the ear itself is such a wonderful detector that high-grade microphones with vacuum tube amplification are essential. The author has obtained excellent results with a General Electric type of meter, which will measure either sound as a whole or, by use of a separate analyzer, measure intensity of the different components present. The meter reads sound level directly in decibels.

**Tuning Forks.**—Much time can be saved in a frequency determination by using a set of tuning forks and noting beats. To determine, for example, whether or not a whistle from a gear is tooth frequency noise, the note frequency is compared with the known frequency of gear tooth impacts.

**Stethoscopes.**—A physician's stethoscope, with a sharp point about 1 in. long fixed to the center of the diaphragm, placed against a vibrating structure is a very sensitive detector of noise transmission which otherwise cannot be heard. By tilting the point in different directions the approximate direction of maximum amplitude can be detected. This instrument can locate node and loop areas on a sounding body of small vibration amplitude.

A steel diaphragm and point near alternating-current electrical apparatus will give a false indication. Disc and point and holder should be of phosphor-bronze or other non-ferrous metals.

**Electro-magnetic Pick-up** based on the phonograph needle principle sometimes is used. Such a device is made for use with the General Electric sound level meter, and very small vibration amplitudes, associated with sound production, can be directly read.

**SOUND LEVEL UNITS. DECIBELS.**—The sound level unit almost universally adopted in commercial work is the *decibel* (db.). Table 2 shows the meaning of the decibel scale in terms of: *a.* Every-day sounds; *b.* Vibrational energy in sound waves; *c.* Amplitude of motion of air particles in a 100- and a 1000-cycle wave; *d.* Rise and fall of pressure in sound waves. The type of noise scale in Table 2 is so constructed that every

**Table 2.—Decibel Scale in Various Units**

| Decibels | Every-day Sounds | Sound Intensity (Threshold = 15 db.) | Half Amplitude of Wave Motion, cm. Cycles 100 | 1000 | Rise and fall of pressure in 1000-cycle wave, dynes per sq. cm. |
|---|---|---|---|---|---|
| 15 | Very low | 1 | ........ | ........ | $10^{-3}$ |
| 25 | Rustle of leaves | 10 | $10^{-7}$ | $10^{-8}$ | .... |
| 35 | Whisper | $10^2$ | ........ | ........ | $10^{-2}$ |
| 45 | Quiet residence | $10^3$ | $10^{-6}$ | $10^{-7}$ | .... |
| 55 | Average residence | $10^4$ | ........ | ........ | $10^{-1}$ |
| 65 | Average business office | $10^5$ | $10^{-5}$ | $10^{-6}$ | .... |
| 75 | Passenger automobile | $10^6$ | ........ | ........ | 1 |
| 85 | 15 to 50 ft | $10^7$ | $10^{-4}$ | $10^{-5}$ | .... |
| 95 | Motor trucks, 15 to 20 ft | $10^8$ | ........ | ........ | 10 |
| 105 | Heavy street traffic in city | $10^9$ | $10^{-3}$ | $10^{-4}$ | .... |
| 115 | Subway express passing through | $10^{10}$ | ........ | ........ | 20 |

Table 3.—Comparison of Noise Values from Several Sources.

| Source 1 | Source 2 | Source 3 | All sources together. Intensity of Source 1 × | Source 1 | Source 2 | Source 3 | All sources together. Intensity of Source 1 × |
|---|---|---|---|---|---|---|---|
| 35 db. | 35 db. | 0 db. | 2 = 38 db. | 40 db. | 40 db. | 0 db. | 2 = about 43 db. |
| 35 " | 35 " | 35 " | 3 = about 40 db. | 40 " | 40 " | 43 " | 4 = " 46 " |
| 35 " | 35 " | 38 " | 4 = " 41 " | | | | |

increase of 10 db. corresponds to 10 times the noise intensity as shown in the third column. Thus a very loud noise of 115 db. is ten billion times as energetic as a very low sound in perfectly quiet surroundings. This scale covers the whole audible sound range in 100 units, and distinguishes noises more nearly as the ear actually does than would a scale based upon energy or pressure units, as given in columns 3 and 6 of Table 2.

For practical purposes it is useful to remember that every 3 db. increase in noise means that the intensity almost exactly doubles. If one noise is 12 db. greater than another, the first noise intensity must be doubled four times, *i.e.*, multiplied by 16, to equal that of the second in intensity, since $^{12}/_3 = 4$. Two noises of equal intensity placed side by side give twice the intensity, or a total noise 3 db. greater than either of them by itself, regardless of their separate noise levels. Examples are shown in Table 3.

**MATHEMATICAL DEFINITION.**—The difference in decibels between two noises of intensities $I_1$ and $I_2$ is defined by the mathematical relation,

$$\text{Difference in db.} = 10 \log_{10} (I_2/I_1); \quad \ldots \ldots \quad [3]$$

$$= 10 \log_{10} (p_1^2/p_2^2) = 20 \log_{10} (p_1/p_2), \quad \ldots \ldots \quad [4]$$

where $p_1$ and $p_2$ are sound pressure amplitudes corresponding to $I_1$ and $I_2$. Expression [4] is multiplied by twice as large a constant as that of [3] because sound intensity is proportional to the square of pressure amplitude of sound waves.

Expressions like [3] and [4] involve only pressure and intensity ratios, and give db. differences. It is thus necessary to define a zero point to fix scale values. Zero decibels loudness level corresponds to sound produced by a 1000-cycle tone with a pressure amplitude of 0.0002 dyne per sq. cm. This sound is just audible in a perfectly quiet room. On such a scale sounds below audibility give negative db. values. Table 3 is derived from these equations, as are the various relations brought out in the discussion of sound level units above.

## 3. NOISE PREVENTION

A noise that cannot be prevented or greatly reduced at the source, by design or by damping, will radiate into the air either directly from the surface of the vibrating body or from parts of the structure to which it is attached. Such noise radiation is *indirect noise*. See Fig. 1, where part of the motor noises radiate directly from the motor and part from panels of the room below, which may resonate to the frequency transmitted.

The following classification is useful when applying a preventive for noise:

| Classification | Method | | |
|---|---|---|---|
| Prevention at Source | Design Changes<br>Damping | | |
| Not readily preventable at source | Direct noise | Screening<br>Absorption<br>Filtration | Higher frequency range, 300 cycles and up. |
| | Indirect noise | Isolation by suspension | Lower frequency range, under 300 cycles. |

A surface vibrating at small amplitude will radiate high frequencies directly into the air much more strongly than low. The latter require much larger vibrating surface area for direct radiation. Hence, in general, direct noises are of higher frequency than indirect; the 300-cycle limit in the classification is an average value.

### Prevention at Source

**DESIGN CHANGES** involve the field of noisy apparatus. This is too large to be considered here, other than to cite a few instances. Remedies must come from studies by manufacturers.

*a.* **Car Wheel Noise.**—Special car wheels with steel rims isolated from the rest of the wheel by rubber are being made (1936) and tested by rubber companies and car wheel manufacturers. Such wheels have greatly reduced track noise of cars.

*b.* **Electric Motors.**—Special studies have produced designs with much lower alternating current electrical hums.

*c.* **Blowers.**—Whistles from blowers in ventilating ducts usually are blade frequency noises, that can be eliminated at the source by design changes.

*d.* **Gear Whistles.**—High-speed gearing can be greatly improved by accurate design to avoid impacts, and by using helical instead of spur gears.

**DAMPING.**—A sounding body, like a gear casing or small blower casing, can be quieted by covering it with a suitable damping material which has the ability to reduce vibrations.

*a.* **Putty Lagging.**—A layer of non-drying putty 1 in. or more thick, covered with varnished cambric or similar material to prevent gradual drying of the putty, sometimes will greatly reduce noise. The putty exerts a powerful damping action, especially on high-frequency whistles and ringing noises, by reducing amplitude of case vibration. Lower frequency rumbling noises are less strongly damped. The higher the frequency, the thinner may be the putty. Suitable non-drying putty can be made by any good chemist. It should have fairly high density and a binder to prevent it falling apart. Before application, the surface should be rubbed with a small amount of putty to improve adhesion.

*b.* **Other Types of Lagging.**—Felt-like materials, when applied as lagging, have far less ability than putty to damp vibrations at the source, though they are sometimes effective as noise reducers. This is due to their high sound absorption coefficients reducing the ability of a surface to radiate noise, and not because they reduce the amplitude of vibration. While these materials are easier to apply, putty is more effective in such applications than any other material used by the author. This is due to its power actually to reduce the surface vibration amplitude.

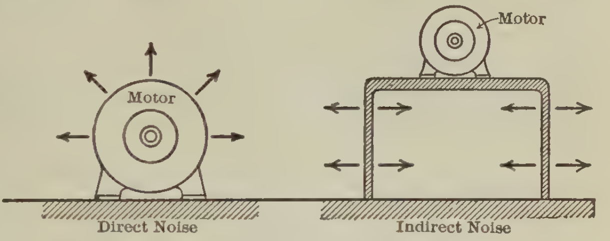


Fig. 1. Direct and Indirect Noise

Resinous and tar-like materials when applied as lagging offer great friction to slow vibratory deflections of large amplitude. They are of relatively little use in damping high frequency vibrations of small amplitude, aside from the increase in surface mass that they produce. These vibrations are so rapid that flow, with its attendant friction, does not occur. The relative vibration absorption ability of different laggings is tested by coating the material on a steel diaphragm welded to a short pipe to support its edges. If the diaphragm be resonated by an electromagnet of variable frequency it emits a loud sound unless the lagging is an effective damper. Impacting with a hammer is a good test if an electro-magnet is not available.

The correct use of acoustic felts is the absorption of sound *after* sound waves are on the air. In this application putty is of small value, compared to felt and similar materials.

## Screening and Transmission

**SCREENING OF SOUND.**—Noise is screened either to confine it in a given space, or to maintain a given space noise-free. In either case the principle used is the same, *i.e.*, to construct screens or walls highly resistant to noise transmission. The problem thus is one of the noise transmission.

**PROCESS OF TRANSMISSION.**—Sound is transmitted through a wall by air pressure oscillations produced on its farther side by leakage, or more commonly, by oscillations produced by vibrations of the wall itself. This latter phenomenon usually occurs regardless of the presence of the former. Porous, felt-like materials transmit largely through leakage of wave oscillations.

Ordinary wood and plaster walls transmit by motion of the wall itself, or by vibration of portions of the wall. This is evident from Table 1 showing wave lengths of various sound frequencies. These are much in excess of the thickness of most walls, except those of high frequency, which are not readily transmitted. Sound waves of a given frequency in wall materials must be several times longer than in air, in proportion to the higher sound transmission velocity of these materials. Only the highest frequencies in the audible range can exist as complete individual waves during transmission through ordinary wall.

**Leakage.**—Small holes or cracks in a wall, otherwise highly resistant to noise transmission, will pass a surprising amount of sound, unless the crack is deep. Considerable damping action then is introduced by air friction on the crack sides. A sound-proof room or box must have tight fitting doors.

**MEASUREMENT OF SOUND TRANSMISSION** involves resonant effects depending upon mass, elastic characteristics and structure of the partition as determined by its dimensions, and reflection effects influencing the observations of the transmitted sound.

In engineering, a sound meter of the type described on p. 16–19 will indicate the intimate effect of the acoustics of the immediate surroundings. For example, the noise level in an ordinary room may fluctuate widely from place to place because of systems of standing sound waves which may be present. The ear will detect nodes and loops when the head is swinging back and forth, if one or two fairly strong tones are present in a room.

**FACTORS INFLUENCING TRANSMISSION.**—Noise is transmitted primarily by vibratory motion of a wall as a whole. Important factors in transmission prevention are mass per unit area and stiffness.

Stiffness is a trouble rather than an aid in the transmission resistance of walls, unless the wall is thick, of relatively small area, and unable to resonate to any objectionable noise harmonics. A small rigid metal case may resist noise transmission largely due to rigidity rather than to mass per unit area.

Ordinary walls have various flexural resonant frequencies well below some objectionable noise harmonics. Stiffness can never aid such cases. It slightly increases wall vibration amplitude, but this effect is negligible for relatively high frequencies.

A third factor whose influence is important is wall structure. A double wall, with air space unbridged by rigid members, may resist noise transmission far more than a single wall of the *same total weight* per sq. ft.

**DESIGN OF WALLS OF LOW TRANSMISSIBILITY.**—A double wall is more desirable than a single wall, being cheaper and less massive for the same transmission loss. A comparatively rigid skeleton of ordinary 2 × 4-in. studding should be used, with inner and outer walls hung from it by spring connections. These walls, of celotex or similar material, should weigh from 1 to 5 lb. per sq. ft. A thin layer of plaster-like reinforcement is helpful. Besides adding mass, it decreases porosity, which is an advantage. The celotex may be reinforced by thin sheet iron. Sheet iron must not be used without celotex, as it is subject to resonances.

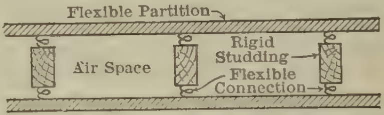


Fig. 2. Wall Construction of Low Transmissibility

Spring connectors eliminate the bridging effect of studding, which results when the walls are directly nailed to it. Steel spring connectors should be set on felt pads to reduce possible high-frequency vibration transmission through the springs. Such a wall has an air space of about 5 in., which is fair spacing. Air spacings of the order of 1 in. are much less effective, while more than 6 in. spacing usually is unnecessary. See Fig. 2. If sound resistance required is not high, celotex may be nailed directly to studding on both sides, but the advantage of the air space, which is large, is then much reduced.

A single wall should be of brick or masonry to have enough mass to be effective. A heavy masonry wall has high transmission resistance, but is expensive and cumbersome. This type of wall is not particularly susceptible to resonances, its transmission resistance being due almost entirely to its mass.

Double walls of comparatively small mass, as recommended above, should have some damping, such as contained in a material like celotex, to avoid resonances.

**TRANSMISSION LOSS.**—The aim of a sound-proof wall or partition is to have sufficient transmission loss to be effective. This increases as the frequency rises. Table 4 gives an average transmission loss over the audible frequency range. It is about the same order as that for a pure 500-cycle sound. Transmission loss is measured by the observed drop in sound intensity in decibels produced by the wall, and is independent of the intensity of sound being screened. A wall causing a drop in noise level from 64 to 34 db. will also cause a drop from 54 to 24 db., *i.e.*, transmission loss is 30 db. The author has no figures at hand for a double wall of the type first recommended, but the transmission loss is high, from 40 to 60 db.

**PRECAUTIONS.**—Several precautions are necessary in the design of transmission resisting walls and screens, after the design of the wall itself has been decided.

**Transmission from Footing.**—Studding should be set on a layer of yielding material to prevent transmission to wall from ground or other support. A layer of felt under the sill is good, but by itself becomes rather hard. By the principles of vibration isolation (see p. 16–14), a gravity elastic deflection of $^1/_8$ in. or more is desirable. Cork is rather hard, but if sufficiently thick should prevent transmission of higher frequencies.

**Transmission through Pipes and Ducts.**—Metallic pipes are effective transmitters

Table 4.—Transmission Loss Produced by Various Partitions.*

| No. | Description | Transmission Loss, db. | Lb. per sq. ft. |
|---|---|---|---|
| 1. | Steel, 1/4 in. thick | 30.0 | .... |
| 2. | Solid oak, 1 3/4 in. thick | 20.9 | .... |
| 3. | Plate glass, single pane, 1/4 in. thick | 22.8 | .... |
| 4. | Plate glass, 2 panes, 3/16 in. thick | 19.7 | .... |
| 5. | Hair felt, 3 in. thick | 11.8 | .... |
| 6. | 3 layers 1-in. felt alternated with 4 layers building paper | 27.9 | .... |
| 7. | 2-in. gypsum tile, unplastered | 20.2 | 10.4 |
| 8. | 3-in. gypsum tile, unplastered | 23.7 | 14.2 |
| 9. | 4-in. clay tile, unplastered | 28.8 | 17.0 |
| 10. | 4-in. clay tile, 1/2-in. plaster | 30.8 | 22.0 |
| 11. | 8-in. brick wall, 1-in. plaster | 46.1 | 88.0 |
| 12. | Double 2-in. solid gypsum tile, unplastered unbridged, 4-in. separation | 51.8 | 20.4 |
| 13. | Same as 12, but bridged at top and bottom | 46.9 | 20.4 |
| 14. | Same as 12, inner faces lined with 1-in. felt | 57.3 | 22.3 |
| 15. | Staggered 2 × 4 in. wood studs, metal lath. 1/2-in. gypsum plaster | 39.1 | 19.8 |
| 16. | Same as 15, with 1/2-in. celotex instead of metal lath | 36.6 | 13.0 |

* Taken from work done under direction of Dr. Paul Sabine at Riverbank Acoustical Laboratory. See P. E. Sabine, Sound Transmission through Walls, *Jour.* Acoustical Soc., June, 1930, pp. 181–207. Also Sabine, Acoustics and Architecture, chap. xii. (McGraw-Hill.)

of sound. Two or three pipes in a well-designed sound screen may greatly reduce its effectiveness. Transmission along a metallic pipe can be greatly reduced by rubber joints to prevent metallic contact of pipe sections. For air ducts, heavy canvas connections are effective. Direct transmission of sound through ventilating ducts may be considerable. See p. 16–24.

**Edges of Double Walls.**—The sound-insulating properties of elastically suspended wall partitions used with double walls requires edges to be free from rigid contacts with the frame, at the same time avoiding cracks through which sound can pass. Cracks can be stuffed with felt or similar material.

## 4. ABSORPTION OF SOUND

If directly radiated noise, as the noise in a business office or in a factory, cannot be screened, a decided reduction in the noise level can be made by means of sound absorbing materials. Sound absorption takes place during wave reflection. It is measured by the fraction of the incident energy which disappears in a single reflection. An open window space reflects practically no sound and represents a maximum obtainable reduction.

Table 5, by Watson, gives sound absorption coefficients of various materials based on unity as absorption of an open window. See F. R. Watson, Absorption of Sound by Materials, Univ. of Ill. *Bull.* 172, 1927. Also P. E. Sabine, Acoustics and Architecture. Felt-like materials have high sound absorbing power, as have certain porous materials. A good absorbent covering has a smooth surface layer of oilcloth or thin metal perforated with many holes, under which is a porous material. Such covering gathers less dust than a felt-like surface and is just as effective.

Table 5.—Sound Absorption Coefficients of Different Materials at 512 Cycles per sec. Pitch

| Description | Coefficient | Description | Coefficient |
|---|---|---|---|
| Open window | 1.0 | Carpet | 0.20 |
| Brick wall | 0.032 | Curtains, heavy folds | 0.40–0.75 |
| Brick wall, painted | 0.017 | Hairfelt, 1 in. thick | 0.59 |
| Wood, planed | 0.06 | Rock wool | 0.57 |
| Wood, varnished | 0.03 | Balsam wool, 1 in. thick | 0.61 |
| Plaster on lath | 0.034 | Good acousti-celotex, 441 small holes per sq. ft | 0.70 |
| Glass (single thickness) | 0.027 | | |
| Marble | 0.01 | | |

Sound absorption and transmission resistance are entirely different processes. A good sound absorber, as felt, is a poor transmission preventive. Such a material may be effectively used, however, as a lining of an enclosed space to further reduce the final noise level obtained by screening. If no sound resisting walls were used with the absorbing layer, however, the noise level would be very much raised.

**CALCULATION OF NOISE REDUCTION THROUGH ABSORPTION.**—When the noise level in a room and the sound absorption coefficients of its walls (Table 5) are known, the possible reduction in noise level through the use of more absorbent materials on the walls can be approximately calculated. From [4],

$$\text{db.} = 10 \log_{10} (a_2/a_1) \quad \ldots\ldots\ldots \quad [5]$$

To find $a_1$, multiply wall area of each part of room by its absorption coefficient. If $S$ = total area, and if the coefficient of $^1/_6\, S = 0.03$, of $^1/_3\, S = 0.01$, and of $^1/_2\, S = 0.02$, then $a_1 = \{(S/6) \times 0.03\} + \{(S/3) \times 0.01\} + \{(S/2) \times 0.02\} = (S/6) \times 0.11$. Assume that the part of the surface whose coefficient is 0.02 to be covered with an absorbent with a coefficient of 0.50. Then

$$a_2 = \{(S/6) \times 0.03\} + \{(S/3) \times 0.01\} + \{(S/2) \times 0.50\} = (S/6) \times 1.55,$$

and $a_2/a_1 = 155/11 = 14.1$. From [5], db. $= 10 \log 14.1 = 10 \times 1.492 = 11.49$. If all the walls and the floor were covered with such a material, $a_2 = (S/6) \times 3.00$; $a_2/a_1 = 300/11 = 29.2$, and db. $= 10 \log 29.2 = 10 \times 1.4654 = 14.65$. Thus covering half the whole area would reduce the noise level by 11.49 db., but covering the whole area would reduce it only a little over 3 db. more.

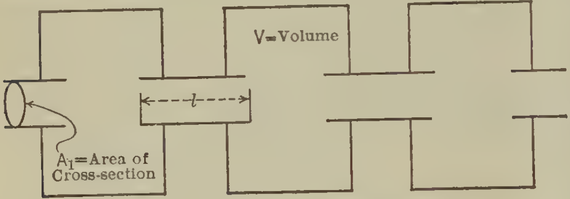


FIG. 3. Low-pass Acoustic Filter

The conclusion is that in reducing noise level by sound absorbing materials on the walls, the greatest reduction is effected by the first part of the walls covered. After this, less and less improvement is obtained from a given area of absorption covering. It usually is sufficient to treat only part of the wall area of a noisy room. Sound absorbing materials should be so used in interiors that they may reduce the noise by many back and forth reflections.

Table 5 is based upon absorption coefficients at 512 cycles per sec. Higher frequencies are absorbed more and lower frequencies less. The use of the correct amount of absorption on the interior walls of lecture and music halls is an important field of acoustic engineering, known as architectural acoustics. See P. E. Sabine, Acoustics and Architecture, chap. viii.

**NOISE ABSORPTION IN DUCTS.**—Noise in ventilating ducts can be greatly reduced by linings of acoustic absorbent materials. This is particularly true of the higher-frequency rushing noises and hisses. Hums and whistles are not so easily eliminated, and require other means of treatment. See Acoustic Filters, below.

## 5. FILTRATION OF SOUND

**ACOUSTIC FILTERS IN GENERAL.**—When direct noise, containing definite notes or harmonics, is transmitted along ducts or through openings, it often is possible to completely eliminate objectionable harmonics by an acoustic filter, which obstructs but little air flow or ventilation. Sound filters are of two classes: 1. Low-pass filters which transmit low frequencies, but filter out the higher ones; 2. High-pass filters which do the opposite. A low-pass filter prevents transmission of the relatively high whistle often present in a ventilating duct. Low frequencies, predominant in explosive discharges, are filtered out by a high-pass filter, as an automobile muffler or Maxim silencer.

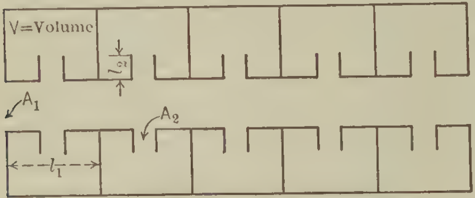


FIG. 4. Type 2, Low-pass Acoustic Filter

Space is the usual limitation in filter applications. This restricts for low-pass filters less than high-pass. Low-pass filters return sound energy to the source, where it dissipates itself through reflections, etc. High-pass filters must themselves dissipate the sound, this being an important reason for their comparative bulk. The individual stages of a filter must be considerably shorter than the wave length of sound being filtered, each stage producing a definite amount of attenuation, as illustrated by the specific types of design which follow.

**LOW-PASS FILTERS** may be divided into two classes of design. Fig. 3 shows the first, and Fig. 4 the second. In designing a filter, the lower limit of frequency $f_0$ is determined, below which it will transmit, and above which the filtering action begins.

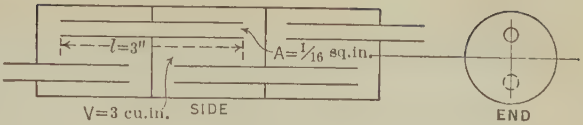


Fig. 5. Special Filter Design

For Type 1 of Fig. 3, $$f_0 = 4200\sqrt{A_1/(lV)} \qquad [6]$$
where $A_1$ = area of cross-section, sq. in.; $l$ = length, in., $V$ = volume, cu. in. and where the symbols refer to Fig. 3.

For Type 2, Fig. 4, $$f_0 = 4200\sqrt{\frac{1}{V\{(l_1/A_1)+4(l_2/nA_2)\}}} \qquad [7]$$
which in most cases can be replaced by the more approximate formula
$$f_0 = 2100\sqrt{nA_2/l_2V} \qquad [8]$$
where the symbols refer to Fig. 4, and $n$ = number of passages per stage of sectional area $A_2$. If $l_2$ is very small, such that $A_2$ is a simple orifice, the ratio $A_2/l_2$, is replaced by $D$, the diameter of the orifice. If there are a number of passages or orifices $n$ in each stage, $A_2/l_2$, or $D$, must be multiplied by that number.

Example of Design of Type 1 Filter.—A whistle of about 400 cycles frequency is emitted from a tube of $1/16$ sq. in. cross-sectional area. A Type 1 filter, Fig. 3, should eliminate this sound. $A_1 = 1/16$ sq. in.; $l$ = 3 in.; $V$ = 3 cu. in. Fig. 5 is a section of the filter in three stages, the two end tubes being shortened. From [6], $f_0 = 4200\sqrt{1/16 \times 1/3 \times 1/3} = 350$ cycles. The lower frequency limit of this filter being 350 cycles, the 400-cycle frequency should be eliminated. The wave length of this sound is $(1100/400) \times 12 = 33$ in., which is greater than the filter stage length. For higher frequencies, where the wave length is below 6 in., the filtering action will fall off.

Example of Design of Type 2 Filter.—Apply this type of design to the previous problem. Type 2 filters are more selective than Type 1, as filtering action is efficient only for frequencies near and just above the lower limiting frequency $f_0$. Fig. 6 is a design which should be satisfactory. Applying approximate formula [8] and design data from Fig. 6, $f_0 = 2100\sqrt{1/4 \times 2 \times 1/16} = 372$ cycles. The more exact formula [7] will give a slightly lower frequency, but sufficiently near 400-cycles to have a strong attenuating action on it. This design also consists of stages which are short compared with the wave length of the sound to be filtered. The filtering action of Type 1 low-pass filter covers a broader range of frequency than Type 2 and is very efficient. Type 2, however, can sometimes be designed to occupy considerably less space than Type 1. The preferred type depends on conditions to be met.

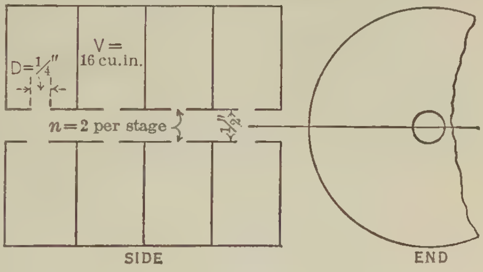


Fig. 6. Special Filter Design

**HIGH-PASS FILTERS** are not so efficient in performance as low-pass filters, and are, as a rule, more bulky. Fig. 7 is a typical high-pass filter of simple design. The limiting frequency, above which such a filter ceases to perform, is
$$f_0 = 2100\sqrt{nA_2/A_1 l_1 l_2} \qquad [9]$$
The symbols are shown in Fig. 7, and, as before, $A_2/l_2 = D$ = hole diameter when $l_2$ is small, and $n$ = number of holes of area $A_2$ per stage.

This type of filter functions through successive side leakage of pressure waves as they pass through the tube. When the wave escapes from the tube end, its pressure amplitude and its intensity are greatly reduced. An appreciable amount of sound, however, escapes through the side holes. Good high-pass filters require an outer shell of considerably larger diameter than the inner tube, the interspace being stuffed with a damping material as metal wool or shavings. Steel wool will rust in gas engine exhaust mufflers.

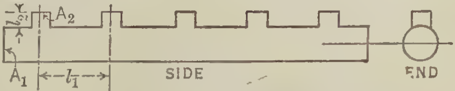


Fig. 7.—High-pass Filter

Fig. 8 is a cross-section of an efficient muffler which has considerably less back pressure than the baffle-type automobile muffler. For high attenuation, many small holes are better than the few large ones in Fig. 6. The use of formula [9] is not necessary for explosion mufflers, as these filters always attenuate low frequency impulse waves. This formula will show that the limit $f_0$, above which attenuation ceases, is well above these low frequencies, being seldom below 100 cycles per sec.

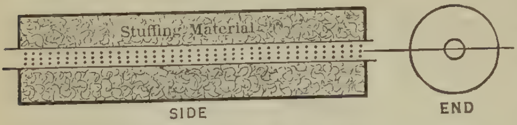


FIG. 8. Explosion Muffler

**PERFORMANCE.**—An ideal sound filter produces a certain percentage drop in sound intensity per stage for every frequency within its range of attenuation. Thus, a 6-stage low-pass filter may attenuate a 300-cycle tone by 4 db. per stage, or 24 db. in all. A sharply tuned Type 2 low-pass filter will attenuate much more. Attenuation in db. per stage is independent of the initial sound intensity.

Performance of a low-pass filter sometimes is disappointing. Tube resonance effects must be avoided. A tube of length equal to a multiple of one-half the wave length of sound of a given frequency will resonate to that frequency. An otherwise well-designed filter may not perform well if such effects enter; placing it in different positions in the duct usually will eliminate them. High-pass filters, as mufflers, are less subject to this trouble. A fairly good knowledge of the elementary principles of sound is desirable in making sound filter applications. For a discussion of sound filtration, see Stewart and Lindsay, Acoustics, chap. vii (D. Van Nostrand Co., N. Y.).

## 6. INDIRECT NOISE

Indirect noise is the result of transmission through a structure of sound vibration from an exciting source which may be quite remote. An ordinary structure has ability to transmit vibration waves a long distance almost perfectly noiselessly to some point. For example, a panel with the same natural frequency as the wave, may resonate with surprising strength. The source of vibration sometimes is hard to find. The treatment of indirect noise is to isolate the source by elastic suspension. Vibration isolation is discussed on p. 16–14; the same methods apply to cases of noise isolation.

# Section 17

# STRUCTURES AND BUILDINGS

## STRESSES IN FRAMED STRUCTURES

## BUILDING CONSTRUCTION

By Theodore Crane

## PROPERTIES OF STRUCTURAL MATERIALS

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# STRUCTURES AND BUILDINGS

## STRESSES IN FRAMED STRUCTURES

Framed structures in general consist of one or more triangles, for the reason that the triangle is the one polygonal form whose shape cannot be changed without distorting one of its sides. Problems in stresses of simple framed structures generally may be solved either by the application of the triangle, parallelogram, or polygon of forces, by the principle of the lever, or by the method of moments.

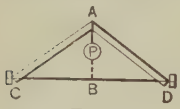

FIG. 1

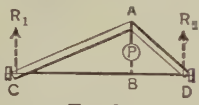

FIG. 2

**MAXIMUM THRUST OR TENSION IN TRUSSES** is at the center of the horizontal chords. It is equal to $WL/8D$, where $W$ = total load supported by truss, including weight of truss; $L$ = length and $D$ = depth of truss.

**TWO DIAGONAL BRACES AND A TIE ROD.**—If the braces are of equal length (Fig. 1), and sustain a single load $P$, compressive stress $S$ in each brace is as follows:

$$S_{AD} = {}^1/_2 P \times (AD/AB); \quad S_{AC} = {}^1/_2 P \times (AC/AB).$$

Each abutment $C$ and $D$ supports one-half the load $P$.

If the braces are of unequal length (Fig. 2),

$$S_{AC} = R_1 \times (AC/AB); \quad S_{AD} = R_2 \times (AD/AB),$$

where $R_1$ and $R_2$ are the reactions on the abutments $C$ and $D$ respectively. Let load $P$ be applied at point $B$ on $CD$. Then

$$R_1 = P \times (BD/CD); \quad R_2 = P \times (BC/CD).$$

Stress in tie $CD$ is

$$S_{CD} = R_1 \times (CB/AB) = R_2 \times (BD/AB).$$

If $\quad CB = BD, R_1 = R_2 \quad$ and $\quad S_{CD} = {}^1/_2 P \times {}^1/_2 (CD/AB).$

If the braces support a uniform load, as in a pair of rafters, the stresses are equivalent to those set up by one-half the load applied at the center. The horizontal thrust of the braces against each other at the apex equals the tensile stress in the tie $CD$.

**KING-POST TRUSS.**—See Fig. 3. Braces $AC$ and $AD$ are in compression; strut $AB$ and tie $CD$ are in tension. A uniform load $P$ may be considered as a concentrated load ${}^1/_2 P$ at the center. Then the stresses in the several members are:

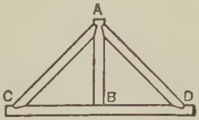

FIG. 3. King Post Truss

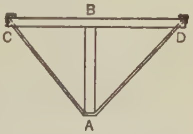

FIG. 4. Inverted King Post Truss

$$S_{AC} = S_{AD} = {}^1/_4 P \times (AD/AB) = {}^1/_4 P \times (AC/AB);$$
$$S_{AB} = {}^1/_2 P; \quad S_{CD} = {}^1/_4 P \times (BD/AB).$$

Horizontal thrust of brace $AD$ at $D = S_{CD}$.

If the truss is inverted, as in Fig. 4, the stress is compression in the strut $AB$ and tension in members $AC$ and $AD$.

**QUEEN-POST TRUSS.**—Members $AB$, $BC$, and $CD$, Fig. 5, are in compression; $AD$, $BE$ and $CF$ are in tension. If the queen-posts $BE$ and $CF$ divide $AD$ into three equal parts, a uniform load may be considered as divided also in three equal parts, two of which are concentrated at $E$ and $F$, the remaining part being divided between the two abutments at $A$ and $D$.

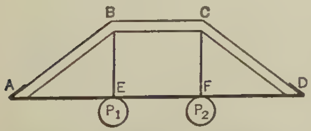

FIG. 5. Queen Post Truss

Only the loads $P_1$ and $P_2$, concentrated at $E$ and $F$ respectively, are considered as affecting the members of the truss. Stresses are as follows:

$$S_{BE} = P_1;\quad S_{CF} = P_2;\quad S_{CD} = S_{AB} = P_1 \times (CD/CF) = P_2 \times (AB/BE);$$
$$S_{AE} = S_{EF} = S_{FD} = P_1 \times (FD/CF) = P_2 \times (AE/BE).$$

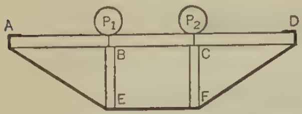

FIG. 6. Inverted Queen Post Truss

Thrust on $BC$ = tension in $EF$. To resist heavy unequal loads, diagonal braces should be provided from $E$ to $C$ and from $B$ to $F$.

If the truss is inverted, as in Fig. 6, $AB$, $BC$, $CD$, $BE$ and $CF$ are in compression. $AE$, $EF$ and $FD$ are in tension.

**BURR TRUSS OF FIVE PANELS** (Fig. 7).—If the panels are of equal length, $^1/_5$ of the load may be considered as being concentrated at each panel point, and $^1/_5$ divided equally between the two abutments. To determine stress in $AB$ and $CD$, consider the truss as a queen-post truss, with loads $P_1$ and $P_2$ concentrated at $E$, and loads $P_3$ and $P_4$ concentrated at $F$. To determine stress in $GE$ and $HF$, consider the portion of the truss between $E$ and $F$ as a smaller queen-post truss with loads $P_2$, $P_3$ at $K$ and $L$ respectively. The stresses are

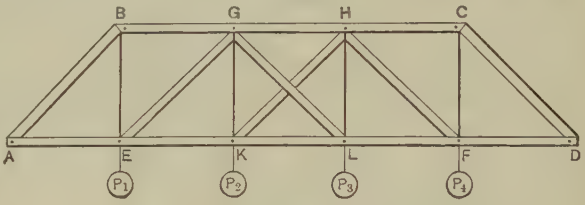

FIG. 7. Burr Truss

$S_{AB} = S_{CD} = (P_1 + P_2) \times (AB/BE) = (P_3 + P_4) \times (CD/CP)$ . [compression]
$S_{GE} = S_{HF} = P_2 \times (GE/GK) = P_3 \times (HF/HL)$ . . . . . . . [compression]
$S_{BE} = S_{CF} = P_1 + P_2 = P_3 + P_4$ . . . . . . . . . . [tension]
$S_{GK} = S_{HL} = P_2 = P_3$ . . . . . . . . . . . . [tension]
$S_{BG} = S_{CH} = (P_1 + P_2) \times (AE/BE) = (P_3 + P_4) \times (FD/FC)$ . . [compression]
$S_{GH} = (P_1 + P_2 + P_3) \times (AE/BE)$ . . . . . . . . . . [compression]
$S_{AE} = S_{FD} = (P_1 + P_2) \times (AE/BE) = (P_3 + P_4) \times (FD/FC)$ . [tension]
$S_{EK} = S_{KL} = S_{LF} = (P_1 + P_2 + P_3) \times (AE/BE)$ . . . . . . [tension]

Stress is received by $GL$ and $HK$ only when the truss is unequally loaded.

**PRATT OR WHIPPLE TRUSS** (Fig. 8).—In this truss, diagonals are ties and verticals are struts. The general formula for stress in the verticals with a uniformly distributed static load is

$$S_v = [\{(n - x) + (n - x)^2 - (x - 1) + (x - 1)^2\}P]/2n,$$

and for a moving load

$$S_{vr} = \frac{\{(n - x) + (n - x)^2 - (x - 1) + (x - 1)^2\}P}{2n} + \frac{r(x - 1) + (x - 1)^2}{2n}$$

where $n$ = number of panels in truss; $x$ = number of any vertical, considered from the nearest end, and counting the end as 1; $r$ = moving load for each panel; $P$ = total load for each panel.

Stress in principal diagonals ($AJ$, $GN$, etc.) is

$$S_D = S_v \sec \theta,$$

where $\theta$ = angle which the diagonal makes with the vertical, *e.g.* angle $AJB$ or angle $HBJ$.

Stress $S_C$ in counterbraces ($BH$, $CJ$, $FO$, etc.) depends on the position of the brace. If the load is uniform, $S_C = \theta$ in the first panel.

In the second panel, $S_C = P \sec \theta \times (1/n)$.

In the third panel, $S_C = P \sec \theta \times \{(1 + 2)/n\}$.

In the fourth panel, $S_C = P \sec \theta \times \{(1 + 2 + 3)/n\}$, etc.

**Calculation by Distribution of Stress.**—Assume a truss of six panels, uniformly loaded with a total load $P$. The portion of the load applied at each panel point may be considered as $^1/_6 P$. The loads transmitted to the abutments from each panel are as follows:

| | Load $P_1$ | $P_2$ | $P_3$ | $P_4$ | $P_5$ |
|---|---|---|---|---|---|
| To abutment $H$.................... | $^5/_6 P_1$ | $^4/_6 P_2$ | $^3/_6 P_3$ | $^2/_6 P_4$ | $^1/_6 P_5$ |
| To abutment $O$.................... | $^1/_6 P_1$ | $^2/_6 P_2$ | $^3/_6 P_3$ | $^4/_6 P_4$ | $^5/_6 P_5$ |

Each of the above loads is transmitted through each strut and tie between its point of application and the abutment. Thus $^5/_6\,P_1$ is transmitted through members $JA$ and $AH$ to abutment $H$ and $^1/_6\,P_1$ is transmitted through members $JC$, $CK$, $KD$, $DL$, $LE$,

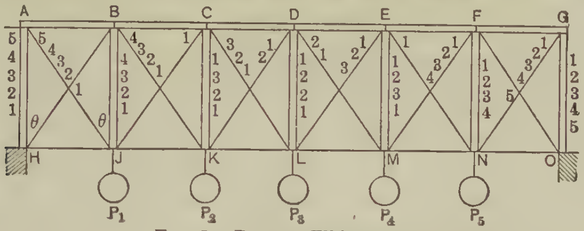

FIG. 8. Pratt or Whipple Truss

etc., to abutment $O$. Similarly, $^4/_6\,P_2$ is transmitted through $KB$, $BJ$, $JA$, and $AH$ to abutment $H$, and $^2/_6\,P_2$ is transmitted through $KD$, $DL$, $LE$, etc., to abutment $O$. As each load is thus analyzed, the numerator of the fraction of load transmitted may be written alongside the member affected as in Fig. 8 or tabulated as follows:

Diagonals in Tension

| | AJ | BH | BK | CJ | CL | DK | DM | EL | EN | FM | FO | GN |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $P_1$........ | 5 | ..... | ...... | 1 | ...... | 1 | ...... | 1 | ...... | 1 | ...... | 1 |
| $P_2$........ | 4 | ..... | 4 | ..... | ...... | 2 | ...... | 2 | ...... | 2 | ...... | 2 |
| $P_3$........ | 3 | ..... | 3 | ..... | 3 | ...... | ...... | 3 | ...... | 3 | ...... | 3 |
| $P_4$........ | 2 | ..... | 2 | ..... | 2 | ...... | 2 | ...... | ...... | 4 | ...... | 4 |
| $P_5$........ | 1 | ..... | 1 | ..... | 1 | ...... | 1 | ...... | 1 | ...... | ...... | 5 |
| Total $T_t$ .. | 15 | 0 | 10 | 1 | 6 | 3 | 3 | 6 | 1 | 10 | 0 | 15 |

Struts in Compression

| | AH | BJ | CK | DL | EM | FN | GO |
|---|---|---|---|---|---|---|---|
| $P_1$........... | 5 | .......... | 1 | 1 | 1 | 1 | 1 |
| $P_2$........... | 4 | 4 | .......... | 2 | 2 | 2 | 2 |
| $P_3$........... | 3 | 3 | 3 | .......... | 3 | 3 | 3 |
| $P_4$........... | 2 | 2 | 2 | 2 | .......... | 4 | 4 |
| $P_5$........... | 1 | 1 | 1 | 1 | 1 | .......... | 5 |
| Total, $T_c$..... | 15 | 10 | 7 | 6 | 7 | 10 | 15 |

$$\text{Tension in the diagonals} = T_t \sec\theta \times {}^1/_6\,P.$$

$$\text{Compression in struts} = T_c \times {}^1/_6\,P.$$

If diagonal struts are used at $HB$ and $FO$, instead of the struts $AH$, $GO$ and the diagonals $AJ$, $GN$, $^5/_6\,P$ will be carried through $FN$ and $NO$. Stress in $BH$ and $FO$ will equal $^{15}/_6\,P\sec\theta$.

Stresses in upper chords are compression, and in lower chords are tension. They increase from the ends to the center. The thrust in each member as $AB$, $BC$, etc., is the horizontal component of the tension in the diagonal of the panel, plus the thrust on the members between it and the end of the truss. Thus,

$$\text{Thrust in } AB \text{ and } FG = {}^{15}/_6\,P \times \tan\theta.$$

$$\text{Thrust in } BC \text{ and } EF = {}^{15}/_6\,P \times \tan\theta + {}^{10}/_6\,P\tan\theta.$$

$$\text{Thrust in } CD \text{ and } DE = {}^{15}/_6\,P\tan\theta + {}^{10}/_6\,P\tan\theta + {}^{6}/_6\,P\tan\theta.$$

The tension in the lower chord of each panel is the same as the thrust in the upper chord of the panel. See Method of Moments below.

If loads $P_1$, $P_2$, $P_3$, etc., are unequal, the value of each must be taken into account in distributing stress. Thus tension in $AJ$, instead of being $^{15}/_6\,P\sec\theta$, would be

$$\sec\theta\,({}^5/_6\,P_1 + {}^4/_6\,P_2 + {}^3/_6\,P_3 + {}^2/_6\,P_4 + {}^1/_6\,P_5).$$

Each panel load should include its fraction of the weight of the truss.

**Method of Moments.**—Let the truss be uniformly loaded with a total load $= W$. Weight supported at each end, or reaction of the abutment $= W/2$. Length of the truss $= L$. Weight on a unit of length $= W/L$. Horizontal distance from nearest abutment to any point (say $M$, Fig. 8) in the chord where stress is to be determined $= x$. Horizontal stress at that point (tension on the lower chord, compression in the upper) $= H$. Depth of the truss $= D$. The difference of the moment about the point $M$ of the reaction of the abutment, and of the moment of the load between $M$ and the abutment, is equated to the moment of resistance or stress in the horizontal chord, considered with reference to the point in the opposite chord about which the truss would turn if the first chord were severed at $M$.

$$\text{Moment of reaction of abutment} = (W/2) \times x.$$

$$\text{Moment of load from abutment to } M = (W/Lx) \times (x/2) = Wx^2/2\,L.$$

$$\text{Moment of stress in chord} = HD = (Wx/2) - (Wx^2/2\,L),$$

whence

$$H = (W/2\,D)\{x - (x^2/L)\}.$$

**THE HOWE TRUSS** (Fig. 9).—In the Howe truss the diagonals are struts, and the verticals are ties. The calculation of stress may be made in the same method as described for the Pratt truss.

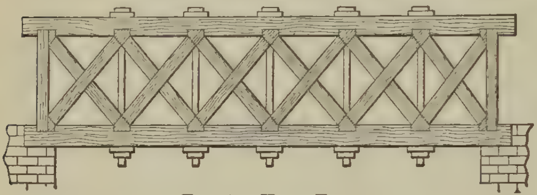

FIG. 9. Howe Truss

**THE WARREN GIRDER** (Fig. 10) has no vertical struts, and the diagonals may transmit either tension or compression. Stresses in the diagonals may be calculated by the method of distribution of stresses as in the case of the rectangular truss.

The loads $P_1$, $P_2$, $P_3$, etc., are respectively $^1/_{10}$, $^3/_{10}$, $^5/_{10}$, etc., of the length of the span from $a$, and the loads transmitted to the abutments from each panel are as follows:

| | Load $P_1$ | Load $P_2$ | Load $P_3$ | Load $P_4$ | Load $P_5$ |
|---|---|---|---|---|---|
| To abutment $a$............... | $0.9\,P_1$ | $0.7\,P_2$ | $0.5\,P_3$ | $0.3\,P_4$ | $0.1\,P_5$ |
| To abutment $g$............... | $0.1\,P_1$ | $0.3\,P_2$ | $0.5\,P_3$ | $0.7\,P_4$ | $0.9\,P_5$ |

These quantities can be written alongside the members affected by the transmission of the several loads, tension values being written to the left and compression values to the right as shown in Fig. 10, or they can be tabulated as below.

| | $a1$ | $b1$ | $b2$ | $c2$ | $c3$ | $d3$ | $d4$ | $e4$ | $e5$ | $g5$ |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Tension Stresses | | | | | |
| $P_1$......... | ....... | ....... | 1 | ....... | 1 | ....... | 1 | ....... | 1 | ....... |
| $P_2$......... | ....... | 7 | ....... | ....... | 3 | ....... | 3 | ....... | 3 | ....... |
| $P_3$......... | ....... | 5 | ....... | 5 | ....... | ....... | 5 | ....... | 5 | ....... |
| $P_4$......... | ....... | 3 | ....... | 3 | ....... | 3 | ....... | ....... | 7 | ....... |
| $P_5$......... | ....... | 1 | ....... | 1 | ....... | 1 | ....... | 1 | ....... | ....... |
| Total...... | 0 | 16 | 1 | 9 | 4 | 4 | 9 | 1 | 16 | 0 |
| | | | | | Compression Stresses | | | | | |
| $P_1$......... | 9 | 1 | ....... | 1 | ....... | 1 | ....... | 1 | ....... | 1 |
| $P_2$......... | 7 | ....... | 7 | 3 | ....... | 3 | ....... | 3 | ....... | 3 |
| $P_3$......... | 5 | ....... | 5 | ....... | 5 | 5 | ....... | 5 | ....... | 5 |
| $P_4$......... | 3 | ....... | 3 | ....... | 3 | ....... | 3 | 7 | ....... | 7 |
| $P_5$......... | 1 | ....... | 1 | ....... | 1 | ....... | 1 | ....... | 1 | 9 |
| Total....... | 25 | 1 | 16 | 4 | 9 | 9 | 4 | 16 | 1 | 25 |
| | | | | | Net Values | | | | | |
| Tension.... | ....... | 15 | ....... | 5 | ....... | ....... | 5 | ....... | 15 | ....... |
| Compression | 25 | ....... | 15 | ....... | 5 | 5 | ....... | 15 | ....... | 25 |

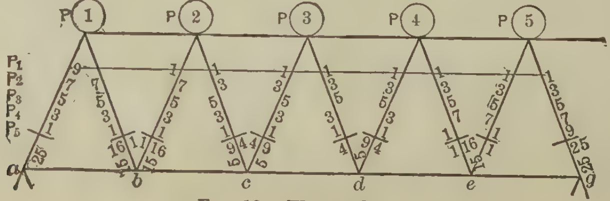

FIG. 10. Warren Girder

To determine the stress in any member, multiply net values as found above by $^1/_{10}$ of one of the loads as $P_1$ and by the secant of $^1/_2$ the angle between the diagonals. Stresses in horizontal chords may be determined by the method of moments. See p. 17–05.

**ROOF TRUSS.—Solution by Method of Moments.**—The calculation of stresses in structures by the method of statical moments consists in taking a cross-section of the structure at a point where there are not more than three members (struts, braces or chords). To find the stress in either one of these members take the moment about the intersection of the other two as an axis of rotation. The sum of the moment of these members must be zero if the structure is in equilibrium. But the moments of the two members that pass through the point of reference or axis are both zero, hence one equation containing one unknown quantity can be found for each cross-section.

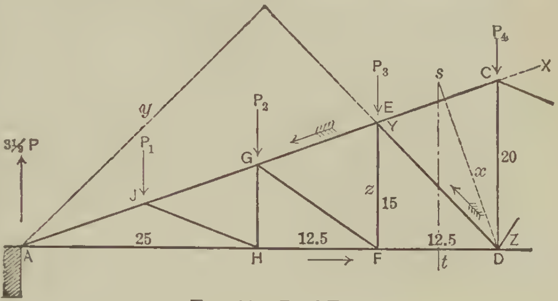

FIG. 11. Roof Truss

In the truss shown in Fig. 11, take a cross-section at $ts$ and determine the stress in the three members cut by it, *viz.*, $CE$, $ED$ and $DF$. Let $X$ = force exerted in direction $CE$, $Y$ = force exerted in direction $DE$, $Z$ = force exerted in direction $FD$.

Moment of $X$ about the intersection of $Y$ and $Z$ at $D = Xx$.
Moment of $Y$ about the intersection of $X$ and $Z$ at $A = Yy$
Moment of $Z$ about the intersection of $X$ and $Y$ at $E = Zz$.

Let $z = 15$; $x = 18.6$; $y = 38.4$; $AD = 50$ ft.; $CD = 20$ ft. Let $P_1$, $P_2$, $P_3$, $P_4$ be equal loads as shown, whence the reaction of abutment $A$ will be $3\,^1/_2\,P$. The sum of all moments about $D$, $A$, or $E$ will be zero when the structure is at rest. Moments in a clockwise direction are designated +; counter-clockwise moments are designated −.

Then $$-Xx + (3.5\,P \times 50) - (P_3 \times 12.5) - (P_2 \times 25) - (P_1 \times 37.5) = 0.$$

Since $$P = P_1 = P_2 = P_3, \quad \text{and} \quad x = 18.6,$$

$$-18.6\,X + 175\,P - 75\,P = 0, \quad \text{and} \quad X = 5.376\,P.$$

Similarly, $$-Yy + (P_3 \times 37.5) + (P_2 \times 25) + (P_1 \times 12.5) = 0, \quad \text{and} \quad Y = 1.953\,P.$$

$$-Zz + (3.5\,P \times 37.5) - (P_1 \times 25) - (P_2 \times 12.5) - (P_3 \times 0) = 0, \quad \text{and} \quad Z = 6.25\,P.$$

Stresses in the other members may be found in the same manner to be: $EG = 6.73\,P$; $GJ = 8.07\,P$; $JA = 9.42\,P$; $JH = 1.35\,P$; $GF = 1.59\,P$; $AH = 8.75\,P$; $HF = 7.50\,P$.

**ANALYSIS OF FINK ROOF TRUSS** (Fig. 12).—Let $W$ = total load on roof; $N$ = number of panels on both rafters; $W/N = P$ = load at each joint, $b$, $d$, $f$, etc.; $V$ = reaction at $A = {}^1/_2\,W = {}^1/_2\,NP = 4\,P$; $AD = S$; $AC = L$; $CD = D$; $t_1$, $t_2$, $t_3$ = tension in $De$, $eg$, $gA$, respectively; $C_1$, $C_2$, $C_3$, $C_4$ = compression in $Cb$, $bd$, $df$, and $fA$ respectively. Then the stresses in the several members will be:

| Member | Stress | Member | Stress | Member | Stress |
|---|---|---|---|---|---|
| 1 or $De$ | $t_1 = 2\,PS/D$ | 5 or $fd$ | $c_3 = {}^7/_2\,PL/D - PD/L$ | 9 or $de$ | $2\,PS/L$ |
| 2 " $eg$ | $t_2 = 3\,PS/D$ | 6 " $db$ | $c_2 = {}^7/_2\,PL/D - 2\,PD/L$ | 10 " $cd$ or $dg$ | ${}^1/_2\,PS/D$ |
| 3 " $gA$ | $t_3 = {}^7/_2\,PS/D$ | 7 " $bC$ | $c_1 = {}^7/_2\,PL/D - 3\,PD/L$ | 11 " $ec$ | $PS/D$ |
| 4 " $Af$ | $c_4 = {}^7/_2\,PL/D$ | 8 " $bc$ or $fg$ | $PS/L$ | 12 " $cC$ | ${}^3/_2\,PS/D$ |

EXAMPLE.—Find the stresses in the members of a Fink truss as shown in Fig. 12. Span 64 ft.;

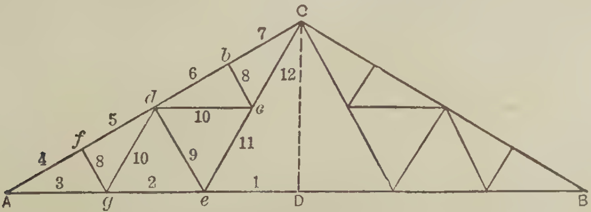

FIG. 12. Fink Roof Truss

depth, 16 ft.; 4 panels on each side; total load, 32 tons, or 4 tons each at points *f*, *d*, *b*, *C*, etc., and 2 tons each at *A* and *B*, which, however, transmit no stress to the truss members.

*Solution.*—$W = 32$; $P = 4$; $S = 32$; $D = 16$; $L = \sqrt{S^2 + D^2} = 2.236\,D$; $L/D = 2.236$; $D/L = 0.4472$; $S/D = 2$; $S/L = 0.8944$. Stresses in the members then are:

| Member | Stress | Member | Stress |
|---|---|---|---|
| 1 | 2 × 4 × 2 = 16 tons | 7 | 31.3 − (12 × 0.447) = 25.94 tons |
| 2 | 3 × 4 × 2 = 24 " | 8 | 4 × 0.8944 = 3.58 " |
| 3 | 7/2 × 4 × 2 = 28 " | 9 | 8 × 0.8944 = 7.16 " |
| 4 | 7/2 × 4 × 2.236 = 31.3 " | 10 | 2 × 2 = 4 " |
| 5 | 31.3 − (4 × 0.447) = 29.52 " | 11 | 4 × 2 = 8 " |
| 6 | 31.3 − (8 × 0.447) = 27.72 " | 12 | 6 × 2 = 12 " |

**GRAPHICAL ANALYSIS OF TRUSSES.**—The graphic method of determining stresses in truss members is applicable to any truss and any manner of loading; it is sufficiently accurate for practical purposes and the work can be done more quickly and with less danger of error than by the numerical method. The graphical method is based on the principle that straight lines drawn to scale to represent the magnitude and direction of forces acting on a point in equilibrium will form a closed polygon. By taking one joint of the truss after another the known external or internal forces acting at the joint are laid out and the unknown forces are determined by closing the polygon.

**Bow's Notation** is the most convenient method of designating areas, members and forces in graphical analyses. In this system, areas between truss members, members and external forces or between external forces are denoted by letters. Any member, panel load or reaction then may be designated by the two letters on each side. Thus in Fig. 13, *AH* represents the diagonal at the end of the truss, *AB* represents the panel load at the apex of the first panel *H*, and *HJ* represents the diagonal between panels *H* and *J*. When the stress diagram (Fig. 14) is drawn, the ends of a line representing the stress in a member or an external force are given the same letters used to denote the corresponding member or load in the truss diagram. Thus, in Fig. 14, the line *ab* represents the panel load *AB*, the line *ah* represents the stress in member *AH*, the line *ga* represents the reaction *GA*, etc.

**Procedure.**—1. Draw the truss diagram representing the truss members and indicate the points of application of all external forces acting on the truss including all loads applied at panel points and the reactions at points of support.

2. Determine, by computation, the vertical loads applied at the panel points and the reactions at points of support. If the panel loads are symmetrical with respect to the center of the span, both reactions are equal to half of the total load on the truss; if the truss is not loaded symmetrically the reactions are obtained by the method of moments.

3. Start the stress diagram (Fig. 14) on the same sheet of paper as the truss diagram, by drawing to scale the vertical load line *af* representing the total load; divide *af* so that lines *ab*, *bc*, *cd*, *de* and *ef* represent the corresponding panel loads. Locate point *g* so that *fg* and *ga* represent to scale the right and left reactions, respectively. In Fig. 13 the panel loads are assumed to be equal and, therefore, $fg = ga$. The polygon of forces acting on the truss is *a b c d e f g a*. The polygon must close as the truss is in equilibrium.

The stresses in the members are obtained in the stress diagram by laying out in clockwise order the forces acting at each joint of the truss, starting with the left reaction and constructing the force polygons for one joint after another until all the unknown stresses are determined. Thus in Fig. 14 the line *ga* represents the magnitude and direction of reaction *GA* in Fig. 13; by drawing lines through points *a* and *g* parallel to *AH* and *HG*, respectively, the force triangle *g a h g* is formed and lines *ga*, *ah* and *hg*, taken in order so as to close the force triangle, represent the direction and magnitude of reaction *GA* and stresses in *AH* and *HG*, respectively. The direction of these forces is indicated on the truss diagram by inserting arrows around the joint in members *AH* and *HG* pointing in the

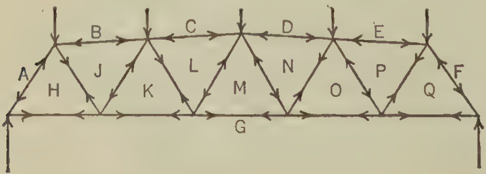

Fig. 13

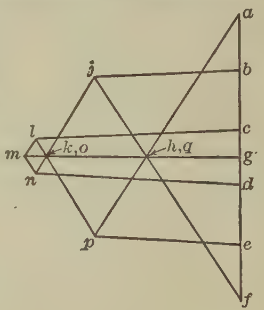

Fig. 14

directions taken by the corresponding stress lines in closing the triangle *g a h g*; therefore, in *AH* the arrow points toward, and in *HG* away from the joint. Next consider the forces acting at the left joint of the upper chord. The lines *ha* (just determined) and *ab* in Fig. 14 represent the magnitude and direction of the stress in *HA* and panel load *AB*, respectively; by drawing lines through *b* and *h* parallel to *BJ* and *JH*, respectively, the force polygon *h a b j h* is found and the lines *bj* and *jh* represent the magnitude and direction of stresses in the truss members *BJ* and *JH*. As before, the direction of forces acting at the joint considered is shown on the truss diagram by inserting arrows around the joint in all members pointing in the directions taken by the corresponding stress lines in closing the polygon *h a b j h*. Similarly, the remaining force polygons are constructed in the order given below to represent the forces acting at the joints indicated:

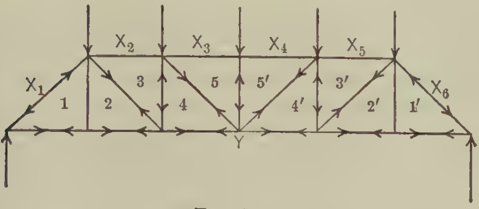

Fig. 15

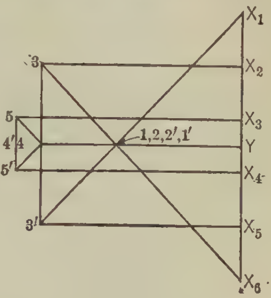

Fig. 16

| Order of Joints Considered | Polygon |
|---|---|
| 2d from left on lower chord | *g h j k g* |
| 2d from left on upper chord | *k j b c l k* |
| 3d from left on lower chord | *g k l m g* |
| 3d from left on upper chord | *m l c d n m* |
| 4th from left on lower chord | *g m n o g* |
| 4th from left on upper chord | *o n d e p o* |
| 5th from left on lower chord | *g o p q g* |
| 5th from left on upper chord | *q p e f q* |
| 6th from left on lower chord | *g q f g* |

At each of the above joints the magnitude and direction of at least two of the forces acting are determined from the polygon constructed for the preceding joint; the stress lines representing these are used as part of the polygon drawn to represent the forces acting at the joint considered.

It will be noted in the truss diagram that in those members where the arrows point toward each other, the stress in the member is tension; where the arrows point in opposite direction, the stress is compression.

The truss and stress diagrams for a six-panel Pratt truss are shown in Figs. 15 and 16, respectively, in which the notation used consists of letters and numbers. It will be noted that in Fig. 16, points 1, 2, 2′ and 1′ are coincident, and, therefore, the stress in verticals 1–2 and 1′–2′ is zero.

In both the Pratt and Warren trusses the end diagonals carry greater stresses than those nearer the center but in the upper and lower chords the members nearest the center carry greater stresses than the members nearer the supports.

**GRAPHICAL ANALYSIS OF TRUSSED BEAM.** (Figs. 17 and 19.)—A beam with rods underneath to take the tension may be considered as an inverted truss. The reactions are determined numerically by the method of moments (see p. 17–05), and the stress diagrams (Figs. 18 and 20) constructed in the same manner as outlined for a Warren or Pratt truss. When the load is suspended under the beam as in Fig. 19, the points *c* and *d* in the stress diagram are coincident and the stress in member *CD* is zero.

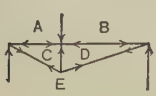

Fig. 17

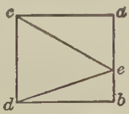

Fig. 18

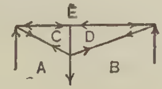

Fig. 19

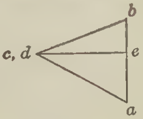

Fig. 20

# BUILDING CONSTRUCTION

By Theodore Crane

## 1. FOUNDATIONS

**LOADS ON SOIL AND ROCK.**—Table 1 gives the maximum loads permitted on various foundation beds by four typical municipal ordinances. These are presumptive bearing capacities of different kinds of soil and rock, and form a guide to determine footing areas, but a most thorough investigation of sub-soil conditions also should be made. On important operations, actual loading tests should be made on the site.

**Table 1.—Maximum Loads on Soils and Rock Allowed by Various Building Ordinances**

| Foundation Bed | Loads in Tons per Square Foot | | | |
|---|---|---|---|---|
| | Boston | San Francisco | Philadelphia | New York |
| Soft clay | 2 * | 1 | 1 * | 1 |
| Fine grained wet sand | 3 * | 2 | 2 * | 2 |
| Dry, fine grained sand or hard clay | 4 | 3 or 4 | 4 | 3 or 4 |
| Coarse gravel without mica or clay | 6 | 6 | 6 | 6 |
| Shale and hardpan | 10 | 10 † | 8 | 10 |
| Medium rock | ... | ... | 24 | 15 |
| Hard rock | ... | 20 | 48 | 40 |
| Solid ledge rock | 100 | ... | ... | ... |

* Protected against lateral displacement, *i.e.*, confined. † Shale only.

When part of a building is supported on a yielding or compressible foundation bed and part on solid rock, the maximum bearing capacity of the soil should be taken at one-half the capacity otherwise allowed. A similar reduction may be advisable when part of the building is supported on piles and part on soil; the pile driving record should be studied before deciding, as the piles may be subject to settlement.

The area of the footing on hard-rock foundation beds is determined by the safe pressure on the concrete immediately above the rock, rather than by the bearing capacity of the rock. A compressive stress of 600 lb. per sq. in. in the concrete, corresponding to 43.2 tons per sq. ft., is the usual figure.

**PILES.**—Timber, concrete, steel tubular or caisson piles are used when suitable soil-bearing value cannot be had at an economic depth. Both timber and concrete piles may be driven to rock, or support their loads by skin friction. Steel piles are driven to bed rock and caisson piles are used to obtain the equivalent of a spread footing without excessive excavation. The first three types may be placed in a double row, preferably staggered beneath a wall, or in groups called *clusters* beneath column footings. The center of gravity of a cluster should coincide with that of the supported load or loads. Piles must not be so closely placed that the bearing capacity of the subsoil is exceeded.

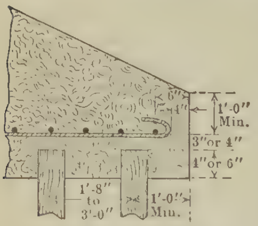

FIG. 1. Reinforced Concrete Footing on Timber Piles

**Timber Piles** may be either hard or soft wood, provided they are comparatively straight and do not split or broom excessively when driven. Minimum diameters are: 6 in. at small end; butt, 10 in. for piles under 25 ft. long, and 12 in. for longer piles; maximum butt diameter, 18 in. Spacing varies from 20 in. to 36 in. on centers. It is imperative that piles be cut off below permanent water level. Butts should extend 6 in. into the base of the superimposed footing. (See Fig. 1.)

**Safe Loads on Timber Piles** are given in the following equations:

$$\text{Pile driven by drop hammer, } P = 2\,Wh/(S + 1);$$

$$\text{Pile driven by single-acting steam hammer, } P = 2\,Wh/(S + 0.1),$$

where $P$ = safe load tons; $h$ = height of fall of hammer, ft.; $W$ = weight of hammer, tons, $S$ = penetration under the last blow, or average under last 5 blows of the hammer, in., provided that the driving has reached a point where successive blows produce approximately equal penetration, and that the maximum load is not over 20 tons.

**Concrete Piles** are desirable if high bearing capacity is required. They may extend above ground-water level, and often effect savings in excavation and foundation cost. Spacing usually is about 3 ft. centers. Concrete piles either may be cast in place or molded before driving. The minimum diameter or lateral dimension of precast piles is 8 in. at the point; average, 12 in. minimum; diameter, at the point, of piles molded in place, 6 in. minimum. The allowable sustaining power, when driven to rock, varies with type of pile employed, but is based on a compressive stress varying from 350 to 500 lb. per sq. in. on the concrete at the average cross-section plus 5000 to 6000 lb. per sq. in. on the vertical reinforcement. When not driven to rock, sustaining power usually is determined by test; in any case, the load on a single pile is limited to 30 or 40 tons.

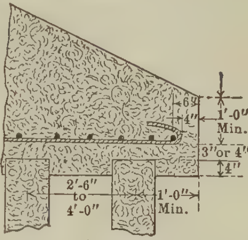

Fig. 2. Reinforced Concrete Footing on Reinforced Concrete Piles

**Steel Tubular Piles** which are filled with concrete after driving usually are 9 in. diam. with a minimum shell thickness of $^5/_{16}$ in. They ordinarily are driven to a full bearing on rock. Bearing capacity may be taken as 500 lb. per sq. in. on the concrete, and 7500 lb. per sq. in. on the steel, disregarding the outer $^1/_{16}$ in. of the shell. Maximum length of such piles is (40 × inside diam. of tube).

**Caisson Piles,** of short, sheet-steel cylinder sections, are sunk by their own weight as earth is dug from below the cutting edge, and later are filled with concrete. The shafts are 36 in. diameter. The base is bell-shaped to provide the required bearing area. Caisson piles can carry the entire weight of a heavily loaded column.

## Footings

**WALL FOOTINGS** under curtain walls and for low buildings of bearing-wall construction, are usually of plain concrete. To determine the width, divide load per linear foot of wall by bearing capacity of the soil; minimum projection on each side of the wall is 6 in. Such footings are constructed either as a slab or with steps. Minimum thickness of bottom block = 12 in.; when stepped, maximum projection of each block is one-half its height. If loads are heavy in proportion to allowable soil pressure, reinforcement may be used in the footing slab. The overhanging portion of each course is designed as a cantilever to resist bending moment and shear caused by upward thrust due to soil pressure.

**FOOTINGS FOR EXTERIOR AND INTERIOR COLUMNS** usually are of reinforced concrete; steel girder footings are occasionally used. Steel slabs or grillages, the latter built of steel beams, are used under heavily loaded columns to distribute the load over a reinforced concrete footing or rock foundation bed. The beams are placed side by side and enclosed in concrete. The protective covering varies from 4 to 8 in.

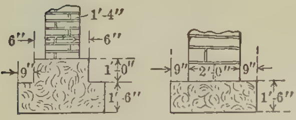

Fig. 3. Alternate Designs for Wall Footings

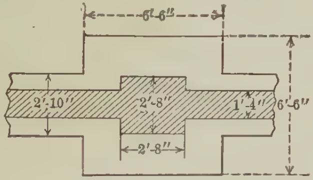

Fig. 4. Footing Below Masonry Pier

With extremely low soil capacities, a reinforced concrete mat over the entire basement of a building may be necessary to utilize all the available area. A mat is designed by the methods used for beam-and-slab or girderless floors, taking the soil reaction as superimposed load. Present practice (1937) favors plain concrete footings for walls and lightly loaded columns. Column footings projecting 18 in. or more beyond the face of column, usually are more economical if reinforced. Steel grillage footings are seldom used except under high buildings resting upon a foundation bed of rock.

For design formulas for reinforced concrete footings and slabs, see Eshbach, Handbook of Engineering Fundamentals, Section 5, forming Vol. 1 of this series.

**PROPORTIONMENT OF FOOTING AREA.**—To equalize the settlements of different parts of a structure resting on a compressible foundation bed, the unit soil pressure should be the same under all footings where soil conditions are similar. Settling of a building is gradual, often long continued, and results from constant dead loads rather

(*Continued on page* 17–13)

### Table 2.—Weights of Building Materials

| Material | Lb. per cu. ft. | Lb. per sq. ft. |
|---|---|---|
| Ashlar masonry, granite, gneiss | 165 | ....... |
| " " limestone, marble | 160 | ....... |
| " " sandstone, bluestone | 140 | ....... |
| Brickwork, common brick | 120 | ....... |
| " pressed brick | 140 | ....... |
| " soft brick | 100 | ....... |
| " 4-in. with 4-in. tile backing | ....... | 60 |
| " 8-in. with 8-in. tile backing | ....... | 75 |
| Cement mortar finish, 1 in. thick | ....... | 12 |
| Cinder fill | 60 | ....... |
| Concrete (cinder) | 100 | ....... |
| " (stone) | 144 | ....... |
| Creosoted wood-block flooring, 3-in. thick | ....... | 15 |
| Gypsum partitions, 3-in. hollow | ....... | 11 1/2 |
| " " 4-in. hollow | ....... | 14 1/2 |
| Hardwood flooring, 7/8-in. thick | ....... | 4 |
| Hollow tile partitions, 3-in. | ....... | 18 |
| " " " 4-in. | ....... | 19 |
| " " " 6-in. | ....... | 25 |
| " " " 8-in. | ....... | 31 |
| " " " 2-in. split-furring | ....... | 8 |
| Pine, spruce or hemlock sheathing | ....... | 2 1/2 |
| Plaster, lime or gypsum, 3/4-in. thick | ....... | 5 |
| Plaster on metal lath, 3/4-in. thick | ....... | 8 |
| Roofing felt, 3-ply, and gravel | ....... | 5 1/2 |
| " " 4-ply, and gravel | ....... | 6 |
| " " 5-ply, and gravel | ....... | 6 1/2 |
| " " 3-ply, and slag | ....... | 4 1/2 |
| " " 4-ply, and slag | ....... | 5 |
| " " 5-ply, and slag | ....... | 5 1/2 |
| Roofing tile, laid in place, Spanish | ....... | 8 to 10 |
| " " laid in place, shingle type, clay | ....... | 12 to 14 |
| " " laid in place, flat, cement | ....... | 15 to 20 |
| " " laid in place, book tile, 2-in. | ....... | 12 |
| " " laid in place, book tile, 3-in. | ....... | 20 |
| Skylights with 3/8-in. wire glass and frame | ....... | 7 1/2 |
| Slate, laid in place, 1/4-in. | ....... | 9 1/2 |
| " " " " 3/8-in. | ....... | 14 1/2 |
| " " " " 1/2-in. | ....... | 19 1/2 |
| Stair construction (steel) | ....... | 50 |
| " " (concrete) | ....... | 150 |
| " " (wood) | ....... | 20 |
| Suspended ceiling (metal lath and cement plaster) | ....... | 10 |
| Windows (glass, frame and sash) | ....... | 8 |

### Table 3.—Minimum Live Load Requirements Used in Design

| Class of Building | Minimum Live Loads per Square Foot of Floor Area | | | | | |
|---|---|---|---|---|---|---|
| | New York, 1930 | Philadelphia, 1929 | Boston, 1926 | Chicago, 1928 | San Francisco, 1928 | Dept. of Commerce 1925 * |
| Dwellings | 40 | 40 | 50 | 40 | 40 | 40 |
| Hotels, tenements, lodging-houses, hospitals | 40 | 40 | 50 | 40 | 40 | 40 |
| Offices, typical floors | 50 | 60 | 60 | 40 | 40 | 50 |
| Schools, classrooms | 60 | 50 | 50 | 75 | 75 | 50 |
| Buildings or rooms for public assembly: | | | | | | |
| With fixed seats | 75 | 60 | 100 | 75 | 75 | 50 |
| Without fixed seats | 100 | 100 | 100 | 125 | 125 | 100 |
| Aisles and corridors | 100 | 100 | 100 | 125 | 125 | 100 |
| Stores, wholesale | 120 | 110 | 250 | 250 | 125 | 100 |
| " retail | 120 | 110 | 125 | 125 | 100 | 75 |
| Manufacturing: | | | | | | |
| Heavy | 120 | 200 | 250 | 250 | 250 | 100 |
| Light | 120 | 120 | 125 | 125 | 125 | 75 |
| Warehouses | 120 | 150 | 125–250 | 125–250 | 125–250 | 100–250 |
| Garages, public | 120 | 100 | 150 | 100 | 100 | 100 |
| " private | 120 | 100 | 75 | 100 | 100 | 80 |
| Sidewalks | 300 | 120 | 250 | 150 | 150 | 250 |

* Building Code Committee of the Bureau of Standards, U. S. Dept. of Commerce.

than occasional live loads. While footing areas must be large enough to avoid a possibility of total live and dead loads exceeding the allowable pressure on the foundation bed, areas must be proportioned on a basis of full dead loads and a percentage of live loads representing probable conditions of occupancy. Divide the sum of the dead load and assumed percentage of live load by the area of the footing having the largest ratio of live to dead load, to obtain the average unit pressure. This average is used to find the size of the remaining footings; in each case use the full dead load plus the assumed percentage of live load.

The following is recommended as a basis for proportioning footing areas for various kinds of buildings:

1. Office buildings, apartment houses, hotels, and schools, *Dead Loads Only;*
2. Industrial buildings for light manufacturing, Dead Load + (0.20 × Live Load);
3. Industrial buildings for heavy manufacturing, with partial storage, Dead Load + (0.40 × Live Load);
4. Warehouses and buildings for storage purposes, Dead Load + (0.60 × Live Load).

**CAISSON AND OPEN-WELL FOUNDATIONS.**—When necessary to carry deep foundations to rock, and below the depth which can be excavated economically, the open-well method may be used if the character of the successive strata of subsoil will permit excavation without compressed air. If character of the soil or pressure of large quantities of water makes this impracticable, pneumatic caissons are used. In either case shafts are later filled with concrete.

## 2. STRUCTURAL DESIGN

**BUILDING ORDINANCES.**—All cities control design of both structural steel and reinforced concrete, by imposing limiting stresses, and by arbitrary limitations in size of structural members and fireproofing requirements. Also, all features concerning fire hazards, or working conditions of the employees, generally are governed by municipal ordinance or State industrial codes. Typical restrictions of this kind, affecting design of industrial work, are the types of buildings in specified zones, height and area limitations, minimum requirements for light and ventilation according to type of occupancy, specifications of materials. The Underwriters' label has become practically a universal standard for all manufactured parts or devices, whose character affects fire risk or safety of occupants in case of fire.

**WORKING STRESSES.**—For steel structures, see Specifications for Structural Steel for Buildings, p. 17–14; for reinforced concrete structures, see Eshbach, Handbook of Engineering Fundamentals, Section 5, forming Vol. 1 of this series.

**SOIL LOADS AND LIVE LOADS** for different types of occupancy, together with weights to be assumed for various materials when computing dead loads, are given in Tables 1, 2 and 3.

**LIVE LOAD REDUCTIONS** may be permitted, for buildings for other than storage purposes, when designing supporting members carrying more than one floor. For example, when computing column sizes for a multi-story building, 100% of the live load is used for the roof and top floor, 90% for next floor below, and correspond decreases for lower floors, but, in no case, less than 50%.

**WIND LOADS.**—See Specification for Structural Steel for Buildings (p. 17–14). The same values may be used for reinforced concrete structures.

**SNOW LOADS** may be taken at 30 lb. per sq. ft. of horizontal projection for slopes up to 20°. Above a 45° slope no snow load need to be considered. For slopes between 20° and 45° the load should be proportionately reduced.

**IMPACT OR DYNAMIC LOADS.**—See p. 7–15.

**ARBITRARY REQUIREMENTS** controlling the thickness of walls and floors, the slenderness ratio of columns, and the maximum unsupported lengths of walls and partitions, are in every building ordinance. Such regulations often control dimensions of structural members, even if actual stresses are less than stipulated by the code.

**FIREPROOFING REQUIREMENTS** for iron or steel require a 3 $^3/_4$ in. thickness of brick, 4 in. of hollow clay tile, or 2 in. of concrete or solid gypsum for columns, and at least 1 $^1/_2$ in. of concrete, or equivalent insulation, for beams and girders.

**EXIT REQUIREMENTS** are important as affecting the plan of a building. Every room of area exceeding 1000 sq. ft., or occupied by more than 75 persons should have at least two doorways, remote from each other, each being a direct exit or furnishing unobstructed egress through other rooms. Every floor area with direct exit to a street, occupied by more than 75 persons, should have at least two exits. Every story without direct exit to the street should have at least one interior stairway or fire tower connected thereto, and at least one additional exit if its area exceeds 2500 sq. ft. Buildings over 60 ft high, except when sprinklered, should have at least one stairway constructed as a fire tower. A fire tower ordinarily is not required if there are two stairways and the structure is not over 100 ft. high, and is sprinklered.

Stairways should be of adequate width to meet conditions of occupancy; at least 44 in. wide when used as legal exits for industrial buildings, and 36 in. wide for storage buildings; the width of standard handrails may be deducted. Stairwells 8 ft. wide between terra cotta or other masonry surfaces provide for up and down runs from intermediate landings which are as wide as the stairs. Minimum width of stair treads usually is 9 1/2 in.; maximum height of risers usually is 7 3/4 in. The product (tread × riser) should be 70 to 75.

## 3. BUILDING FRAMES

Structural steel and reinforced concrete are universally used for frames of fire-resistant buildings. Steel generally is more suitable for structures supporting comparatively light live loads or over 20 stories high. Reinforced concrete is suitable for comparatively low buildings sustaining heavy live loads, and for certain types of monumental buildings of moderate height. For multi-story garages, warehouses and industrial buildings, concrete usually is more satisfactory. For the one-story garage, hangar or manufacturing plant, steel-frame construction is more economical.

For apartment houses, hotels, hospitals and most institutions, four stories high or more, either reinforced concrete or steel may be better, depending on local conditions. Both types of frame usually are enclosed by a veneer; as the architectural appearance is the same in either case, the choice between steel and concrete may be made on a purely economic basis.

Columns of a steel frame usually are H-sections, continuous through two stories, and often reinforced with plates through the lower stories. Built-up columns seldom are used on typical construction. Floors are supported by girder-beams or I-beams, unless the length of span or the amount of load requires plate girders or trusses. Spandrel beams at each story carry wall loads to the columns. The various floor systems appropriate for steel and concrete frames are described on page 17–24.

### Specifications for the Design, Fabrication, and Erection of Structural Steel for Buildings *

**1. MATERIALS.**—*a.* Structural steel shall conform to A.S.T.M. standard specifications for steel for buildings (serial designation A9) as amended to date. *b.* Rivet steel shall conform to A.S.T.M. standard specifications for structural rivet steel (serial designation A 141) as amended to date. *c.* Alloy steels, cast steel, cast iron and other metals shall conform to the applicable specifications of the A.S.T.M. as amended to date. *d.* Stock material shall be of quality substantially equal to that called for by specifications of the A.S.T.M. for classifications covering its intended use; and mill test reports shall constitute sufficient record as to quality of material carried in stock. Unidentified stock material, if free from surface imperfections, may be used for short sections of minor importance, or for small unimportant details, where the precise physical properties of the material would not affect the strength of the structure.

**2. LOADS AND FORCES.**—Steel structures shall be designed to sustain the following loads and forces: Dead load, live load, impact, wind, erection loads, and other forces. *a.* The dead load shall consist of the weight of steel-work and all material fastened thereto or supported thereby. *b.* The live load shall be that stipulated by the Code under which the structure is designed or that required by conditions involved. In general, the live loads should not be less than those recommended by the Building Code Committee of the National Bureau of Standards, November, 1924, under the caption "Minimum Live Loads for Use in the Design of Buildings." *c.* For structures, carrying live loads, inducing impact or vibrational forces, design stresses shall be increased by a percentage of the live load stresses sufficient to provide for such forces. *d.* Proper provision shall be made for stresses caused by wind, both during erection and after completion of building. Wind pressure is dependent on conditions of exposure and geographical location. Allowable stresses specified in Sections 4c and 5 are based on the steel frame being designed to carry a wind pressure of not less than 20 lb. per sq. ft. on the vertical projection of exposed surfaces during erection, and 15 lb. per sq. ft. on the vertical projection of the finished structure. *e.* Proper provision shall be made for temporary stresses caused by erection. *f.* Structures in localities subject to earthquakes, hurricanes, and similar extraordinary conditions, shall be designed with due regard for such conditions.

**3. REVERSAL OF STRESS.**—Members subject to live loads producing alternate tensile and compressive stresses shall be proportioned as follows: To net total compressive and tensile stresses add 50% of the smaller of the two, and proportion the member to resist either of the increased stresses resulting therefrom. Connections shall be proportioned to resist the larger of the two increased stresses.

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* Abstracted from Specifications of Am. Inst. of Steel Construction, revised 1936.

**4. COMBINED STRESSES.**—*a.* Members subject to both axial and bending stresses shall be so proportioned that the quantity $\{(f_a/F_a) + (f_b/F_b)\}$ shall not exceed unity, in which $F_a$ = axial unit stress that would be permitted by this Specification if axial stress only existed; $F_b$ = bending unit stress that would be permitted by this Specification if bending stress only existed; $f_a$ = axial unit stress (actual) = axial stress divided by area of member; $f_b$ = bending unit stress (actual) = bending moment divided by section modulus of member. *b.* Rivets subject to shearing and tensile forces shall be so proportioned that the combined unit stress will not exceed the allowable unit stress for rivets in tension only. *c.* Members subject to stresses produced by a combination of wind and other loads may be proportioned for unit stresses 33 1/3% greater than those specified in Section 7, provided the section thus required is not less than that required for the combination of dead load, live load, and impact (if any).

**5. MEMBERS CARRYING WIND ONLY.**—Members subject only to stresses produced by wind forces may be proportioned for unit stresses 33 1/3% greater than those specified in Section 7.

**6. EFFECTIVE SPAN LENGTH.**—*a.* Beams, girders and trusses ordinarily shall be designed on the basis of simple spans whose effective length is equal to the distance between centers of gravity of the members to which they deliver their end reactions. *b.* If, on the assumption of end restraint, full or partial, based on continuous or cantilever action, beams, girders and trusses are designed for a shorter effective span length than that specified in Section 6*a*, their sections, as well as the sections of the members to which they connect, shall be designed to carry the shears and moments so introduced, in addition to all other forces, without exceeding at any point the unit stresses prescribed in Section 7.

**7. ALLOWABLE UNIT STRESSES.**—All parts of the structure shall be so proportioned that the unit stress in lb. per sq. in. shall not exceed the following:

*a.* **Structural and Rivet Steel**

| | | |
|---|---|---|
| **Tension.** | Structural steel, net section | 20,000 |
| | Rivets, on area based on nominal diameter | 15,000 |
| **Compression.*** | Columns, gross section, $(l/r) \not> 120$ | $17{,}000 - 0.485\,(l^2/r^2)$ |
| | Columns, gross section, $(l/r) > 120$ | $18{,}000/\{1 + (l^2/18{,}000\,r^2)\}$ |
| | Plate girder stiffeners, gross section | 20,000 |
| | Webs of rolled sections at toe of fillet (Crippling, see Section 16*h*) | 24,000 |
| **Bending.*** | Tension on extreme fibers of rolled sections, plate girders, and built-up members (see Section 16*a*) | 20,000 |
| | Compression on extreme fibers of rolled sections, plate girders, and built-up members, for values of $l/b \not> 40$ | $22{,}500/\{1 + (l^2/18{,}000\,b^2)\}$ |
| | Maximum | 20,000 |
| | Stress on extreme fibers of pins | 30,000 |
| **Shearing.** | Rivets, pins, and turned bolts in reamed or drilled holes | 15,000 |
| | Unfinished bolts | 10,000 |
| | Webs of beams and plate girders, gross section | 13,000 |

| | | Double Shear | Single Shear |
|---|---|---|---|
| **Bearing.** | Rivets and turned bolts in reamed or drilled holes | 40,000 | 32,000 |
| | Unfinished bolts | 25,000 | 20,000 |
| | Pins | 32,000 | |
| | Contact area, milled stiffeners and other milled surfaces | | 30,000 |
| | Fitted stiffeners | | 27,000 |
| | Expansion rollers and rockers, lb. per lin. in. | | 600 *d* |

*b.* **Cast Steel.**—Compression and bearing same as structural steel.
Other unit stresses 75% of those for structural steel.

*c.* **Masonry** (**Bearing**).—Granite 800; sandstone and limestone 400; concrete 600.

**8. SLENDERNESS RATIO.**—The ratio of unbraced length to least radius of gyration $(l/r)$ for compression members shall $\not>$ 120 for main compression members, and 200 for bracing and other secondary compression members.

**9. UNSUPPORTED COMPRESSION FLANGES.**—The ratio of unbraced length to width of flange $(l/b)$ for compression flanges of rolled sections, plate girders, and built-up members subject to bending shall $\not>$ 40.

**10. MINIMUM THICKNESS OF MATERIAL.**—*a.* The minimum thickness of steel except for linings, filler, and webs of rolled beams and channels, shall be: For exterior construction 5/16 in.; for interior construction 1/4 in. (These provisions do not apply to light structures such as skylights, marquees, fire-escapes, light one-story buildings, or light miscellaneous steel-work.) *b.* Gusset plates for trusses with end reactions > 35,000 lb. shall $\not<$ 3/8 in. *c.* In compression members consisting of segments connected by cover plates or lacing, or segments connected by webs, the thickness of webs of segments shall $\not<$ 1/32 of the unsupported distance between the nearest rivet lines, or roots of the flanges in case of rolled section. The thickness of cover plates or webs connecting segments shall $\not<$ 1/40 of the unsupported distance between the nearest lines of their connecting rivets, or roots of their flanges in case of rolled sections. *d.* Provision shall be made for parts subject to corrosive agents either by increasing the thickness of material or by effective protection.

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* Notation for the formulas in this specification is as follows: $l$ = unbraced length, in.; $r$ = radius of gyration, in.; $b$ = width of compression flange, in.; $d$ = diameter of roller or rocker, in.

**11. GROSS AND NET SECTIONS.**—*a.* The gross section of a member at any point shall be determined by summing the products of thickness and gross width of each element, measured normal to the axis of the member. The net section shall be determined by substituting for gross width, the net width computed in accordance with paragraphs *c* to *g* of this Section. *b.* Unless otherwise specified, tension members shall be designed on the basis of net section. Columns shall be designed on the basis of gross section. Beams and girders shall be designed in accordance with Section 16 *a.* *c.* In the case of a chain of holes extending across a part in any diagonal or zigzag line, the net width of the part shall be obtained by deducting from the gross width the sum of diameters of all holes in the chain, and adding, for each gage space in the chain, the quantity $(S^2/4g)$.* The critical net section of the part is obtained from that chain which gives the least net width. *d.* For angles, the gross width shall be the sum of the widths of legs less the thickness. The gage for holes in opposite legs shall be sum of the gages from back of angle less the thickness. *e.* For splice members, the thickness shall be only that part of the thickness of member which has been developed by rivets beyond the section considered. *f.* In computing net area the diameter of a rivet hole shall be taken as $1/8$ in. greater than the nominal diameter of rivet. *g.* In pin-connected tension members, the net section across the pin hole, transverse to the axis of member, shall $\geqq$ 140%, and the net section beyond the pin hole, parallel with the axis of the member shall $\geqq$ 100%, of the net section of the body of member. In all pin-connected riveted members, the net width across the pin hole, transverse to the axis of the member, shall preferably not exceed (12 × thickness of member at the pin).

**12. EXPANSION.**—Proper provision shall be made for expansion and contraction.

**13. CONNECTIONS.**—*a.* Connections carrying calculated stresses, except for lacing, sag bars, and girts, shall have not fewer than 2 rivets. *b.* Members meeting at a point shall have their gravity axes meet at a point if practicable; if not, provision shall be made for their eccentricity. *c.* The rivets at the ends of any member transmitting stresses into that member preferably should have their centers of gravity on the gravity axis of the member; otherwise, provision shall be made for the effect of the resulting eccentricity. Pins may be so placed as to counteract the effect of bending due to dead load. *d.* When beams, girders or trusses are designed on the basis of simple spans in accordance with Section 6*a*, their end connections ordinarily may be designed for the reaction shears only. If, however, the eccentricity of the connection is excessive, provision shall be made for the resulting moment. *e.* When beams, girders or trusses are subject both to reaction, shear and end moment, due to the restraint specified in Section 6*b*, their connections shall be specially designed to carry both shear and moment without exceeding at any point the unit stresses prescribed in Section 7. Ordinary end connections comprising only a pair of web angles, with not more than a nominal seat and top angle, shall not be assumed to provide for this kind of end moment. *f.* In truss construction, when rivets carrying computed stress pass through fillers, the fillers shall be extended beyond connected member and extension secured by sufficient rivets to develop strength of filler. Fillers under plate girder stiffeners at end bearings or points of concentrated loads shall be secured by sufficient rivets to prevent excessive bending and bearing stresses. *g.* Compression members when faced for bearings shall be spliced sufficiently to hold connecting members accurately in place. Other joints in riveted work, whether in tension or compression, shall be spliced to transfer the stress to which the member is subject.

**14. RIVETS AND BOLTS.**—*a.* In proportioning rivets, the nominal diameter of the rivet shall be used. *b.* The effective bearing area of pins, bolts, and rivets shall be the diameter multiplied by the length in bearing; except that for countersunk rivets, half the depth of the countersink shall be deducted. *c.* Rivets carrying calculated stress, and whose grip exceeds five diameters, shall have their number increased 1% for each additional $1/16$ in. in the rivet grip. Special care shall be used in heating and driving such rivets. *d.* All field connections may be made with unfinished bolts, except as provided in *e* hereof. *e.* Rivets shall be used for the following connections: Connections for supports of running machinery, or of other live loads which produce impact or reversal; column splices in all tier structures 200 ft. high or more; column splices in tier structures 100 to 200 ft. high, if the least horizontal dimension is < 40% of the height; column splices in tier structures less than 100 ft. high, if the least horizontal dimension is < 25% of the height; connections of all beams and girders to columns, and of any other beams and girders on which the bracing of columns is dependent, in structures over 125 ft. high; roof-truss splices and connections of trusses to columns, column splices, column bracing, and crane supports, in all structures carrying cranes of over 5-ton capacity; any other connections stipulated on the design plans. The height of a tier structure shall be taken as the vertical distance from the curb level to the highest point of the roof beams, in the case of flat roofs, or to the mean height of the gable, in the case of roofs having a rise of more than 1 to a 4 1/2 run. Where the curb level has not been established, or where the structure does not adjoin a street, the mean level of the adjoining land shall be used instead of curb level. Penthouses may be excluded in computing height of structure. *f.* Turned bolts in reamed or drilled holes, as specified in Section 22*d* may be used in shop or field work where it is impossible to drive satisfactory rivets. The finished shank shall be long enough to provide full bearing, and washers shall be used under the nuts to give full grip when the nuts are turned tight. *g.* The end reaction stresses of trusses, girders, or beams, and the axial stresses of tension or compression members which are carried on rivets or bolts, shall have such stresses developed by the shearing and bearing values of the rivets or bolts.

**15. RIVET SPACING.**—*a.* The minimum distance between centers of rivet holes shall be three diameters of the rivet; but the distance shall preferably be not less than 4 1/2 in. for 1 1/4-in. rivets, 4 in. for 1 1/8-in. rivets, 3 1/2 in. for 1-in. rivets, 3 in. for 7/8-in. rivets, 2 1/2 in. for 3/4-in. rivets, 2 in.

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* $s$ = longitudinal spacing (pitch) of any two successive holes. $g$ = transverse spacing (gage) of the same two holes.

for $5/8$-in. rivets, and 1 $3/4$ in. for $1/2$-in. rivets. *b.* The maximum pitch in the line of stress of compression members composed of plates and shapes shall not exceed (16 × thickness of the thinnest outside plate or shape), nor (20 × thinnest enclosed plate or shape), with a maximum of 12 in., and the distance between lines of rivets, at right angles to the direction of stress, shall not exceed (30 × thickness of the thinnest plate or shape). For angles in built sections with two gage lines, with rivets staggered, the maximum pitch in the line of stress in each gage line shall not exceed (24 × thickness of the thinnest plate), with a maximum of 18 in. *c.* The pitch of rivets at the ends of built compression members shall not exceed 4 diameters of the rivets for a length equal to (1 $1/2$ × maximum width of the member). *d.* In tension members composed of two angles, a pitch of 42 in. will be allowed, and in compression members, 24 in., but the ratio $l/r$ for each angle between the rivets shall be not more than $3/4$ of that for the whole member. *e.* The minimum distance from the center of any punched rivet hole to any edge shall be that given in Table A.

**Table A.—Minimum Rivet Distances**

| Rivet diameter, in. | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 |
|---|---|---|---|---|---|---|---|
| Min. edge distance for punched holes, in. | | | | | | | |
| In sheared edge | 1 | 1 1/8 | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/4 |
| In rolled edge, sections with parallel flanges | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 1/2 | 1 3/4 | 2 |
| In rolled edge, sections with sloping flanges | 3/4* | 7/8* | 1* | 1 1/8* | 1 1/4 | 1 1/2* | 1 3/4* |

* May be decreased $1/8$ in. when holes are near end of beam.

*f.* The distance from center of any rivet under computed stress, and that end or other boundary of the connected member toward which the pressure of the rivet is directed, shall $\geqq$ the shearing area of the rivet shank (single or double shear respectively) divided by the plate thickness. This end distance may however be decreased in such proportion as the stress per rivet is less than that permitted under Section 7*a*; and the requirement may be disregarded if the rivet in question is one of three or more in a line parallel to the direction of stress. *g.* The maximum distance from the center of any rivet to the near edge shall be (12 × thickness of plate), but shall not exceed 6 in.

**16. PLATE GIRDERS AND ROLLED BEAMS.**—*a.* Riveted plate girders, cover-plated beams, and rolled beams shall, in general, be proportioned by the moment of inertia of the gross section. No deduction shall be made for standard shop or field rivet holes in either flange; except that in special cases where the reduction of the area of either flange by such rivet holes, calculated in accordance with the provisions of Section 11, exceeds 15% of the gross flange area, the excess shall be deducted. If such members contain other holes, as for bolts, pins, or countersunk rivets, the full deduction for such holes shall be made. The deductions thus applicable to either flange shall be made also for the opposite flange if the corresponding holes are there present. *b.* Plate girder webs shall have a thickness $\geqq 1/170$ of the unsupported distance between flanges. *c.* Cover plates, when required, shall be equal in thickness or shall diminish in thickness from the flange angles outward. No plate shall be thicker than the flange angles. Unstiffened cover plates shall not extend more than 6 in. nor more than (12 × thickness of the thinnest plate) beyond the outer row of rivets connecting them to the angles. *d.* Rivets connecting flanges to the web shall be proportioned to resist the horizontal shear due to bending as well as any loads applied directly to the flange. *e.* Stiffeners shall be placed on the webs of plate girders at the ends and at points of concentrated loads. Such stiffeners shall have a close bearing against the flanges, shall extend as closely as possible to the edge of the flange angles, and shall not be crimped. They shall be connected to the web by enough rivets to transmit the stress. Only that portion of the outstanding legs outside of the fillets of the flange angles shall be considered effective in bearing. Intermediate stiffeners shall be required at all points where $h/t$ exceeds $(8000/\sqrt{s})$; $h$ = clear depth between flanges, in.; $t$ = thickness of web, in.; $s$ = greatest unit shear in panel, lb. per sq. in., under any condition of complete or partial loading. The clear distance between intermediate stiffeners, when stiffeners are required by the foregoing, shall not exceed 84 in. or that given by the formula $d = (270{,}000\,t/s)\sqrt[3]{st/h}$; $d$ = clear distance between stiffeners, in. Intermediate stiffeners may be crimped over the flange angles. Plate girder stiffeners shall be in pairs, one on each side of the web, and shall be connected to the web by rivets spaced not more than (8 × their nominal diameter). *f.* Web splices in plate girders shall be proportioned to transmit the full shearing and bending stresses in the web at the point of splice. Flange splices shall be proportioned to develop full strength of the members cut. *g.* Flanges of plate girders supporting cranes or other moving loads shall be proportioned to resist any lateral forces produced by such loads. *h.* Rolled beams shall be so proportioned that the compression stress at the web toe of the fillets, resulting from concentrated loads, shall not exceed the value of 24,000 lb. per sq. in. allowed in Section 7.

The governing formulas shall be:

For interior loads $R/t(A + 2N) \leqq 24{,}000$;

For end reactions $R/t(A + N) \leqq 24{,}000$;

$R$ = concentrated interior load or end reactions, in.-lb.; $t$ = thickness of web. in.; $A$ = length of bearing, in.; $N$ = distance from outer face of flange to web toe of fillet, in.

**17. TIE PLATES.**—*a.* Open sides of compression members shall be provided with lacing having tie plates at each end, and at intermediate points if lacing is interrupted. Tie plates shall be as near the ends as practicable. In main members carrying calculated stresses end tie plates shall have a length $\geqq$ distance between lines of rivets connecting them to the segments of the member, and intermediate ones $\geqq 1/2$ this distance. Thickness of tie plates shall $\geqq 1/50$ of distance between lines of rivets connecting them to the segments of the members, and the rivet pitch shall be ⋝ 4 diameters. Tie plates shall be connected to each segment by at least 3 rivets. *b.* Tie plates shall be used to secure the parts of tension members composed of shapes. They shall have a length ⋝ 2/3 of the length specified for tie plates in compression members. The thickness shall ⋝ 1/50 of the distance between the lines of rivets connecting them to the segments of the member and they shall be connected to each segment by at least 3 rivets.

**18. LACING.**—*a.* Lacing bars of compression members shall be so spaced that the ratio $l/r$ of the flange included between their connections shall ⋜ 3/4 of that of the member as a whole. *b.* Lacing bars shall be proportioned to resist a shearing stress normal to axis of member equal to 2% of the total compressive stress in the member. In determining the section required, the compression formula shall be used, $l$ being taken as the length of the bar between the outside rivets connecting it to the segment for single lacing and 70% of that distance for double lacing. The ratio $l/r$ shall ⋜ 140 for single lacing and 200 for double lacing. *c.* The thickness of lacing bars shall for single lacing ⋝ 1/40, and for double lacing shall ⋝ the distance between end rivets; their minimum width shall be (3 × diameter of rivets connecting them to the segments). *d.* The inclination of lacing bars to the axis of members preferably shall ⋝ 45° for double lacing and 60° for single lacing. When the distance between the rivet lines in the flanges > 15 in. the lacing shall be double and riveted at the intersection if bars are used, or else shall be made of angles.

**19. ADJUSTABLE MEMBERS.**—The total initial stress in adjustable members shall be assumed as ⋝ 5000 lb.

**20. COLUMN BASES.**—*a.* Provision shall be made to transfer the column loads, and moments if any, to footings and foundations. *b.* Column bases shall be set level and to correct elevation, with full bearing on masonry. *c.* Column bases shall be finished to accord with following: 1. Rolled steel bearing plates, 2 in. or less thick, may be used without planing, provided a satisfactory contact bearing is obtained; rolled steel bearing plates, over 2 in., but 4 in. or less thick, may be straightened by pressing (planed on all bearing surfaces if presses are not available) to obtain a satisfactory contact bearing; rolled steel bearing plates, over 4 in. thick, shall be planed on all bearing surfaces (except as noted under 3). 2. Column bases, other than rolled steel bearing plates, shall be planed on all bearing surfaces (except as noted under 3). 3. Bottom surfaces of bearing plates and column bases which rest on masonry foundations and are grouted to insure full bearing contact need not be planed.

**21. ANCHOR BOLTS.**—Anchor bolts shall be of sufficient size and number to develop the computed stress.

**22. WORKMANSHIP.**—*a.* All workmanship shall be equal to the best practice in modern structural shops. *b.* All material shall be clean and straight. If straightening or flattening is necessary, it shall be done by a process and in a manner that will not injure the material. Sharp kinks or bends shall be cause for rejection. *c.* Rolled sections, except for minor details, preferably shall not be heated, or, if heated, shall be annealed, but this restriction does not apply to gas cutting, Section 22*l*. *d.* Holes for rivets or unfinished bolts shall be 1/16 in. larger than the nominal diameter of the rivet or bolt. If the thickness of the material is not greater than the nominal diameter of the rivet or bolt plus 1/8 in., the holes may be punched. If the thickness of the material is greater than the nominal diameter of the rivet or bolt plus 1/8 in., the holes either shall be drilled from the solid, or sub-punched and reamed. The die for all sub-punched holes, and the drill for all sub-drilled holes, shall be 1/16 in. smaller than the nominal diameter of the rivet or bolt. Holes for turned bolts shall be 1/50 in. larger than the external diameter of the bolt. If the bolts are to be inserted in the shop, the holes either may be drilled from the solid, or sub-punched and reamed. If the bolts are to be inserted in the field, the holes shall be sub-punched in the shop and reamed in the field. All drilling or reaming for turned bolts shall be done after the parts to be connected are assembled. Drifting to enlarge unfair holes shall not be permitted. Holes that must be enlarged to admit the rivets shall be reamed. Poor matching of holes shall be cause for rejection. *e.* Planing or finishing of sheared plates or shapes will not be required unless specifically called for on the drawings. *f.* All parts of riveted members shall be well pinned or bolted and rigidly held together while riveting. Drifting done during assembling shall not distort the metal or enlarge the holes. *g.* All rivets are to be power-driven hot. Rivets driven by pneumatically- or electrically-operated hammers are considered power-driven. Standard rivet heads shall be of approximately hemi-spherical shape and of uniform size throughout the work for the same size rivet, full, neatly finished, and concentric with the holes. Rivets after driving, shall be tight, completely filling the holes, and with heads in full contact with the surface. Rivets shall be heated uniformly to a temperature not exceeding 1950° F.; they shall not be driven after their temperature is below 1000° F. Loose, burned, or otherwise defective rivets shall be replaced. *h.* Compression joints depending on contact bearing shall have the bearing surfaces truly machined to a common plane after the members are riveted. All other joints shall be cut straight. *i.* Ends of lacing bars shall be neat and free from burrs. *j.* Finished members shall be true to line and free from twists, bends and open joints. Compression members may have a lateral variation ⋜ 1/1000 of the axial length between points which are to be laterally supported. A variation of 1/32 in. is permissible in the overall length of members with both ends milled. Members without milled ends which are to be framed to other steel parts of the structure may have a variation from the detailed length not greater than 1/16 in. for members 30 ft. long or less, and not greater than 1/8 in. for members over 30 ft. long. *k.* All steel castings shall be annealed. *l.* The use of a cutting torch is permissible if the metal being cut is not carrying stresses during the operation. To determine the effective width of members so cut, 1/8 in. shall be deducted from each gas-cut edge. The radius of re-entrant gas-cut fillets shall be as large as practicable, but never less than 1 in.

**23. SHOP PAINTING.**—*a.* Before leaving the shop, all steel-work shall be thoroughly cleaned, by effective means, of all loose mill scale, rust and foreign matter. Except where encased in concrete, all steel-work shall be given one coat of approved metal protection, applied thoroughly and evenly and well worked into the joints and other open spaces. All paint shall be applied to dry surfaces. *b.* Parts inaccessible after assembly shall be given two coats of shop paint, preferably of different colors. *c.* Contact surfaces shall be cleaned, by effective means, before assembly, but not painted. *d.* Machine finished surfaces shall be protected against corrosion by a suitable coating.

**24. ERECTION.**—Frames of all steel-skeleton buildings shall be carried up true and plumb, and temporary bracing shall be introduced wherever necessary to take care of all loads to which the structure may be subjected, including erection equipment, and the operation of same. Such bracing shall be left in place as long as required for safety. *b.* As erection progresses the work shall be securely bolted up to take care of all dead load, wind and erection stresses. *c.* Wherever piles of material, erection equipment or other loads are carried during erection, proper provision shall be made to take care of stresses resulting from same. *d.* No riveting shall be done until the structure has been properly aligned. *e.* Rivets driven in the field shall be heated and driven with the same care as those driven in the shop. *f.* Holes for turned bolts to be inserted in the field shall be reamed in the field as specified in Section 22 *d.* *g.* All field rivets and bolts, also all serious abrasions to the shop coat, shall be spot painted with the material used for the shop coat, or an equivalent, and all mud and other firmly attached and objectionable foreign materials shall be removed, before general field painting. Responsibility for this touch-up and cleaning, as well as for general field painting, shall be allocated in accordance with accepted local practices, and this allocation shall be set forth explicitly in the contract.

**25. INSPECTION.**—*a.* Material and workmanship at all times shall be subject to the inspection of experienced engineers representing the purchaser. All inspection, as far as possible, shall be made at the place of manufacture, and the contractor or manufacturer shall co-operate with the inspector, permitting access for inspection to all places where work is being done. Material or workmanship not conforming to the provisions of this specification may be rejected at any time defects are found during the progress of the work.

**WELDED CONNECTIONS.**—Structural members in steel-frame buildings are welded by flowing fused metal between parts to be welded. The new metal, usually a steel rod 1/8 to 1/4 in. diam. and of the same grade as the connected parts, is deposited as a triangular fillet in a corner between adjoining surfaces. Both electric arc-welding and the oxy-acetylene process are used. As strength of the connection depends on quality of workmanship, only experienced welders should be employed. Connections should be designed to facilitate the work.

The strength of the joint is (allowable load per linear inch of fillet × length of weld, in.). Connections usually are so arranged that shear is the principal stress in the welding material. The connection may be designed either on the basis of shear or by shear and moment. The former method is easier of application and is satisfactory if allowable stresses do not exceed the values of Table 4.

**Table 4.—Allowable Loads per Lineal Inch of Triangular Fillet**

| | | | | | |
|---|---|---|---|---|---|
| Size of Fillet, in. | 1/2 | 7/16 | 3/8 | 5/16 | 1/4 |
| Allowable Load, lb. per lineal in. | 4000 | 3500 | 3000 | 2500 | 2000 |

**Table 5.—Working Stresses in Masonry in Compression**

(From Building Code Recommended by the National Board of Fire Underwriters)

| | Portland Cement Mortar [1] | Natural Cement Mortar [2] | Cement Lime Mortar [3] | Lime Mortar [4] |
|---|---|---|---|---|
| | Stress, lb. per sq. in. | | | |
| Brickwork (see Notes) | 175 | 140 | 140 | 75 |
| Hollow walls of brick | 125 | 100 | 100 | ... |
| Ashlar masonry, dressed granite | 800 | 640 | 640 | 400 |
| " " dressed limestone | 500 | 400 | 400 | 250 |
| " " dressed marble | 500 | 400 | 400 | 250 |
| " " dressed sandstone | 400 | 320 | 320 | 160 |
| Rubble stone | 140 | 100 | 100 | ...... |
| Hollow block,[5] gross area | 80 | 70 | 70 | ...... |
| Solid block,[6] gross area | 125 | 100 | 100 | ...... |
| Concrete [7] (Portland cement) | ...... | 250 | | ...... |
| Concrete [8] (natural cement) | ...... | 150 | | ...... |
| Grout, neat Portland cement in thin layers under bases only | ...... | 1000 | | ...... |

NOTES: 1. 1 part Portland cement, not more than 3 parts sand, proportioned by volume; allowable addition of hydrated lime not over 15% of the cement by volume. 2. Same limitations as for Portland cement mortar, but using natural cement. 3. 1 part cement, 1 part slaked lime or hydrated lime, not more than 6 parts sand, proportioned by volume. 4. 1 part slaked lime or hydrated lime, not more than 4 parts sand, proportioned by volume. 5. A cellular building unit of burnt clay or concrete, whose minimum gross cubic content is 50% greater than the standard size of brick, and whose cellular spaces exceed 25% of the gross cubic content of the unit. 6. A building unit of burnt clay, or of stone, gravel, or cinder concrete, whose minimum gross cubic content is 50% greater than the standard size of brick, and having no cellular spaces exceeding in the aggregate 25% of the gross cubic content of the unit. 7. 1 part Portland cement, not more than 5 parts coarse aggregate, proportioned by volume. 8. Same limitations as for Portland cement, but using natural cement.

## 4. BEARING WALL CONSTRUCTION

For low buildings, generally less than 4 stories high, exterior bearing walls, with an interior frame of timber, structural steel or reinforced concrete supporting the floor construction, are economical. Dormitories, hospitals, club houses and others designed for comparatively light live loads, fall in this class. Architectural design and local conditions control the choice of the structural frame; buildings 4 or 5 stories high may be built economically by either method. Residences and institutional buildings 1 to 3 stories high almost always are of bearing wall construction.

**MINIMUM THICKNESS OF BEARING WALLS** for business and storage buildings are shown in Figs. 5 and 6; for comparison, non-bearing walls also are shown. Minimum thickness of walls over 100 ft. long between cross walls or piers, 16 in. Maximum height of masonry wall between successive floors, or other substantial lateral support (20 × wall thickness), unless reinforced by adequate cross walls, or piers, at intervals not exceeding (20 × wall thickness). Parapets, usually 12 in. thick, are carried 3 ft. above the roof except in residences.

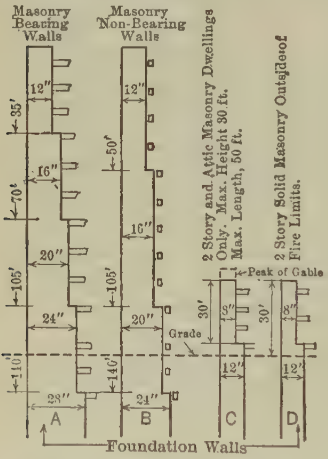

Masonry Bearing Walls
Masonry Non-Bearing Walls
2 Story and Attic Masonry Dwellings Only. Max. Height 30 ft. Max. Length, 50 ft.
2 Story Solid Masonry Outside of Fire Limits.
12″
35′
16″
70′
20″
105′
24″
110′
28″
12″
50′
16″
105′
20″
140′
24″
Peak of Gable
30′
8″
Grade
12″
30′
8″
12″
A
B
C
D
Foundation Walls

FIG. 5. Minimum Wall Thickness for Residence, Institutional and Public Buildings (*Natl. Board of Fire Underwriters*)

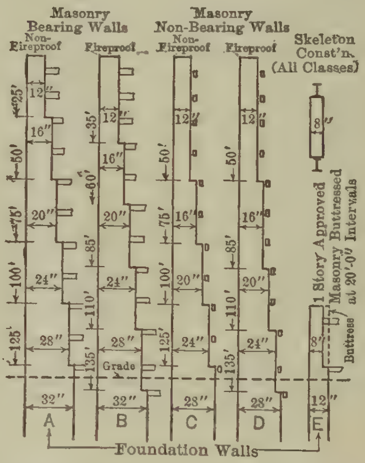


FIG. 6. Minimum Thickness for Business and Storage Buildings (*Natl. Board of Fire Underwriters*)

**WORKING STRESSES.—Natural Stone in Compression,** in lb. per sq. in., may be taken as: Sandstone, 400; marble, 600; limestone, 700; slate, 1000; granite, 1000. **Masonry in Compression.**—See Table 5.

## 5. INDUSTRIAL BUILDINGS

**MULTI-STORY CONSTRUCTION.**—Multi-story buildings for industrial purposes usually are built of reinforced concrete. For structures 3 or more bays wide, with columns located to give approximately square bays of about equal size, the girderless type usually is preferable for superimposed loads of 100 lb. or more per sq. ft. It is more economical than the beam and girder type, except for unusually heavy loads. The ceiling is comparatively flat. Approximately equal column spacing, giving spans of 18 to 22 ft., is economical for average loads from 125 to 300 lb. per sq. ft. Except in warehouses, all exterior wall spaces between columns or piers usually are of glass. Sawtooth roofs may be built if required, but usually are not important in multi-story construction.

Exterior columns usually are of rectangular section; interior columns usually are round, varying from 16 in. to 4 ft. diam., depending on height of building and floor loads. The floor system may be designed with or without drops over the columns; the flared column capital characterizes girderless design and is required by all building ordinances.

Roof surfaces usually are built in the same manner as the floors, if overhead light is not required in the upper story. For drainage, fill is used or the entire area given a slight pitch. Membrane waterproofing often is laid without grade other than that required around parapets and drains.

**SINGLE-STORY CONSTRUCTION.**—Single-story buildings may be of reinforced concrete, but usually are of steel construction. If land is comparatively cheap, the single-story, steel structure has many advantages, *viz.*: no floor space occupied by elevator shafts, stairways and their enclosing walls; no first or operating cost of elevators; less space for columns; better lighting and ventilation; easier supervision of manufacturing processes; a more efficient routing of materials, aiding straight-line production. Also, use of upper floors of a multi-story building is limited by the size and weight of products that can be handled in elevators. The cost of usable floor area in a single-story building often is 25% less per sq. ft. than in a multi-story building of the same gross floor area.

A comparatively recent development (1933) in this field is the controlled conditions plant, without exterior windows; ventilation, air conditioning, and lighting are designed to produce the most desirable working conditions.

**SLOW-BURNING CONSTRUCTION.**—Slow-burning, or mill, construction is seldom used, as reinforced concrete provides greater fire resistance at a relatively small additional cost. Mill construction, originally developed in New England, comprises exterior walls of brick or concrete, with interior framing of heavy timbers carried on wooden posts, supporting floors and roofs of thick planks. Wall openings are supported by masonry arches, or steel or reinforced concrete lintels. Steel lintels over 4 ft. should have 2-in. fireproofing.

**Wooden Columns** should be not less than 8 in. least width; all corners should be rounded or chamfered. Connections between columns and girders should be fire-resistant or protected by materials with a minimum fire-resistance rating of 1 hr.

**Steel or Cast-iron Columns** can be used if protected as required for fire-resistant construction. Stone, cast-iron or steel bases are used under columns resting on floor timbers or masonry foundations.

Beams and girders should be not less than 6 in. least dimension, nor less than 10 in. deep. If built of two or more pieces, sections should be bolted together. Where beams or girders rest on masonry walls, wall plates, boxes of self-releasing type, or suitable hangers should be used. Connections between girders and columns are made by steel or iron straps; girders are fitted closely around columns or butted tightly against them.

**Plank Floors** generally span from girder to girder without cross beams; this design facilitates placing sprinkler pipes and avoids sharp angles to ignite in event of fire. Beams, when used, should rest on top of girders or be supported by steel hangers in which the ends of beams fit closely. If supported on masonry walls, wooden beams and girders should have at least 8 in. of masonry between their ends and the outside face of the wall, and the same thickness between sides of adjacent beams entering a wall from opposite sides. Floors may be tongue-and-groove plank not less than 3 in. thick, with 7/8-in. top flooring laid crosswise or diagonally, or planks not less than 4 in. wide set on edge and spiked together every 18 in. Joints should be broken and located over supports or at the quarter points of the spans. A 1/2-in. space, covered by a molding, between the edges of the floors and walls permits the floor to expand, if wet, without injury to the masonry. All openings or shafts should be protected.

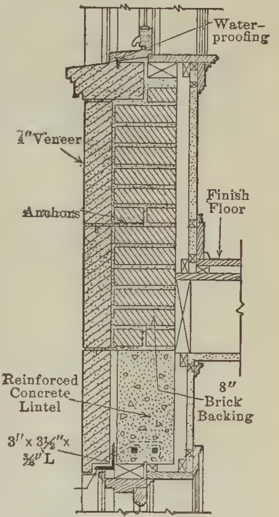


Fig. 7. Stone Veneer with Brick Backing

Roofs are constructed in the same manner as floors, except that plank thickness may be 2 1/2 in. and the roof beams may be 6 in. least dimension. Roof surfaces, covered with membrane waterproofing, usually pitch 1/2 in. per ft.

## 6. WALLS, PIERS AND PARTITIONS

### Stone Work

**MASONRY** of rectangular stones is called **ashlar;** laid in courses with continuous horizontal joints, it is **coursed ashlar;** laid with broken courses and stones of various sizes, it is **broken** or **random ashlar.** **Rubble masonry** is undressed stones of irregular shapes and various sizes. In building construction, moldings, sills, cornice members and ornamental facings of doors and windows are called trimming.

**STONE VENEER,** Fig. 7, usually is 4 in. thick; at least 20% of the face area should be header stones with not less than 4 in. of bond into the backing. Although non-corrodible anchors attach both brick and stone veneer, at least 4 in. of actual bonding, is required to make the veneer part of the legal wall thickness specified by building ordinances.

**MORTAR.**—One part Portland cement, 3 parts sand, 3/20 part hydrated lime, all by volume, generally is used for stone piers, footings, foundations, and heavy masonry below grade. Stone veneer, trim, and exposed work should be set with mortar of non-staining cement, clean, well-graded sand, and hydrated or thoroughly slaked lump lime. Proportions vary for different varieties of stone; an average specification is 1 part non-staining Portland cement, 2 parts sand, and 1/5 part hydrated lime, all by volume. For pointing, the same mixture is used with a fine, white sand. The setting mortar should be applied as parging, 3/4 in. thick, against inner surfaces of all light-colored stone veneers, to prevent staining by the Portland cement used for backing. It is desirable to lay the exterior 4 in. of backing in the non-staining mortar. Black waterproofing compounds, extensively used to protect inside faces of stone veneer, are not a substitute for parging.

**JOINTS** in rubble masonry ordinarily are struck, flush or wiped-out. Ashlar joints usually are raked out 1 in. from the face of the stone and later pointed. Fig. 8 shows various forms of finished joints. All bed joints should be a full bed of mortar, 1/4 in. to 1/2 in. wide. Vertical joints, of the same width, should be thoroughly filled and pointed. Wooden wedges or, preferably, lead buttons give an exact thickness of bed joints.

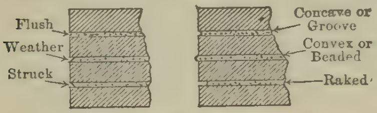


FIG. 8. Types of Brick Joints

**BACKING** for ashlar masonry may be rough stone, common brick, hollow terra cotta or concrete blocks, carried up at the same time as the veneer.

**CLEANING** of marble, limestone, and sandstone should be done with soap and water and bristle brushes, after pointing is finished. Sand blasting, wire brushes, and acids injure the surface.

## Brickwork

Common brick are used for structural walls, piers and for backing stone or terra cotta; well-burned brick are used for surfacing walls. The term *face brick* usually denotes one made for exposed surfaces where a special color, shape, texture or contour is required. Enameled and glazed brick form smooth, impervious surfaces. The Common Brick Mfrs. Assoc. standard size for common brick is 2 1/4 × 3 3/4 × 8 in. Face, enameled, and glazed brick are made in various sizes, approximating these dimensions.

**JOINTS** should be thoroughly filled and each brick laid on a full bed of mortar. For heavy loads, the mortar beds should be smooth, and filled vertical joints are necessary for strength and weather resistance. On purely structural work, mortar joints usually are cut off with a trowel, flush with the wall face, as the work is laid up. On exposed work, joints are finished as for ashlar masonry; the surface of the mortar should be thoroughly compacted to better resist infiltration of rain.

**QUALITY OF BRICK** selected for any particular operation depends on structural or architectural requirements. Well molded units, neither over- nor under-burned give best results. Soft "salmon" brick never should be used. Both highly impervious brick, of low absorption, or highly porous brick having too much absorption, are unsuitable for exterior use. Brick with 5 to 10% absorption will form the most water-tight masonry.

**BONDS** most generally used are Common bond, English and Flemish. Common bond, consisting of 5 or 6 courses of stretchers, followed by a course of headers, is cheapest for ordinary work. In English bond, courses of stretchers and headers alternate. In Flemish bond, headers and stretchers alternate in each course. Face brick may be tied to common brick or hollow tile backing by galvanized iron ties built into the joints, but they then are not part of the legal wall thickness; such veneer should not be considered in computing thickness for stability or bearing.

**MORTAR.**—One part Portland cement, 1 part hydrated lime, 6 parts sand, all by volume, is more desirable for exterior brickwork than mixtures with more cement. Piers or units subject to heavy loads, however, should be laid in mortar of 1 part Portland

**Table 6.—Quantities of Materials Required for 1 Cubic Yard of Mortar**

These quantities allow for loose measurement of aggregates

| Proportions | Cement, bbl. | Sand, cu. yd. | Lime, lb. |
|---|---|---|---|
| 1 Cement: 1 Lime: 6 Sand | 1.25 | 1.1 | 500 |
| 1 Cement: 3 Sand: 10% Lime | 2.50 | 1.1 | 100 |

Mortar required for rubble masonry = 1/3 (total volume of masonry); mortar required for 1000 brick = 17 cu. ft. (3/8-in. joints).

cement, 3 parts sand, 3/20 part hydrated lime, all by volume. Natural cements, also used for brickwork and occasionally for stone masonry, are more plastic than Portland cement; they are mixed 1 part cement, 3 parts, by volume, of sand without lime. Waterproofing compounds added to mortars, particularly those for brick masonry, reduce capillary action. In any case, it is essential to use a well-graded sand and to thoroughly fill the joints; otherwise leaks and efflorescence will result.

**Table 7.—Approximate Quantities of Materials Required for 1 Cubic Yard of Concrete**

These quantities allow for loose measurement of aggregates, and for grouting and waste

| Proportions | Cement, bbl. | Sand, cu. yd. | Crushed Stone, cu. yd. |
|---|---|---|---|
| 1 Cement : 2 Sand : 4 Stone .................. | 1.5 | 0.45 | 0.9 |
| 1 Cement : 1 1/2 Sand : 3 Stone .............. | 2.0 | 0.45 | 0.9 |

## Terra Cotta Tile

**ARCHITECTURAL TERRA COTTA TILE**, used as veneer, are from 12 to 30 in. wide, usually 4 in. thick, and of a height determined by the architectural design. The blocks are hollow and without backs, comprising an outer shell braced by partitions about 6 in. apart. They are attached to the backing by non-corrodible anchors; the joints are finished with cement mortar. Sills, copings, cornices, other architectural members, and veneer, are designed and fitted by the manufacturers for each individual job. Adequate support and thorough drainage are the most essential factors controlling character of the work.

**STRUCTURAL TERRA COTTA TILE** are hollow blocks open at two ends, with interior webs 7/8 in. thick, dividing the units into longitudinal cells. Exterior surfaces are smooth, or scored to give a key for stucco or plaster. Load-bearing tile, for exterior walls and bearing partitions of light buildings, and partition tile for interior non-bearing partitions, are made 12 in. wide and 12 in. high; thickness varies from 3 to 12 in. A variety of other designs, as shoe-tile and backer-tile, are available for backing either stone or brick veneer in load-bearing or curtain-wall construction.

Besides the various varieties of floor tile, fireproofing tile and book-tile (see Floor and Roof construction, p. 17–24), **terra cotta furring tile** is used to line inside surfaces of exterior masonry walls. These usually are 3 or 4 in. thick, scored by the manufacturers, and split to 1 1/2 or 2 in. thick. When placed against an exterior wall, the voids between the ribs provide a dead air space, forming insulation against cold or dampness. The ribs are set vertically in mortar, and held in place by metal ties or 10d. wire nails bent down over the tops of the tile. Partition blocks, 3 in. thick, tied to the wall by metal anchors, also are used for furring.

**LOAD-BEARING TILE** are used for exterior walls of residences, small stores and garages; most building codes prohibit them in walls over 40 ft. high. The cells are placed vertically, and special tile are used for jambs, sills, and corners. Mortar is the same as for brick masonry.

**BACKER TILE** of various designs is used for veneered bearing walls and for structural portions of curtain walls in skeleton construction. They weigh less than common brick, lay more quickly, and furnish better insulation, thereby eliminating furring.

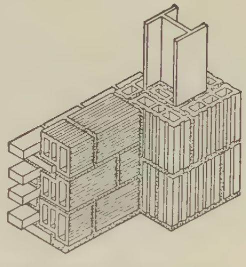


Fig. 9. Brick Veneer with Tile Backing. Note Column Fireproofing

**PARTITION TILE** are lighter and generally more porous than load-bearing tile, and are used for interior partitions in fire-resistant buildings. A 6- or 8-in. tile usually is required for stair or elevator enclosures; 4-in. tile for corridors; 3-in. tile for room partitions. Salt-glazed and enameled surfaces also are made, as well as plain and scored surfaces.

## Concrete Blocks and Cast Stone

**HOLLOW CONCRETE BLOCKS** are made both with sand and cinder aggregates. The latter is more widely used, as units weigh less, are better insulators, are more nailable, and make a better bond when used as a base for exterior stucco.

Concrete blocks are made in various sizes; a typical unit is 8 × 8 × 16 in. When used for exterior walls of light buildings, they usually are stuccoed on the exterior; when used for other than garages or machine shops, interior surfaces should be furred. Cinder blocks are used for backing panel wall construction and for interior partitions. The shape is adapted to each particular use. As all cement products shrink during hardening, they should be at least 28 days old before use. Both cement-sand and cement-cinder units are laid in the same mortar as brick masonry.

**CAST STONE** is used architecturally as a cheaper substitute for natural stones and to produce color and texture effects impossible with natural products. It usually is composed of either white cement or standard Portland cement, mixed with appropriate aggregate to the consistency of thin concrete, and poured into sand, wood, metal, plaster or glue molds. Dry pigments give the desired color.

Rectangular blocks, for veneer of buildings and molded units of simpler shape, are cast in sand molds, using strips of wood or metal embedded in the sand to form curved surfaces. The more complicated shapes are cast in plaster molds; glue, or a similar elastic composition, is required for undercut work. If the surface color or texture requires a costly mixture, the body of the unit is made of cheaper materials, otherwise the entire member is of the same composition. After hardening and curing, the surface film of cement is removed by fine water spray or acids. To avoid surface cracking or crazing and the shrinkage of cast stone products after placement, only the best materials should be used; the mixture and surface treatment should insure proper weathering; and the units must be thoroughly cured before use.

### Gypsum Tile

Besides the use of gypsum tile in floors and roofs, gypsum is used for partitions, furring and fireproofing. The tile are either solid or hollow; the latter have circular cell spaces.

**PARTITION TILE** generally are made 12 in. high, 30 in. long and 2 to 6 in. thick. Gypsum tile are lighter than terra cotta, are sawed easily, and offer better nailability.

**FURRING TILE** are made 3 and 4 in. thick. They are scored by the makers and split to 1 1/2 or 2 in. thick. They are set the same as terra cotta furring.

**FIREPROOFING TILE** are made in shapes to fit the lower flanges of I-beams. Both soffit and angle tile are made for fireproofing; solid and hollow partition tile are used to insulate structural steel members.

**MORTAR** for all gypsum units should be 1 part gypsum plaster, 2 or 3 parts, by weight, of clean sand. Portland cement mortar should not be used with gypsum blocks.

**GYPSUM PRODUCTS** should not be used where moisture is present. When dry, they are excellent for non-bearing partitions, but have not the structural strength of terra cotta.

### Metal Lath and Plaster Board Partitions

**FIRE-RESISTANT PARTITIONS** may be built by supporting metal lath on a light, rolled-steel framework and applying two coats of Portland cement mortar to each side. This design is useful for enclosing small areas, as office spaces, in certain types of industrial buildings. The supports are bolted to floor and ceiling; total thickness approximates 2 in.; 2-in. channels are used for door frames.

**PLASTER BOARDS,** composed chiefly of gypsum, and many types of fiber boards, may be used as a base for gypsum plaster; in some cases no surface treatment is given. Most of these are fire-resistant, but the actual fire rating allowed by the Underwriters should be checked for each particular construction before use.

## 7. FLOORS

About 20 systems of fire-resistant floor construction are in use. Each has its advantages and limitations, and the system best adapted to the particular building should be chosen. Certain types are suitable only for reinforced-concrete framing; others are adapted only to structural steel. The type of occupancy being known, the floor loads may be determined from the local building ordinances. Floor loads will dictate the floor construction, some systems being adapted to light and others to heavy loads. Many designs are suitable only for short-span construction; some are one-way systems only; others are either one- or two-way systems; these features control their adaptability to the architectural plan. Supplementing purely structural features, final choice between acceptable designs usually will depend on economic conditions.

### Types of Fire-resistant Floor Construction

In addition to the various types of reinforced concrete, hollow tile, gypsum and concrete block systems of floors (see below), the metal pan system generally is used in both steel-frame and reinforced-concrete buildings. The metal dome system has been used with reinforced concrete framing. Metal lumber and open-web steel joists are used in buildings for light occupancy; steel plate designs in certain types of industrial work. They are not adapted to a reinforced concrete frame. Concretes with light-weight aggregates, as Haydite, and light-weight concretes, as Aerocrete and Porete, and gypsum, are precast into slabs supported by structural steel in short span construction; slabs are poured in place on steel-frame buildings.

**THE GIRDERLESS REINFORCED CONCRETE FLOOR** is particularly adapted to multi-story industrial work. For design procedure, see Eshbach, Handbook of Engineering Fundamentals, section 5, forming Vol. 1 of this series.

**THE BEAM-AND-SLAB SYSTEM** usually costs more than girderless designs. It is used for comparatively heavy live loads where an economical column spacing is impracticable, or where flared column heads are objectionable. This system may be used with reinforced concrete frames, bearing wall construction, or structural steel framing.

Floor systems in steel-frame, multi-story buildings are: 1. Short-span construction, including cinder concrete arches and flat or segmental terra cotta arches; 2. Long-span construction.

**CINDER CONCRETE ARCHES** usually span between I-beams framed into girders at $^1/_3$ points of girder spans. Usual floor thickness is 4 in. on clear spans up to 8 ft.; corresponding roof thickness is 3 $^1/_2$ in. Concrete mixture by volume is 1 part Portland cement, 2 parts sand, 5 parts clean, well burned steam-boiler cinders. Wire fabric reinforcement is used, based on empirical data. See Table 8. According to the New York City ordinance, $L \leqq \{(h \times A)/S^2\} \times c$, where $L$ = gross load on floor, lb. per sq. ft.; $h$ = depth of reinforcement below top of slab, in.; $A$ = cross-sectional area of tension steel, sq. in. per ft. width; $S$ = span, ft.; $c$ = constant whose values are as follows for cinder concrete: 14,000 for reinforcement not continuous over supports; 18,000 for reinforcement of rods or other shapes securely hooked over or attached to supports; 26,000 for steel fabric reinforcement continuous over supports; for stone concrete the corresponding values of $c$ are 16,000, 20,000 and 30,000 respectively. If permitted by building ordinances, cinder-arch construction is economical for live loads up to 75 lb. per sq. ft.; it is used only between structural steel beams.

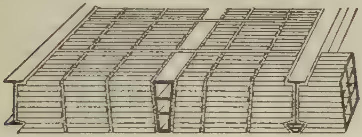


Fig. 10. Typical Section End Construction Hollow Tile Flat Floor Arch. (*Natl. Fireproofing Co.*)

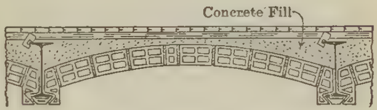


Fig. 11. Typical Section Segmental Hollow Tile Floor Arch (*Natl. Fireproofing Co.*)

**HOLLOW TILE ARCHES** usually are of shorter span than cinder concrete arches. They are supported by I-beams framed into girders at $^1/_3$ or $^1/_4$ points of girder spans. This construction is excellent for hotels and office buildings, permits alteration more easily than competing long-span systems, but first cost is higher. Flat hollow tile arches use tile from 6 to 16 in. deep, depending on load and span. Minimum depth is 1 $^1/_2$ in. per foot of clear span, but never less than 6 in. Lower flanges of supporting members are

**Table 8.—Area of Steel Wire Reinforcement for 4-in. Cinder Concrete Arches**

Load uniformly distributed. Spans limited to 8 ft., 0 in. Concrete mix: 1 cement : 2 sand: 5 cinders, by volume. Reinforcement, steel fabric, continuous over supports, placed 1 in. from bottom of slab. Weight of 4-in. slab = 36 lb. per sq. ft. Values are based on empirical data approved by New York City Building Dept. for short-span construction.

| Safe Superimposed Load, lb. per sq. ft. | Spans, feet and inches | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 4′ 0″ | 4′ 6″ | 5′ 0″ | 5′ 6″ | 6′ 0″ | 6′ 6″ | 7′ 0″ | 7′ 6″ | 8′ 0″ |
| | Steel Areas, sq. in. per foot width | | | | | | | | |
| 50 | 0.043 | 0.043 | 0.043 | 0.043 | 0.043 | 0.047 | 0.054 | 0.062 | 0.071 |
| 75 | .043 | .043 | .043 | .043 | .051 | .060 | .070 | .080 | .091 |
| 100 | .043 | .043 | .044 | .053 | .063 | .074 | .086 | .098 | .112 |
| 125 | .043 | .043 | .052 | .063 | .074 | .087 | .101 | .116 | .132 |
| 150 | .043 | .048 | .060 | .072 | .086 | .101 | .117 | .134 | .153 |
| 175 | .043 | .055 | .068 | .082 | .097 | .114 | .133 | .152 | .173 |
| 200 | .048 | .061 | .076 | .092 | .109 | .128 | .148 | .170 | .194 |
| 225 | .054 | .068 | .084 | .101 | .121 | .141 | .164 | .188 | .214 |
| 250 | .059 | .074 | .092 | .111 | .132 | .155 | .180 | .206 | .235 |
| 275 | .064 | .081 | .100 | .121 | .144 | .169 | .195 | .224 | .255 |
| 300 | .069 | .087 | .108 | .130 | .155 | .182 | .211 | .242 | .276 |

### Table 9.—Safe Loads for End-construction Hollow Tile Flat Floor Arches

Loads shown in table are safe superimposed loads. Net sectional areas of tile are for keys and are taken per foot of tile parallel to beams. The system is Natco combination arch of side construction skews and keys, with end construction inters, made by National Fireproofing Co.

| Depth of Arch | 6 in. | 7 in. | 8 in. | 9 in. | 10 in. | 12 in. | 15 in. |
|---|---|---|---|---|---|---|---|
| Net Sectional Areas, sq. in. | 27 | 27 | 27 | 27 | 36 | 36 | 36 |
| Wt. of Arch, lb. per sq. ft. | 26 | 30 | 32 | 36 | 40 | 48 | 56 |
| Span | Safe Load, Uniformly Distributed, lb. per sq. ft. | | | | | | |
| 4 ft –0 in. | 210 | 246 | 283 | 318 | 485 | 582 | 730 |
| 4 –3 | ..... | 214 | 247 | 278 | 425 | 510 | 641 |
| 4 –6 | ..... | 188 | 217 | 243 | 375 | 449 | 566 |
| 4 –9 | ..... | ..... | 191 | 215 | 332 | 399 | 502 |
| 5 –0 | ..... | ..... | 169 | 191 | 296 | 354 | 448 |
| 5 –3 | ..... | ..... | 150 | 169 | 265 | 317 | 401 |
| 5 –6 | ..... | ..... | ..... | 151 | 237 | 285 | 361 |
| 5 –9 | ..... | ..... | ..... | 135 | 214 | 257 | 325 |
| 6 –0 | ..... | ..... | ..... | 121 | 193 | 232 | 294 |
| 6 –3 | ..... | ..... | ..... | ..... | 174 | 210 | 266 |
| 6 –6 | ..... | ..... | ..... | ..... | 158 | 190 | 242 |
| 6 –9 | ..... | ..... | ..... | ..... | ..... | 173 | 220 |
| 7 –0 | ..... | ..... | ..... | ..... | ..... | 158 | 201 |
| 7 –6 | ..... | ..... | ..... | ..... | ..... | 130 | 167 |
| 8 –0 | ..... | ..... | ..... | ..... | ..... | 109 | 141 |

### Table 10.—Safe Loads, for Segmental Hollow Tile Floor Arches

Loads shown in table are safe superimposed loads, uniformly distributed. Values apply to tile as made by National Fireproofing Co., having the following sectional areas per foot of arch parallel with beams: 4-in. arch, 28 sq. in.; 6-in. arch, 36 sq. in.; 8-in. arch, 43 sq. in.

| Span | Rise of Arch, in. per ft. | Safe Load, lb. per sq. ft. | | Span | Rise of Arch, in. per ft. | Safe Load, lb. per sq. ft. | | Span | Rise of Arch, in. per ft. | Safe Load, lb. per sq. ft. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 6-in. Arch, Wt., 30 lb. per sq. ft. | 8-in. Arch, Wt., 36 lb. per sq. ft. | | | 6-in. Arch, Wt., 30 lb. per sq. ft. | 8-in. Arch, Wt., 36 lb. per sq. ft. | | | 6-in. Arch, Wt., 30 lb. per sq. ft. | 8-in. Arch, Wt., 36 lb. per sq. ft. |
| 4 ft. 0 in. | 3/4 | 902 | 1078 | 6 ft. 6 in. | 3/4 | 551 | 658 | 9 ft. 0 in. | 3/4 | 386 | 461 |
| | 1 | 1184 | 1414 | | 1 | 724 | 864 | | 1 | 518 | 619 |
| | 1 1/4 | 1485 | 1774 | | 1 1/4 | 902 | 1077 | | 1 1/4 | 645 | 770 |
| | 1 1/2 | 1740 | 2079 | | 1 1/2 | 1058 | 1264 | | 1 1/2 | 758 | 906 |
| | 1 3/4 | 1986 | 2373 | | 1 3/4 | 1218 | 1455 | | 1 3/4 | 871 | 1041 |
| | 2 | 2233 | 2667 | | 2 | 1358 | 1622 | | 2 | 977 | 1167 |
| 4 ft. 6 in. | 3/4 | 792 | 946 | 7 ft. 0 in. | 3/4 | 508 | 606 | 10 ft. 0 in. | 3/4 | 344 | 411 |
| | 1 | 1044 | 1247 | | 1 | 669 | 799 | | 1 | 462 | 552 |
| | 1 1/4 | 1313 | 1568 | | 1 1/4 | 834 | 996 | | 1 1/4 | 576 | 688 |
| | 1 1/2 | 1539 | 1838 | | 1 1/2 | 981 | 1171 | | 1 1/2 | 683 | 816 |
| | 1 3/4 | 1775 | 2121 | | 1 3/4 | 1127 | 1346 | | 1 3/4 | 784 | 937 |
| | 2 | 1975 | 2359 | | 2 | 1264 | 1510 | | 2 | 879 | 1050 |
| 5 ft. 0 in. | 3/4 | 709 | 847 | 7 ft. 6 in. | 3/4 | 471 | 563 | 11 ft. 0 in. | 3/4 | 315 | 376 |
| | 1 | 957 | 1143 | | 1 | 621 | 741 | | 1 | 421 | 503 |
| | 1 1/4 | 1172 | 1400 | | 1 1/4 | 774 | 925 | | 1 1/4 | 519 | 621 |
| | 1 1/2 | 1379 | 1647 | | 1 1/2 | 920 | 1099 | | 1 1/2 | 617 | 737 |
| | 1 3/4 | 1592 | 1902 | | 1 3/4 | 1049 | 1253 | | 1 3/4 | 709 | 847 |
| | 2 | 1773 | 2118 | | 2 | 1176 | 1405 | | 2 | 794 | 948 |
| 5 ft. 6 in. | 3/4 | 641 | 766 | 8 ft. 0 in. | 3/4 | 439 | 525 | 12 ft. 0 in. | 3/4 | 285 | 341 |
| | 1 | 864 | 1032 | | 1 | 588 | 703 | | 1 | 383 | 458 |
| | 1 1/4 | 1062 | 1269 | | 1 1/4 | 724 | 864 | | 1 1/4 | 477 | 569 |
| | 1 1/2 | 1266 | 1512 | | 1 1/2 | 859 | 1026 | | 1 1/2 | 566 | 676 |
| | 1 3/4 | 1439 | 1719 | | 1 3/4 | 987 | 1179 | | 1 3/4 | 649 | 776 |
| | 2 | 1619 | 1933 | | 2 | 1099 | 1312 | | 2 | 727 | 869 |
| 6 ft. 0 in. | 3/4 | 585 | 699 | 8 ft. 6 in. | 3/4 | 411 | 491 | | | | |
| | 1 | 788 | 941 | | 1 | 551 | 658 | | | | |
| | 1 1/4 | 969 | 1157 | | 1 1/4 | 678 | 810 | | | | |
| | 1 1/2 | 1154 | 1379 | | 1 1/2 | 806 | 963 | | | | |
| | 1 3/4 | 1315 | 1570 | | 1 3/4 | 926 | 1106 | | | | |
| | 2 | 1476 | 1763 | | 2 | 1037 | 1239 | | | | |

fireproofed by special soffit blocks. Segmental hollow tile arches, and occasionally brick arches, are used for heavy loads; both systems have been largely superseded by more economical designs. Segmental arches require tie rods just below centers of adjacent beams, and in line across a floor area. For hollow tile arches, in interior bays, sizes and spacings are as follows.

| | | | |
|---|---|---|---|
| Span, ft | 6 or less | 6–7 | 7–9 |
| Diam. of rod, in. | $^3/_4$ | $^7/_8$ | $^7/_8$ |
| Center distance of rods, ft | 5 | 5 | 4 |

**LONG-SPAN CONSTRUCTION,** comprising essentially reinforced concrete joists separated by terra cotta, concrete or gypsum blocks or by voids formed by wood or metal cores, may be used with bearing walls, reinforced concrete or structural steel framing. Fillers usually are 12 to 20 in. wide; joists 4 to 5 in., spaced 16 to 25 in. centers. A continuous slab, or topping, 2 to 3 in. thick forms the floor surface. These systems are used for buildings other than industrial, carrying live loads of 40 to 125 lb. per sq. ft. They support the floor loads of an entire bay, and intermediate beams are unnecessary. Dead load is also much less than with solid concrete slabs.

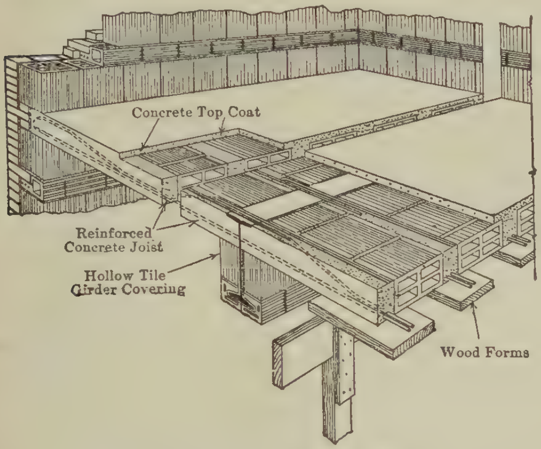


FIG. 12. Long-span Hollow Tile Construction (*Natl. Fireproofing Co.)*

The use of filler block (terra cotta, concrete or gypsum) gives a more substantial floor than when joists are separated by voids. Block systems are heavier and more costly than metal or wooden pan designs, but permit plaster to be applied directly to soffits of the blocks; otherwise metal lath must be used for a flat ceiling across soffits of the ribs.

Both terra cotta and concrete blocks are used as fillers in two-way systems. Such designs, comprising two sets of reinforced concrete joists, parallel to the sides of the panels. are economical for square or nearly square rectangular bays, *i.e.*—(longer side ÷ shorter side $\overset{=}{<}$ 1.25). Metal pans also are made in domes adapted to two-way design. They are used only in reinforced concrete buildings.

**PRESSED STEEL SECTIONS** (metal lumber) are used in buildings for light occupancy, where great fire resistance is not required. These sections, normally 8 to 16 in. deep, supported by steel girders spanning between columns, are used on 12 to 30 in. centers to carry a thin concrete floor slab, usually 2 $^1/_2$ in. thick, poured over a closely spaced metal lath which acts both as forms and reinforcement.

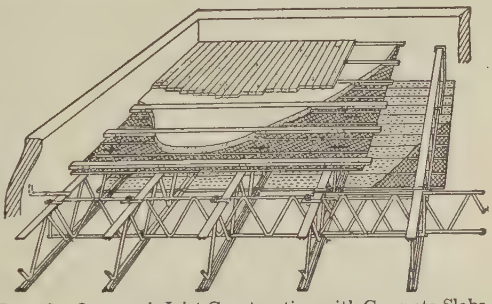


FIG. 13. Open-web Joist Construction with Concrete Slabs, Sleepers and Wood Flooring (*Concrete Steel Co.)*

**OPEN-WEB STEEL JOISTS** have largely superseded pressed steel sections. They are used in the same manner and are satisfactory for short, medium and comparatively long spans for light live loads, and where the plastered lath used as insulation for the soffits of the joists furnishes sufficient fire resistance.

**Open-web Steel Joists and Structural Aerocrete Construction,** topped with stone concrete is economical for light and medium live loads on spans up to 25 ft. It can be quickly

(*Continued on page* 17–32)

### Table 11.—Core Depth and Steel Area for Long-span Metal Core Construction

Cores 25 in. base; 5 in. reinforced concrete joists, 30 in. on centers; 2 1/2-in. concrete top. Compressive unit stress in extreme fibers of concrete = $f_c$ = 700 lb. per sq. in. Tensile unit stress in steel = $f_s$ = 18,000 lb. per sq. in. For safe loads using 16,000 lb. per sq. in. in steel, and 650 lb. per sq. in. in concrete, multiply loads given by 0.89, and subtract from the result the factors shown in each section of the table. Loads to left of first zigzag line produce shears under 40 lb. per sq. in. Loads between zigzag lines require tapered-end cores for shear and for compression over continuous supports. Loads to right of second zigzag line develop higher stresses and should be investigated. Data were supplied by Berger Mfg. Co.

| | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | $wl^2/12$ | | | | | | | | |
| Steel Area, sq. in.... Size & No. of bars.. | 0.22<br>2 3/8″ ○ | 0.30<br>1 1/2″ ○<br>1 3/8″ ○ | 0.40<br>2 1/2″ ○ | 0.45<br>1 1/2″ □<br>1 1/2″ ○ | 0.50<br>2 1/2″ □ | 0.60<br>2 5/8″ ○ | 0.75<br>1 3/4″ ○<br>1 5/8″ ○ | 0.88<br>2 3/4″ ○ | 1.04*<br>1 7/8″ ○<br>1 3/4″ ○ | 1.20*<br>2 7/8″ ○ | 1.38<br>1 1″ ○ | 1.50<br>2 1″ ○ | 1.67<br>1 1″ □<br>1 1″ ○ | 1.70<br>1 1″ □<br>1 1″ ○ | 2.0<br>2 1″ □ |
| | | | | | | | $wl^2/10$ | | | | | | | | |
| Steel Area, sq. in.... Size & No. of bars.. | 0.26<br>1 1/2″ ○<br>1 3/8″ ○ | 0.36<br>2 1/2″ ○ | 0.48<br>2 1/2″ □ | 0.54<br>1 5/8″ ○<br>1 1/2″ □ | 0.60<br>2 5/8″ ○ | 0.72<br>1 3/4″ ○<br>1 5/8″ ○ | 0.90<br>2 3/4″ ○ | 1.05*<br>1 7/8″ ○<br>1 3/4″ ○ | 1.25*<br>2 7/8″ ○ | 1.44†<br>2 1″ ○ | 1.66‡<br>1 1″ □<br>1 1″ ○ | 1.80‡<br>1 1″ □<br>1 1″ ○ | ......... | 2.04<br>2 1″ □ | ......... |
| | | | | | | | $wl^2/8$ | | | | | | | | |
| Steel Area, sq. in.... Size & No. of bars.. | 0.33<br>1 1/2″ ○<br>1 3/8″ ○ | 0.45<br>1 1/2″ □<br>1 1/2″ ○ | 0.60<br>2 5/8″ ○ | 0.68<br>1 3/4″ ○<br>1 1/2″ □ | 0.75<br>1 3/4″ ○<br>1 5/8″ ○ | 0.90<br>2 3/4″ ○ | 1.13<br>2 7/8″ ○ | 1.32<br>1 1″ ○<br>1 7/8″ ○ | 1.56<br>2 1″ ○ | 1.80<br>1 1″ □ | 2.07<br>1 1 1/8″ □<br>1 1″ ○ | ......... | ......... | ......... | ......... |
| Span, ft. | Depth of core, 6 in. Dead load per sq. ft. = 44 lb. Safe superimposed loads, uniformly distributed, lb. per sq. ft. | | | | | | | | | | | | | | |
| 8 | 117 | 171 | 243 | 278 | 314 | 385 | ......... | ......... | ......... | ......... | ......... | ......... | ......... | ......... | ......... |
| 9 | 83 | 125 | 183 | 210 | 238 | 296 | ......... | ......... | ......... | ......... | ......... | ......... | ......... | ......... | ......... |
| 10 | 58 | 93 | 139 | 162 | 185 | 231 | 293 | 353 | 416 | ......... | ......... | ......... | ......... | ......... | ......... |
| 11 | 40 | 69 | 108 | 126 | 145 | 183 | 235 | 283 | 336 | ......... | ......... | ......... | ......... | ......... | ......... |
| 12 | 28 | 51 | 83 | 98 | 115 | 147 | 190 | 231 | 275 | ......... | ......... | ......... | ......... | ......... | ......... |
| 13 | ......... | 38 | 64 | 78 | 91 | 118 | 156 | 191 | 228 | ......... | ......... | ......... | ......... | ......... | ......... |
| 14 | ......... | 26 | 49 | 61 | 73 | 96 | 128 | 158 | 191 | ......... | ......... | ......... | ......... | ......... | ......... |
| 15 | ......... | ......... | 38 | 48 | 58 | 78 | 106 | 133 | 160 | ......... | ......... | ......... | ......... | ......... | ......... |
| 16 | ......... | ......... | 28 | 37 | 45 | 63 | 88 | 111 | 135 | ......... | ......... | ......... | ......... | ......... | ......... |
| 17 | ......... | ......... | ......... | ......... | 35 | 51 | 73 | 93 | 114 | ......... | ......... | ......... | ......... | ......... | ......... |
| 18 | ......... | Factor, 4.9 | | ......... | 27 | 41 | 60 | 78 | 98 | ......... | ......... | ......... | ......... | ......... | ......... |
| 19 | ......... | | | ......... | ......... | ......... | 49 | 66 | 83 | ......... | ......... | ......... | ......... | ......... | ......... |
| 20 | ......... | ......... | ......... | ......... | ......... | ......... | 40 | 54 | 71 | ......... | ......... | ......... | ......... | ......... | ......... |

| Span, ft. | Depth of core, 8 in. Dead load per sq. ft. = **49** lb. Safe superimposed loads, uniformly distributed, lb. per sq. ft. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 81 | 126 | 184 | 214 | 243 | 302 | 383 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 11 | 58 | 96 | 143 | 168 | 193 | 241 | 309 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 12 | 42 | 73 | 113 | 133 | 154 | 195 | 252 | 303 | 362 | ........ | ........ | ........ | ........ | ........ | ........ |
| 13 | 28 | 55 | 89 | 107 | 124 | 158 | 208 | 251 | 301 | ........ | ........ | ........ | ........ | ........ | ........ |
| 14 | 17 | 41 | 70 | 85 | 101 | 130 | 172 | 210 | 253 | 299 | ........ | ........ | ........ | ........ | ........ |
| 15 | ........ | 28 | 55 | 68 | 81 | 107 | 143 | 177 | 213 | 254 | ........ | ........ | ........ | ........ | ........ |
| 16 | ........ | 19 | 43 | 54 | 66 | 88 | 120 | 149 | 182 | 218 | ........ | ........ | ........ | ........ | ........ |
| 17 | ........ | ........ | 32 | 42 | 53 | 72 | 101 | 126 | 156 | 188 | ........ | ........ | ........ | ........ | ........ |
| 18 | ........ | ........ | 23 | 32 | 41 | 59 | 84 | 108 | 133 | 162 | ........ | ........ | ........ | ........ | ........ |
| 19 | ........ | ........ | ........ | 24 | 32 | 48 | 71 | 91 | 115 | 140 | ........ | ........ | ........ | ........ | ........ |
| 20 | ........ | Factor, 5.4 | | ........ | 24 | 39 | 59 | 78 | 99 | 122 | ........ | ........ | ........ | ........ | ........ |
| 21 | ........ | | | ........ | 18 | 30 | 48 | 67 | 85 | 105 | ........ | ........ | ........ | ........ | ........ |
| 22 | ........ | ........ | ........ | ........ | ........ | 23 | 41 | 56 | 73 | 92 | ........ | ........ | ........ | ........ | ........ |
| 23 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 63 | 80 | ........ | ........ | ........ | ........ | ........ |

| Span, ft. | Depth of core, 10 in. Dead load per sq. ft. = **54** lb. Safe superimposed loads, uniformly distributed, lb. per sq. ft. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 13 | ........ | 73 | 114 | 141 | 157 | 198 | 258 | 312 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 14 | ........ | 55 | 92 | 115 | 128 | 163 | 215 | 262 | 315 | 372 | ........ | ........ | ........ | ........ | ........ |
| 15 | ........ | 41 | 73 | 94 | 104 | 135 | 180 | 221 | 268 | 316 | ........ | ........ | ........ | ........ | ........ |
| 16 | ........ | 29 | 57 | 77 | 85 | 113 | 152 | 188 | 228 | 272 | 321 | 353 | ........ | ........ | ........ |
| 17 | ........ | 19 | 44 | 63 | 69 | 93 | 128 | 160 | 196 | 234 | 278 | 307 | ........ | ........ | ........ |
| 18 | ........ | ........ | 34 | 50 | 55 | 78 | 108 | 137 | 169 | 203 | 242 | 268 | ........ | ........ | ........ |
| 19 | ........ | ........ | 25 | 40 | 44 | 64 | 93 | 117 | 146 | 177 | 212 | 235 | ........ | ........ | ........ |
| 20 | ........ | ........ | 18 | 32 | 34 | 53 | 78 | 101 | 127 | 154 | 186 | 207 | ........ | ........ | ........ |
| 21 | ........ | ........ | ........ | 24 | 26 | 43 | 65 | 86 | 110 | 135 | 163 | 183 | ........ | ........ | ........ |
| 22 | ........ | ........ | ........ | ........ | 19 | 33 | 55 | 73 | 95 | 118 | 144 | 162 | ........ | ........ | ........ |
| 23 | ........ | ........ | ........ | ........ | ........ | 27 | 46 | 63 | 83 | 103 | 128 | 143 | ........ | ........ | ........ |
| 24 | ........ | Factor, 6.0 | | ........ | ........ | ........ | 38 | 53 | 72 | 91 | 112 | 127 | ........ | ........ | ........ |
| 25 | ........ | | | ........ | ........ | ........ | 30 | 45 | 61 | 79 | 99 | 113 | ........ | ........ | ........ |
| 26 | ........ | ........ | ........ | ........ | ........ | ........ | 23 | 38 | 53 | 69 | 88 | 100 | ........ | ........ | ........ |
| 27 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 45 | 60 | 78 | 89 | ........ | ........ | ........ |

*See footnotes at end of table.*

*Table continued on following page.*

**Table 11.—Core Depth and Steel Area for Long-span Metal Core Construction**—*Continued*

| | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $wl^2/12$ | | | | | | | | | | | | | | |
| Steel Area, sq. in.... | 0.22 | 0.30 | 0.40 | 0.45 | 0.50 | 0.60 | 0.75 | 0.88 | 1.04* | 1.20* | 1.38 | 1.50 | 1.67 | 1.70 | 2.0 |
| Size & No. of bars. . | 2 3/8″ ○ | 1 1/2″ ○<br>1 3/8″ ○ | 2 1/2″ ○ | 1 1/2″ □<br>1 1/2″ ○ | 2 1/2″ □ | 2 5/8″ ○ | 1 3/4″ ○<br>1 5/8″ ○ | 2 3/4″ ○ | 1 7/8″ ○<br>1 3/4″ ○ | 2 7/8″ ○ | 1 1″ ○ | 2 1″ ○ | 1 1″ □<br>1 1″ ○ | 1 1″ □<br>1 1″ ○ | 2 1″ □ |
| | $wl^2/10$ | | | | | | | | | | | | | | |
| Steel Area, sq. in.... | 0.26 | 0.36 | 0.48 | 0.54 | 0.60 | 0.72 | 0.90 | 1.05* | 1.25* | 1.44† | 1.66‡ | 1.80‡ | ........ | 2.04 | ........ |
| Size & No. of bars. . | 1 1/2″ ○<br>1 3/8″ ○ | 2 1/2″ ○ | 2 1/2″ □ | 1 5/8″ ○<br>1 1/2″ □ | 2 5/8″ ○ | 1 3/4″ ○<br>1 5/8″ ○ | 2 3/4″ ○ | 1 7/8″ ○<br>1 3/4″ ○ | 2 7/8″ ○ | 2 1″ ○ | 1 1″ □<br>1 1″ ○ | 1 1″ □<br>1 1″ ○ | ........ | 2 1″ □ | ........ |
| | $wl^2/8$ | | | | | | | | | | | | | | |
| Steel Area, sq. in.... | 0.33 | 0.45 | 0.60 | 0.68 | 0.75 | 0.90 | 1.13 | 1.32 | 1.56 | 1.80 | 2.07 | ........ | ........ | ........ | ........ |
| Size & No. of bars. . | 1 1/2″ ○<br>1 3/8″ ○ | 1 1/2″ □<br>1 1/2″ ○ | 2 5/8″ ○ | 1 3/4″ ○<br>1 1/2″ □ | 1 3/4″ ○<br>1 5/8″ ○ | 2 3/4″ ○ | 2 7/8″ ○ | 1 1″ ○<br>1 7/8″ ○ | 2 1″ ○ | 1 1″ □ | 1 1 1/8″ □<br>1 1″ ○ | ........ | ........ | ........ | ........ |
| Span, ft. | **Depth of core, 12 in. Dead load per sq. ft. = 59 lb.** Safe superimposed loads, uniformly distributed, lb. per sq. ft. | | | | | | | | | | | | | | |
| 13 | ........ | 89 | 138 | 164 | 188 | 238 | 309 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 14 | ........ | 68 | 112 | 133 | 155 | 198 | 259 | 314 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 15 | ........ | 52 | 90 | 108 | 128 | 164 | 218 | 266 | 321 | ........ | ........ | ........ | ........ | ........ | ........ |
| 16 | ........ | 38 | 72 | 88 | 104 | 137 | 184 | 227 | 274 | 326 | 384 | 424 | 476 | ........ | ........ |
| 17 | ........ | 28 | 56 | 72 | 86 | 114 | 157 | 194 | 237 | 282 | 333 | 367 | 415 | ........ | ........ |
| 18 | ........ | 18 | 44 | 58 | 70 | 96 | 133 | 167 | 204 | 244 | 291 | 321 | 363 | ........ | ........ |
| 19 | ........ | ........ | 33 | 46 | 57 | 80 | 113 | 143 | 178 | 213 | 255 | 283 | 320 | ........ | ........ |
| 20 | ........ | ........ | 24 | 35 | 46 | 67 | 97 | 123 | 155 | 188 | 224 | 249 | 283 | ........ | ........ |
| 21 | ........ | ........ | 18 | 26 | 35 | 54 | 82 | 107 | 135 | 165 | 198 | 221 | 252 | ........ | ........ |
| 22 | ........ | ........ | ........ | 18 | 28 | 45 | 70 | 93 | 118 | 145 | 175 | 195 | 224 | ........ | ........ |
| 23 | ........ | ........ | ........ | ........ | 20 | 36 | 58 | 78 | 102 | 128 | 155 | 174 | 200 | ........ | ........ |
| 24 | ........ | ........ | ........ | ........ | ........ | 28 | 49 | 68 | 89 | 112 | 138 | 155 | 178 | ........ | ........ |
| 25 | ........ | ........ | ........ | ........ | ........ | 22 | 41 | 58 | 78 | 98 | 123 | 138 | 160 | ........ | ........ |
| 26 | ........ | ........ | ........ | ........ | ........ | ........ | 33 | 49 | 68 | 87 | 109 | 123 | 143 | ........ | ........ |
| 27 | ........ | ........ | Factor, 6.6 | | ........ | ........ | 27 | 41 | 58 | 76 | 97 | 110 | 129 | ........ | ........ |
| 28 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 34 | 50 | 67 | 86 | 98 | 116 | ........ | ........ |
| 29 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 43 | 58 | 75 | 88 | 103 | ........ | ........ |
| 30 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 51 | 68 | 78 | 93 | ........ | ........ |

| Span, ft. | Depth of core, 14 in. Dead load per sq. ft. = 65 lb.<br>Safe superimposed loads, uniformly distributed, lb. per sq. ft. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 15 | ...... | ...... | 106 | 128 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 16 | ...... | ...... | 85 | 105 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| 17 | ...... | ...... | 68 | 85 | 102 | 135 | 183 | 226 | 277 | 329 | ...... | ...... | ...... | ...... | ...... |
| 18 | ...... | ...... | 54 | 69 | 84 | 113 | 157 | 195 | 239 | 286 | ...... | ...... | ...... | ...... | ...... |
| 19 | ...... | ...... | 42 | 56 | 68 | 95 | 133 | 168 | 208 | 251 | 297 | 329 | ...... | ...... | ...... |
| 20 | ...... | ...... | 32 | 43 | 56 | 80 | 115 | 146 | 182 | 220 | 262 | 291 | ...... | 338 | 408 |
| 21 | ...... | ...... | 22 | 33 | 44 | 66 | 98 | 126 | 159 | 193 | 232 | 257 | ...... | 301 | 365 |
| 22 | ...... | ...... | ...... | 25 | 35 | 55 | 83 | 109 | 139 | 170 | 206 | 229 | ...... | 268 | 327 |
| 23 | ...... | ...... | ...... | 17 | 26 | 44 | 71 | 94 | 122 | 151 | 182 | 204 | ...... | 239 | 293 |
| 24 | ...... | ...... | ...... | ...... | 18 | 36 | 60 | 82 | 107 | 132 | 162 | 182 | ...... | 215 | 264 |
| 25 | ...... | ...... | ...... | ...... | ...... | 28 | 50 | 70 | 93 | 117 | 144 | 162 | ...... | 192 | 238 |
| 26 | ...... | ...... | ...... | ...... | ...... | 21 | 41 | 60 | 81 | 103 | 129 | 146 | ...... | 174 | 215 |
| 27 | ...... | ...... | ...... | ...... | ...... | ...... | 33 | 50 | 70 | 91 | 114 | 131 | ...... | 157 | 195 |
| 28 | ...... | ...... | ...... | ...... | ...... | ...... | 27 | 42 | 61 | 81 | 102 | 117 | ...... | 141 | 177 |
| 29 | ...... | ...... | ...... | ...... | ...... | ...... | 20 | 35 | 52 | 71 | 91 | 105 | ...... | 127 | 161 |
| 30 | ...... | Factor, 7.2 | | ...... | ...... | ...... | ...... | 28 | 44 | 62 | 81 | 93 | ...... | 114 | 146 |
| 31 | ...... | | | ...... | ...... | ...... | ...... | 22 | 37 | 53 | 71 | 83 | ...... | 102 | 132 |
| 32 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 32 | 46 | 62 | 74 | ...... | 92 | 120 |
| 33 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 55 | 66 | ...... | 83 | 109 |
| 34 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 49 | 58 | ...... | 75 | 99 |

* Does not apply to 6-in. depth of core. † Does not apply to 8-in. depth of core. ‡ Does not apply to 10-in. depth of core.

## Table 12.—Extreme Dimensions of Floor Plates with Raised Tread

Thickness is exclusive of Raised Tread

(Carnegie-Illinois Steel Corp., Pittsburgh, 1936)

| Width, in....... | 6 to 12 | 12 to 24 | 24 to 36 | 36 to 48 | 48 to 60 | 60 to 66 | 66 to 72 | Wt. per sq. ft., lb. |
|---|---|---|---|---|---|---|---|---|
| Thickness, in. | Length, Inches | | | | | | | |
| 3/4 | ....... | 120 | 180 | 280 | 264 | 240 | 218 | 31.7 |
| 5/8 | ....... | 180 | 200 | 300 | 300 | 260 | 240 | 26.6 |
| 1/2 | 120 | 240 | 240 | 320 | 360 | 300 | 240 | 21.5 |
| 7/16 | 120 | 240 | 240 | 340 | 360 | 300 | 240 | 19.0 |
| 3/8 | 120 | 240 | 300 | 340 | 360 | 300 | 240 | 16.4 |
| 5/16 | 120 | 240 | 300 | 320 | 360 | 300 | 240 | 13.9 |
| 1/4 | 120 | 240 | 300 | 320 | 360 | 240 | 220 | 11.3 |
| 3/16 | 120 | 240 | 300 | 320 | 360 | 240 | 200 | 8.8 |

erected, gives smooth ceilings and has low dead load; structural aerocrete weighs 60 lb. per cu. ft. Spans of 22 ft. will carry a 60-lb. superimposed load on a total thickness of 10 $^1/_2$ in., including 1-in. fireproofing below, and 1 $^1/_2$ in. above, joists. Topping would be of 1 : 2 : 4 stone concrete, 2 $^1/_2$ in. thick, with joists spaced 4 ft. 0 in. with adequate bridging. Near the upper surface $^1/_4$-in. rounds, 1 ft. 6 in. centers, are run across the joists and a continuous sheet of welded wire fabric reinforces the aerocrete slab between joists.

**STEEL PLATE FLOOR CONSTRUCTION.**—Comparatively shallow I-beams spanning between girders and carrying sections of steel plate welded together to form the floor, are used in industrial buildings. Such floors are very shallow, as the I-beams are stiffened by the plates welded to the upper flanges; when properly insulated, they are economical for certain types of industrial work.

**CELLULAR STEEL FLOOR CONSTRUCTION,** comprising thin steel sheets formed and welded to produce a hollow structural unit 12 in. or 24 in. wide, can be used on short and medium spans to make a light and inexpensive floor which may receive any desired surface treatment. This system offers particular advantages for running conduit within the floor thickness and can be used economically with a structural steel frame where fireproofing requirements can be met.

**FLOOR FILLING** in fire-resistant buildings usually is cinder compacted by tamping and often mixed with Portland cement (1 part cement, 8 parts cinder by volume). The depth of fill is 3 or 4 in., providing space for pipes and conduits. With open-web steel joists or hung ceilings, fill usually is unnecessary except for the embedment of sleepers for finished wood floors, or for insulation.

Small-sized conduit may be placed in structural members of concrete ribbed floors; with solid reinforced concrete slabs the conduit is embedded in the slab. For most commercial, institutional and residence construction floor fills usually are required.

**Table 13.—Tile Depth and Steel Area for Long-span Hollow Tile Construction**

Blocks 12 × 12 in.; 4-in. reinforced concrete joists; 2-in. concrete top. Compressive unit stress in extreme fibers of concrete = $f_c$ = 650 lb. per sq. in. Tensile unit stress in steel = $f_s$ = 16,000 lb. per sq. in. Shear limited to 60 lb. per sq. in. $T$ = tile thickness, in.; $S$ = steel area in each joist, sq. in.; $^3/_4$ in. concrete protection below reinforcement.

| Span, ft. | Total Safe Load, uniformly distributed, lb. per sq. ft. * | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Continuous Span; $M = WL/12$ † | | | | | | | | | | | | | | | | | |
| | 150 | | 165 | | 180 | | 195 | | 210 | | 225 | | 240 | | 260 | | 300 | |
| | Semi-continuous Span; $M = WL/10$ | | | | | | | | | | | | | | | | | |
| | 125 | | 135 | | 150 | | 160 | | 175 | | 185 | | 200 | | 220 | | 250 | |
| | Simple Span; $M = WL/8$ | | | | | | | | | | | | | | | | | |
| | 100 | | 110 | | 120 | | 130 | | 140 | | 150 | | 160 | | 175 | | 200 | |
| | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ | $T$ | $S$ |
| 6 | ... | ..... | ... | ..... | ... | ..... | ... | ..... | ... | ..... | 3 | 0.20 | 3 | 0.22 | 3 | 0.24 | 3 | 0.28 |
| 7 | 3 | 0.19 | 3 | 0.20 | 3 | 0.22 | 3 | 0.24 | 3 | 0.26 | 3 | .28 | 3 | .30 | 4 | .26 | 4 | .30 |
| 8 | 3 | .24 | 3 | .26 | 3 | .29 | 3 | .31 | 3 | .34 | 3 | .36 | 4 | .31 | 4 | .34 | 5 | .32 |
| 9 | 3 | .31 | 3 | .33 | 3 | .37 | 3 | .40 | 4 | .34 | 4 | .36 | 4 | .39 | 5 | .35 | 5 | .40 |
| 10 | 3 | .38 | 3 | .41 | 3 | .46 | 4 | .39 | 4 | .42 | 4 | .45 | 5 | .40 | 5 | .44 | 6 | .42 |
| 11 | 3 | .46 | 4 | .39 | 4 | .44 | 4 | .47 | 5 | .42 | 5 | .45 | 5 | .48 | 6 | .45 | 7 | .45 |
| 12 | 4 | .43 | 4 | .47 | 4 | .52 | 5 | .46 | 5 | .50 | 5 | .53 | 6 | .49 | 6 | .54 | 7 | .53 |
| 13 | 4 | .51 | 4 | .55 | 5 | .50 | 5 | .54 | 5 | .59 | 6 | .53 | 6 | .57 | 7 | .55 | 8 | .55 |
| 14 | 4 | .59 | 5 | .53 | 5 | .58 | 5 | .63 | 6 | .58 | 6 | .61 | 7 | .58 | 7 | .64 | 8 | .64 |
| 15 | 5 | .56 | 5 | .61 | 5 | .67 | 6 | .61 | 6 | .67 | 7 | .61 | 7 | .66 | 8 | .65 | 9 | .66 |
| 16 | 5 | .64 | 5 | .69 | 6 | .65 | 6 | .70 | 7 | .66 | 7 | .70 | 8 | .67 | 9 | .66 | 10 | .68 |
| 17 | 6 | .61 | 6 | .67 | 6 | .74 | 7 | .68 | 7 | .75 | 8 | .70 | 8 | .76 | 9 | .75 | 10 | .77 |
| 18 | 6 | .69 | 6 | .75 | 6 | .82 | 7 | .77 | 8 | .74 | 8 | .78 | 9 | .76 | 10 | .76 | 12 | .73 |
| 19 | 6 | .76 | 6 | .83 | 7 | .80 | 7 | .86 | 8 | .83 | 9 | .78 | 9 | .85 | 10 | .85 | 12 | .81 |
| 20 | 7 | .74 | 7 | .80 | 7 | .89 | 8 | .84 | 8 | .91 | 9 | .87 | 10 | .85 | 12 | .79 | 12 | .90 |
| 21 | 7 | .81 | 7 | .89 | 8 | .87 | 8 | .93 | 9 | .91 | 9 | .96 | 10 | .94 | 12 | .87 | 15 | .80 |
| 22 | 7 | .89 | 8 | .86 | 8 | .95 | 9 | .91 | 9 | .99 | 10 | .95 | 12 | .87 | 12 | .95 | 15 | .88 |
| 23 | 8 | .86 | 8 | .93 | 8 | 1.03 | 9 | .98 | 10 | .98 | 10 | 1.03 | 12 | .94 | 12 | 1.03 | 15 | .95 |
| 24 | 8 | .94 | 8 | 1.02 | 9 | 1.01 | 9 | 1.08 | 10 | 1.07 | 12 | 0.96 | 12 | 1.03 | 15 | 0.92 | 15 | 1.05 |
| 25 | 8 | 1.02 | 9 | 1.00 | 9 | 1.10 | 10 | 1.07 | 12 | 0.98 | 12 | 1.03 | 12 | 1.12 | 15 | 1.00 | 15 | 1.13 |

WEIGHT OF LONG-SPAN HOLLOW TILE CONSTRUCTION PER SQ. FT.

Blocks 12 × 12 in.; 4-in. reinforced concrete joists; 2-in. concrete top. Data from National Fireproofing Co.

| Tile, in. | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 12 | 15 |
|---|---|---|---|---|---|---|---|---|---|---|
| Weight, lb. | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 90 | 105 |

* For safe superimposed load, deduct from total load shown in table, the weight of the construction.

† If moment $WL/12$ is used, investigate design for shear.

## Table 14.—Total Safe Loads, Uniformly Distributed, Lb. per Sq. Ft. for Open-web Steel Joists *

This table is based on the standard specification for steel joists of the Steel Joist Institute. Spacing limited to 24 in. in floors and 30 in. in roofs, except where roofing consists only of wood, or sheet metal, the spacing may be increased to 7 ft. All joists must have adequate bridging, bearing and anchorage.

| Span, ft. | Spacing Center to Center of Joists, in. | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 28 | 30 |
| **8-inch Joists—Type 81** | | | | | | | | | | | | | |
| 8 | 231 | 217 | 205 | 194 | 184 | 176 | 168 | 160 | 154 | 148 | 142 | 132 | 123 |
| 9 | 182 | 171 | 162 | 153 | 145 | 138 | 132 | 127 | 121 | 116 | 112 | 104 | 97 |
| 10 | 148 | 139 | 131 | 125 | 118 | 113 | 108 | 103 | 99 | 95 | 91 | 85 | 79 |
| 11 | 121 | 114 | 108 | 102 | 97 | 92 | 89 | 85 | 81 | 78 | 75 | 70 | 65 |
| 12 | 102 | 96 | 91 | 86 | 82 | 78 | 74 | 71 | 68 | 65 | 63 | 58 | 55 |
| 13 | 87 | 82 | 78 | 73 | 70 | 66 | 63 | 61 | 58 | 56 | 54 | 50 | 46 |
| 14 | 75 | 71 | 67 | 63 | 60 | 57 | 55 | 52 | 50 | 48 | 46 | 43 | 40 |
| 15 | 66 | 62 | 58 | 55 | 52 | 50 | 48 | 46 | 44 | 42 | 40 | 37 | 35 |
| 16 | 58 | 54 | 51 | 49 | 46 | 44 | 42 | 40 | 38 | 37 | 35 | 33 | 31 |
| **10-inch Joists—Type 102** | | | | | | | | | | | | | |
| 12 | 219 | 206 | 194 | 184 | 175 | 167 | 159 | 152 | 146 | 140 | 135 | 125 | 117 |
| 13 | 187 | 176 | 166 | 157 | 149 | 142 | 136 | 130 | 124 | 119 | 115 | 107 | 99 |
| 14 | 161 | 152 | 143 | 135 | 129 | 123 | 117 | 112 | 107 | 103 | 99 | 92 | 86 |
| 15 | 141 | 132 | 125 | 118 | 112 | 107 | 102 | 98 | 94 | 90 | 87 | 80 | 75 |
| 16 | 123 | 116 | 110 | 104 | 99 | 94 | 90 | 86 | 82 | 79 | 76 | 70 | 66 |
| 17 | 109 | 103 | 97 | 92 | 87 | 83 | 79 | 76 | 73 | 70 | 67 | 62 | 58 |
| 18 | 97 | 92 | 86 | 82 | 78 | 74 | 71 | 68 | 65 | 62 | 60 | 56 | 52 |
| 19 | 87 | 82 | 78 | 74 | 70 | 66 | 63 | 61 | 58 | 56 | 54 | 50 | 47 |
| 20 | 79 | 74 | 70 | 66 | 63 | 60 | 57 | 55 | 53 | 51 | 49 | 45 | 42 |
| **12-inch Joists—Type 123** | | | | | | | | | | | | | |
| 15 | 204 | 192 | 182 | 172 | 163 | 156 | 148 | 142 | 136 | 131 | 126 | 117 | 109 |
| 16 | 180 | 170 | 160 | 152 | 144 | 137 | 131 | 125 | 120 | 115 | 111 | 103 | 96 |
| 17 | 159 | 150 | 142 | 134 | 127 | 121 | 116 | 111 | 106 | 102 | 98 | 91 | 85 |
| 18 | 142 | 134 | 126 | 120 | 114 | 108 | 103 | 99 | 95 | 91 | 88 | 81 | 76 |
| 19 | 128 | 120 | 113 | 107 | 102 | 97 | 93 | 89 | 85 | 82 | 79 | 73 | 68 |
| 20 | 115 | 108 | 102 | 97 | 92 | 88 | 84 | 80 | 77 | 74 | 71 | 66 | 61 |
| 21 | 104 | 98 | 93 | 88 | 83 | 79 | 76 | 73 | 70 | 67 | 64 | 60 | 56 |
| 22 | 95 | 90 | 85 | 80 | 76 | 73 | 69 | 66 | 63 | 61 | 59 | 54 | 51 |
| 23 | 87 | 82 | 77 | 73 | 70 | 66 | 63 | 61 | 58 | 56 | 54 | 50 | 46 |
| 24 | 80 | 75 | 71 | 67 | 64 | 61 | 58 | 55 | 53 | 51 | 49 | 46 | 42 |
| **14-inch Joists—Type 145** | | | | | | | | | | | | | |
| 19 | 217 | 204 | 193 | 183 | 173 | 165 | 158 | 151 | 145 | 139 | 134 | 124 | 115 |
| 20 | 195 | 184 | 174 | 164 | 156 | 149 | 142 | 136 | 130 | 125 | 120 | 112 | 104 |
| 21 | 177 | 166 | 157 | 149 | 142 | 135 | 129 | 123 | 118 | 113 | 109 | 101 | 94 |
| 22 | 162 | 152 | 144 | 136 | 129 | 123 | 117 | 112 | 108 | 103 | 100 | 92 | 86 |
| 23 | 147 | 139 | 131 | 124 | 118 | 112 | 107 | 103 | 98 | 95 | 91 | 84 | 79 |
| 24 | 135 | 127 | 120 | 114 | 108 | 103 | 99 | 94 | 90 | 87 | 83 | 77 | 72 |
| 25 | 125 | 117 | 111 | 105 | 100 | 95 | 91 | 87 | 83 | 80 | 77 | 71 | 66 |
| 26 | 115 | 108 | 102 | 97 | 92 | 88 | 84 | 80 | 77 | 74 | 71 | 66 | 61 |
| 27 | 107 | 101 | 95 | 90 | 86 | 82 | 78 | 75 | 71 | 69 | 66 | 61 | 57 |
| 28 | 100 | 94 | 89 | 84 | 80 | 76 | 72 | 69 | 66 | 64 | 61 | 57 | 53 |
| **16-inch Joists—Type 166** | | | | | | | | | | | | | |
| 23 | 209 | 197 | 186 | 176 | 167 | 159 | 152 | 145 | 139 | 134 | 128 | 119 | 111 |
| 24 | 200 | 188 | 178 | 168 | 160 | 152 | 145 | 139 | 133 | 128 | 123 | 114 | 107 |
| 25 | 186 | 175 | 165 | 156 | 148 | 141 | 135 | 129 | 124 | 119 | 114 | 106 | 99 |
| 26 | 172 | 162 | 153 | 145 | 137 | 131 | 125 | 119 | 115 | 110 | 106 | 98 | 92 |
| 27 | 159 | 150 | 141 | 134 | 127 | 121 | 116 | 111 | 106 | 102 | 98 | 91 | 85 |
| 28 | 148 | 139 | 131 | 124 | 118 | 113 | 107 | 103 | 99 | 95 | 91 | 84 | 79 |
| 29 | 138 | 130 | 123 | 116 | 111 | 105 | 100 | 96 | 92 | 88 | 85 | 79 | 74 |
| 30 | 129 | 121 | 114 | 109 | 103 | 98 | 94 | 90 | 86 | 83 | 79 | 74 | 69 |
| 31 | 121 | 114 | 107 | 102 | 97 | 92 | 88 | 84 | 81 | 77 | 74 | 69 | 64 |
| 32 | 113 | 107 | 101 | 96 | 91 | 86 | 82 | 79 | 76 | 73 | 70 | 65 | 60 |

* For safe superimposed load deduct from total load shown in table the weight of the joists as follows: 8-in., 3.42 lb.; 10-in., 4.00 lb.; 12-in., 5.00 lb.; 14-in., 6.92 lb.; 16-in., 8.30 lb. Weights are per lineal feet of joist. Data from Truscon Steel Co.

### Table 15.—Safe Loads, Uniformly Distributed, Pounds per Square Foot, for Floor Joists

$f$ = extreme fiber stress, lb. per sq. in., $v$ = horizontal shear, lb. per sq. in.

NOTE.—Loads shown in tables are live and dead loads computed for common grade on exact sizes. Loads to right of heavy lines are safe loads but will cause deflections greater than 1/360 span. Values in beam table are for beams or girders 1 in. thick and laterally supported; multiply by the actual, *not* nominal, width of member.

| Size of Joist, in. (nominal) | Spacing, in., Center to Center | Douglas Fir, Coast Region $f = 1200$, $v = 72$ | | | | | | | | Southern Yellow Pine $f = 1200$, $v = 88$ | | | | | | | | Douglas Fir, Rocky Mountain Region, Red, White and Sitka Spruce $f = 880$, $v = 68$ | | | | | | | | Redwood $f = 960$, $v = 56$ | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Span in Feet | | | | | | | | | | | | | | | | Span in Feet | | | | | | | | | | | | | | | |
| | | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 |
| 2 × 8...... | 12 | 147 | 117 | 85 | 63 | ... | ... | ... | ... | 179 | 123 | 85 | 63 | ... | ... | ... | ... | 139 | 90 | 62 | ... | ... | ... | ... | ... | 114 | 91 | 68 | 50 | ... | ... | ... | ... |
| | 16 | 110 | 88 | 64 | ... | ... | ... | ... | ... | 134 | 92 | 64 | ... | ... | ... | ... | ... | 104 | 68 | ... | ... | ... | ... | ... | ... | 85 | 68 | 50 | ... | ... | ... | ... | ... |
| | 20 | 88 | 70 | 51 | ... | ... | ... | ... | ... | 107 | 74 | 51 | ... | ... | ... | ... | ... | 83 | 54 | ... | ... | ... | ... | ... | ... | 68 | 55 | ... | ... | ... | ... | ... | ... |
| 2 × 10..... | 12 | 185 | 148 | 124 | 100 | 76 | 60 | 49 | ... | 227 | 181 | 136 | 100 | 76 | 60 | 49 | ... | 175 | 140 | 100 | 73 | 56 | ... | ... | ... | 144 | 115 | 96 | 80 | 61 | ... | ... | ... |
| | 16 | 139 | 111 | 93 | 75 | 57 | ... | ... | ... | 170 | 136 | 102 | 75 | 57 | ... | ... | ... | 131 | 105 | 75 | 55 | ... | ... | ... | ... | 108 | 86 | 72 | 60 | ... | ... | ... | ... |
| | 20 | 111 | 89 | 74 | 60 | ... | ... | ... | ... | 136 | 109 | 82 | 60 | ... | ... | ... | ... | 105 | 84 | 60 | ... | ... | ... | ... | ... | 86 | 69 | 58 | 48 | ... | ... | ... | ... |
| 2 × 12..... | 12 | 224 | 180 | 150 | 128 | 112 | 88 | 72 | 59 | 274 | 220 | 183 | 146 | 112 | 88 | 72 | 59 | 212 | 170 | 141 | 107 | 82 | 65 | 53 | ... | 175 | 140 | 116 | 100 | 87 | 71 | 57 | ... |
| | 16 | 168 | 135 | 113 | 96 | 84 | 66 | 54 | ... | 206 | 165 | 137 | 110 | 84 | 66 | 54 | ... | 159 | 128 | 106 | 80 | 62 | 49 | ... | ... | 131 | 105 | 87 | 75 | 65 | 53 | ... | ... |
| | 20 | 134 | 108 | 90 | 77 | 67 | 53 | ... | ... | 164 | 132 | 110 | 87 | 67 | 53 | ... | ... | 127 | 102 | 85 | 64 | 49 | ... | ... | ... | 105 | 84 | 70 | 60 | 52 | ... | ... | ... |
| 3 × 8...... | 12 | 236 | 189 | 137 | 100 | 77 | 61 | 49 | ... | 288 | 197 | 137 | 100 | 77 | 61 | 49 | ... | 223 | 144 | 100 | 73 | 56 | ... | ... | ... | 184 | 147 | 109 | 80 | 62 | 49 | ... | ... |
| | 16 | 177 | 142 | 103 | 75 | 58 | ... | ... | ... | 216 | 148 | 103 | 75 | 58 | ... | ... | ... | 167 | 108 | 75 | 55 | ... | ... | ... | ... | 138 | 110 | 82 | 60 | ... | ... | ... | ... |
| | 20 | 142 | 113 | 82 | 60 | ... | ... | ... | ... | 173 | 118 | 82 | 60 | ... | ... | ... | ... | 134 | 86 | 60 | ... | ... | ... | ... | ... | 110 | 88 | 65 | 48 | ... | ... | ... | ... |
| 3 × 10..... | 12 | 299 | 239 | 200 | 161 | 124 | 97 | 79 | 65 | 366 | 293 | 219 | 161 | 124 | 97 | 79 | 65 | 282 | 226 | 161 | 118 | 91 | 72 | 58 | ... | 232 | 186 | 155 | 129 | 99 | 78 | 63 | 52 |
| | 16 | 224 | 179 | 150 | 121 | 93 | 73 | 59 | 49 | 275 | 220 | 164 | 121 | 93 | 73 | 59 | 49 | 212 | 170 | 120 | 88 | 68 | 54 | ... | ... | 174 | 140 | 116 | 97 | 74 | 59 | ... | ... |
| | 20 | 179 | 143 | 120 | 97 | 74 | 58 | ... | ... | 220 | 176 | 131 | 97 | 74 | 58 | ... | ... | 169 | 136 | 97 | 71 | 55 | ... | ... | ... | 139 | 112 | 93 | 77 | 59 | ... | ... | ... |
| 3 × 12..... | 12 | 364 | 291 | 242 | 208 | 181 | 143 | 115 | 96 | 445 | 355 | 296 | 236 | 181 | 143 | 115 | 96 | 344 | 275 | 229 | 173 | 132 | 104 | 85 | 70 | 284 | 227 | 189 | 162 | 145 | 114 | 93 | 76 |
| | 16 | 273 | 218 | 182 | 156 | 136 | 107 | 86 | 72 | 334 | 266 | 222 | 177 | 136 | 107 | 86 | 72 | 258 | 206 | 172 | 130 | 99 | 78 | 64 | ... | 213 | 170 | 142 | 121 | 109 | 86 | 70 | 57 |
| | 20 | 218 | 175 | 145 | 125 | 109 | 86 | 69 | 58 | 267 | 213 | 178 | 142 | 109 | 86 | 69 | 58 | 206 | 165 | 137 | 104 | 79 | 62 | 51 | ... | 171 | 136 | 113 | 97 | 88 | 68 | 56 | ... |
| 3 × 14..... | 12 | 426 | 340 | 284 | 243 | 212 | 189 | 160 | 132 | 521 | 416 | 347 | 297 | 249 | 197 | 160 | 132 | 402 | 322 | 268 | 230 | 183 | 144 | 117 | 97 | 331 | 265 | 220 | 189 | 166 | 147 | 128 | 105 |
| | 16 | 320 | 255 | 213 | 183 | 159 | 142 | 120 | 99 | 392 | 312 | 270 | 223 | 187 | 148 | 120 | 99 | 302 | 242 | 201 | 173 | 137 | 108 | 88 | 73 | 248 | 199 | 165 | 142 | 124 | 110 | 96 | 79 |
| | 20 | 255 | 204 | 170 | 146 | 127 | 113 | 96 | 79 | 313 | 250 | 208 | 178 | 149 | 118 | 96 | 79 | 241 | 193 | 161 | 138 | 110 | 86 | 70 | 58 | 199 | 141 | 132 | 113 | 100 | 88 | 77 | 63 |

### Safe Uniform Loads, lb., for Beams 1 in. Thick

| Depth of Beam, in. | Span in Feet | | | | | | Span in Feet | | | | | | Span in Feet | | | | | | Span in Feet | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 10 | 12 | 14 | 16 | 18 | 20 | 10 | 12 | 14 | 16 | 18 | 20 | 10 | 12 | 14 | 16 | 18 | 20 | 10 | 12 | 14 | 16 | 18 | 20 |
| 6................... | 405 | 335 | 290 | 250 | 220 | 200 | 405 | 335 | 290 | 250 | 220 | 200 | 295 | 245 | 210 | 185 | 165 | 150 | 320 | 270 | 230 | 200 | 180 | 160 |
| 8................... | 720 | 625 | 535 | 470 | 415 | 375 | 750 | 625 | 535 | 470 | 415 | 375 | 550 | 460 | 390 | 345 | 305 | 275 | 560 | 500 | 430 | 375 | 335 | 300 |
| 10................... | 910 | 910 | 860 | 750 | 670 | 600 | 1115 | 1005 | 860 | 750 | 670 | 600 | 860 | 735 | 630 | 550 | 490 | 440 | 710 | 710 | 690 | 605 | 535 | 480 |
| 12................... | 1100 | 1100 | 1100 | 1100 | 980 | 885 | 1350 | 1350 | 1260 | 1105 | 980 | 885 | 1040 | 1040 | 925 | 810 | 720 | 650 | 860 | 860 | 860 | 860 | 785 | 705 |
| 14................... | 1290 | 1290 | 1290 | 1290 | 1290 | 1210 | 1585 | 1585 | 1585 | 1515 | 1350 | 1210 | 1220 | 1220 | 1220 | 1110 | 990 | 890 | 1010 | 1010 | 1010 | 1010 | 1010 | 970 |

**PRECAST SLABS,** supported by steel framing, form good structural surfaces and may be used for floors carrying light loads. Commercial slabs, 2 to 3 in. thick, of light weight concretes or gypsum, reinforced with small rods, span 2 to 4 ft.

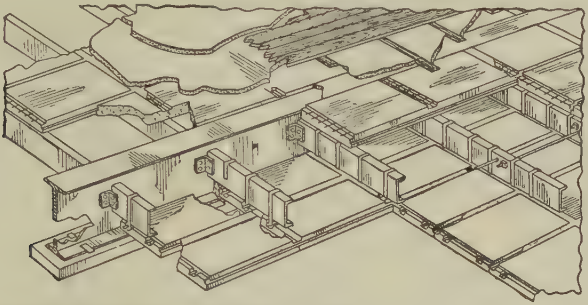


Fig. 14. Gypsteel Precast Floor Construction Showing Steel Channels and Supporting Girders

## Wood Joist Floors

For other than fire-resistant buildings, wood is the prevailing material used for floors. Lally columns and structural steel girders have replaced timber posts and wood girders. Wood joists 2 in. thick × 10 or 12 in. deep are used; on long spans a 3 × 14-in. joist may be required. Joists are usually set 16-in. centers with lines of $^7/_8 \times 3$-in. bridging spaced not over 8 ft. apart. For finished floors an underflooring of $^7/_8 \times 6$-in. tongue-and-groove

**Table 16.—Working Stresses for Structural Lumber and Timber, Lb. per Sq. In., for Dry Locations**

(Recommended by National Lumber Manufacturers' Assoc.)

| Specie and Grade | Extreme Fiber Stress in Bending | | Maximum Horizontal Shear | Compression Parallel to Grain | Compression Perpendicular to Grain | Modulus of Elasticity |
|---|---|---|---|---|---|---|
| | Joist and Plank, 4 in or Less Thick | Beams and Girders, 5 in. or More Thick | | | | |
| Douglas Fir, Coast Region | | | | | | 1,600,000 |
| Dense Super-structural | 2000 | 2000 | 120 | 1466 | 380 | |
| Super-structural and Dense Structural | 1800 | 1800 | 105 | 1300 | 345 | |
| Structural | 1600 | 1600 | 90 | 1200 | 345 | |
| Common Structural | 1200 | 1400 | 84 | 1100 | 325 | |
| Douglas Fir, Inland Empire | | | | | | 1,600,000 |
| Dense Super-structural | 2000 | 2000 | 120 | 1466 | 380 | |
| Dense Structural | 1800 | 1800 | 105 | 1300 | 345 | |
| No. 1 Common Dimension and Timbers | 1135 | 1135 | 70 | 1010 | 315 | |
| Larch, Western | | | | | | 1,300,000 |
| No. 1 Common Dimension and Timbers | 1135 | 1135 | 70 | 1010 | 325 | |
| Pine, Southern Yellow | | | | | | 1,600,000 |
| Extra Dense Select Structural | 2300 | 2300 | 200 | 1600 | 475 | |
| Select Structural | 2000 | 2000 | 175 | 1450 | 375 | |
| Extra Dense Heart | 2000 | 2000 | 175 | 1450 | 475 | |
| Dense Heart | 1800 | 1800 | 150 | 1300 | 375 | |
| Structural Square Edge and Sound | 1600 | 1600 | 125 | 1200 | 375 | |
| No. 1 Common | 1200 | 1200 | 100 | 1000 | 325 | |
| Redwood | | | | | | 1,200,000 |
| Super-structural | 2133 | 1707 | 93 | 1422 | 267 | |
| Prime Structural | 1707 | 1494 | 82 | 1245 | 267 | |
| Select Structural | 1280 | 1322 | 70 | 1100 | 267 | |
| Heart Structural | 1024 | 1150 | 56 | 1000 | 267 | |

roofers is laid diagonally over the joists and nailed with at least two 8d. wire nails in each board. The diagonal sub-floor strengthens wood-frame buildings, and the finished floor may be laid in any direction. Waterproofing paper should be placed between the two floors in basements, and ordinary building paper in the upper stories.

**FLOORS LAID ON THE GROUND.**—To prevent dampness, floors on the ground must be drained; to prevent settlement the surface must be uniformly firm and well compacted. On dry, well-drained soils, 6 in. of cinder, thoroughly tamped and evenly surfaced suffice. Stone-concrete (1 part Portland cement, 3 parts sand, 5 parts crushed stone or gravel, by volume), 4 in. thick is satisfactory for cellar floors with granolithic finish. For damp sub-soil, 12 in. of broken stone should be under the cinder fill; if water is present, sub-drainage is necessary; even if there is no hydrostatic head, a dampproofing course of membrane waterproofing, and a layer of hollow tile, or both, may be used if finished floor is wood, or if the rooms are used for living quarters. If actual hydrostatic head exists at basement level, adequate waterproofing is required. See p. 17–44.

## Floor Finishes

In factories and warehouses, cement floor finishes are common; wood blocks are preferable for heavy traffic. Cement floors are used in public, commercial and institutional buildings, with clay tile, cork, rubber tile and various composition floorings in baths, lobbies, offices, etc. Wood flooring, excepting wood block floors, is used in residences, apartments, club-houses, assembly halls and gymnasiums.

**CEMENT FLOORS** may be laid integrally with the concrete base, or bonded to the base later, or laid on cinder fill. The integral floor is cheapest if conditions permit a monolithic finish as the floor is laid. A thin layer of dry mortar (1 part cement, 1 $^1/_2$ parts sand, by volume) is spread on the slab within $^1/_2$ hour after laying the latter, and finished with a steel trowel. The surface may be left exposed or covered with linoleum. The amount of material depends on the smoothness of the slab; a fair estimate is 2 $^1/_2$ bbl. Portland cement and 15 cu. ft. well-graded, screened sand per 1000 sq. ft. of floor.

Bonded cement floors are of cement mortar (1 part cement, 1 part sand, 1 part finely crushed stone, called grits), laid about 1 $^1/_4$ in. thick. For each 1000 sq. ft. allow 15 bbl. Portland cement, 2.25 cu. yd. screened sand and 2.25 cu. yd. grits. The slab is roughened, cleaned, thoroughly wet, coated with a grout of cement and water, immediately before laying mortar. The latter, mixed to a stiff consistency, is tamped and finished with a steel trowel. Space for pipes is provided by laying the floor finish on a moistened and compacted bed of steam cinder covering the pipes, the surface being reinforced with wire fabric. All cement finishes should be protected by paper or burlap as soon as the surface has hardened, and covered with soft wood shavings, earth or sand kept wet for 10 days. Proprietary compounds, as hardeners, give a hard, dustless surface.

**TERRAZZO FLOORS** comprise a 1-in. surface layer of cement and graded aggregates, on a 2-in. layer of stone concrete on slab or floor fill. The surface is finished by polishing machines which expose the aggregate. To prevent cracking, brass strips $^1/_8$ in. wide, extending through the entire thickness of concrete and finish, divide the floor into panels.

**CLAY TILE,** vitreous and semi-vitreous, usually are laid on a $^1/_2$-in. mortar setting bed over a 2-in. bed of stone concrete or 2 $^1/_2$ in. of cinder concrete on the slab. A 3-in. minimum should be used over earth or fill. The setting bed should be reinforced with wire fabric to prevent cracking. The ceramic mosaic designs of small sizes of unglazed tile are used for the floors of bathrooms and toilets. Ceramic tile are arranged in various patterns, glued on sheets of paper which are soaked off after placement.

**CORK TILE,** composed of cork shavings pressed into squares and rectangles, are made in various shades of brown, from $^5/_{16}$ to $^9/_{16}$ in. thick. They are laid in paste over cement floors, giving a resilient, noiseless finish, but are too expensive for ordinary work.

**LINOLEUM** is finely ground cork pressed into wide strips. It ranges from $^1/_8$ to $^1/_4$ in. thick. The $^1/_4$-in. thickness is battleship linoleum. It is pasted to cement floors without lining, but on wooden floors a felt lining is used.

**RUBBER TILE FLOORING,** laid directly on cement floors, or with a lining on wood floors, makes a resilient, waterproof surface. Individual units are square or rectangular, from $^3/_{16}$ to $^1/_2$ in. thick.

**COMPOSITION FLOORING** usually is magnesite or asphalt, laid in a plastic state. Magnesite flooring usually is laid in two coats of $^1/_2$ in. total thickness, on wood or concrete base. It comes in a variety of colors, is resilient and waterproof. Asphalt compositions usually contain asbestos fiber mixed with mineral pigments. They are laid as tile or in the plastic state. Thickness for heavy foot traffic is $^1/_4$ to $^3/_4$ in.; for trucking, 1 to 2 in. Asphalt often is mixed with marble chips or crushed rock, formed in blocks, and ground down to obtain the effect of terrazzo.

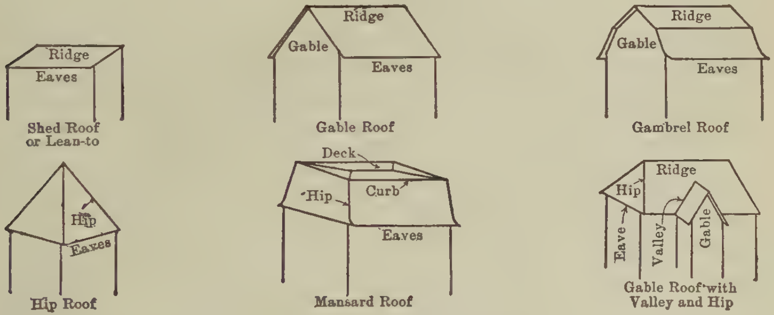
**WOOD BLOCKS** of yellow pine or redwood, with grain vertical, are used to resist heavy traffic in warehouses and factories. Laid on a concrete base, in pitch or asphalt binder, they form a lasting surface. Thickness usually is 2 to 4 in.

**STRIP FLOORING** of oak, yellow pine or maple is used for residences and institutions. A $^{7}/_{8} \times 6$ in. tongue-and-groove diagonal underflooring first is laid on the wood joists, with building paper between it and the finish floor; in basements waterproof paper should be used. With fire-resistant floors the underflooring is nailed to sleepers embedded about 20 in. apart in cinder concrete. Occasionally nailing strips are embedded in the upper surface of the slab, or 2 in. of Nailcrete or Nailcote over the floor fill provides nailing for the underflooring. Strip floorings usually are $^{13}/_{16}$ or $^{25}/_{32}$ in. thick; for heavy traffic the thickness may be up to 2 $^{5}/_{8}$ in. Face widths vary from 1 $^{1}/_{2}$ to 3 $^{1}/_{4}$ in.; both sides and ends are tongue-and-groove, and the backs slightly hollowed to prevent warping.

## 8. ROOFS

### Method of Supporting Roofs

**FLAT ROOFS** for multi-story reinforced concrete buildings, are designed the same as floors; the live load is less and the structural members correspondingly lighter. The structural slab is pitched $^{1}/_{4}$ in. per foot for drainage, or cinder filling may be used. A 5-ply built-up membrane, flashed into parapets and penthouses, generally is used as surfacing. If shallow pools of water on the wearing surface are not objectionable, roofs may be laid absolutely flat.



Shed Roof or Lean-to

Gable Roof

Gambrel Roof

Hip Roof

Mansard Roof

Gable Roof with Valley and Hip

Fig. 15. Types of Roofs

**PITCHED ROOFS** may be framed with concrete, steel or wood. Concrete is not economical unless the structural members are elements of a rigid frame; such construction is not used generally in the U. S., but in Germany and France, it is used in auditorium and hangar construction.

Steel trusses are used for pitched roofs of both wall-bearing and steel-frame buildings. Wood trusses, particularly the bow string type, are economical for garages and some

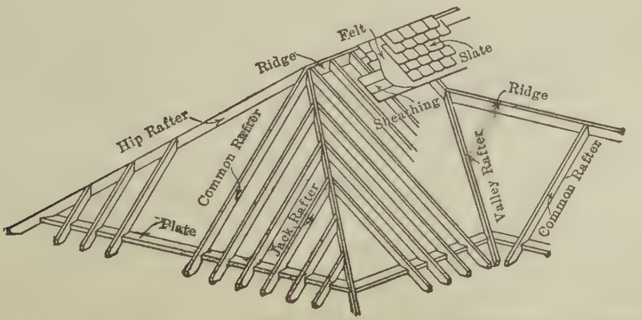


Fig. 16. Wood Frame Roof Construction Showing Method of Applying Slate (*Nat'. Slate Assoc.*)

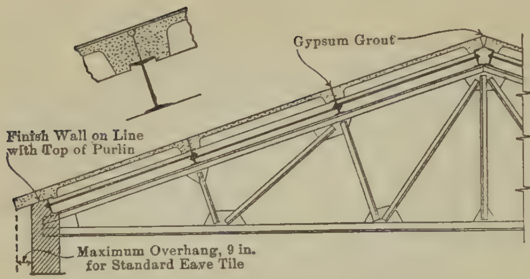

Gypsum Grout
Finish Wall on Line with Top of Purlin
Maximum Overhang, 9 in. for Standard Eave Tile

FIG. 17. Typical Gable Roof Design

industrial buildings. Lamella designs, both steel and wood, are used for auditoriums and industrial plants. Steel arches may be economical in one-story structures.

A complete roof system, supported by trusses, consists of roof deck, rafters, purlins and trusses. A bay is the roof area between two trusses. Purlins are horizontal beams between trusses. Roofs may be laid directly on purlins, or on sub-purlins parallel to the trusses. Roof trusses comprise inclined or horizontal top chords, horizontal or cambered lower chords, and web systems. Panel points are joints where web members intersect chords; the distance between them is the panel length.

The **height** of a truss is the vertical distance from peak to a horizontal line passing through both supports. **Pitch** is the ratio of rise to span length. **Slope** is the ratio of rise to $1/2$ span. A truss of 10 ft. rise and 40 ft. span has a pitch of $1/4$, a slope of $1/2$ or of 6 in 12. If architectural design does not control the slope, a pitch of $1/4$ generally is economical.

## Loads on Roofs

**THE DEAD LOAD** on a roof truss comprises the weight of the truss, purlins, rafters, roof deck and covering, and any suspended ceilings or floors. The approximate weight of roof trusses up to 75 ft. span, under normal conditions, is about 10% of the load to be carried. When the truss is designed, actual weight should be computed and compared with assumed weight.

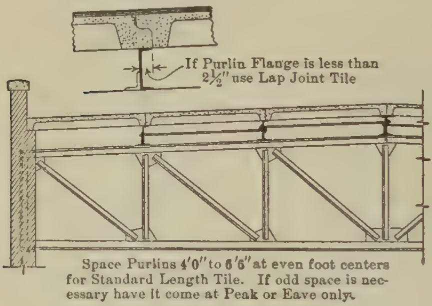

If Purlin Flange is less than 2½" use Lap Joint Tile
Space Purlins 4'0" to 6'6" at even foot centers for Standard Length Tile. If odd space is necessary have it come at Peak or Eave only.

FIG. 18. Typical Flat Roof Design

**THE LIVE LOAD** is the sum of probable snow and wind loads. The former depends on geographical location, roof slope and character of roof covering. Table 17 gives snow loads for various locations and roof slopes.

**Wind Pressure** on an inclined roof is that acting in a direction normal to the surface. Duchemin's formula, the most widely used, is

$$P_n = P\{2 \sin A/(1 + \sin^2 A)\},$$

where $P$ = unit pressure, lb. per sq. ft.; $A$ = angle of roof surface to the horizontal. Table 18 gives values of $P_n$ when $P = 20$ and $P = 30$. The value of the wind load carried to each panel point of the truss is $(P_n \times L \times d)$, where $L$ = panel length and $d$ = distance between trusses, ft. Maximum wind pressure stresses are assumed by considering the wind acting in one direction on one side of the roof and then in the reverse direction on the opposite side.

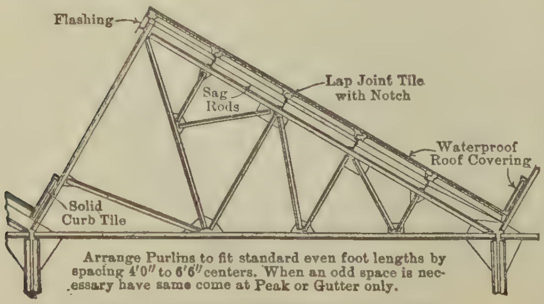

Flashing
Sag Rods
Lap Joint Tile with Notch
Waterproof Roof Covering
Solid Curb Tile
Arrange Purlins to fit standard even foot lengths by spacing 4'0" to 6'6" centers. When an odd space is necessary have same come at Peak or Gutter only.

FIG. 19. Typical Sawtooth Roof Design

**Combined Roof Loads** include both snow and wind loads. They are taken as a uniform vertical load acting over the entire roof surface. Values are given in Table 19; the total uniform vertical load is the tabular

Table 17.—Allowance for Snow Loads

| Location | Pitch.... | 1/2 | | 1/3 | | 1/4 | | 1/5 | 1/6 |
|---|---|---|---|---|---|---|---|---|---|
| | Slope.... | 12 in 12 | | 8 in 12 | | 6 in 12 | | 4.8 in 12 | 4 in 12 |
| | | Lb. per sq. ft. of Roof Surface | | | | | | | |
| | | * | † | * | † | * | † | | |
| Southern States and Pacific Slope | | 0 | 0 | 0 | 5 | 0 | 5 | 5 | 5 |
| Central States | | 0 | 5 | 7 | 10 | 15 | 20 | 22 | 30 |
| Rocky Mountain States | | 0 | 10 | 10 | 15 | 20 | 25 | 27 | 35 |
| New England States | | 0 | 10 | 10 | 15 | 20 | 25 | 35 | 40 |
| Northwestern States | | 0 | 12 | 12 | 18 | 25 | 30 | 37 | 45 |

* For slate, tile or metal roofs. † For shingle roofs.

Table 18.—Normal Wind Pressure, Lb. per Sq. Ft. of Roof Surface

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Pitch | 1/6 | 1/5 | 1/4 | | 1/3 | 1/2 | |
| Slope | 4/12 | 4.8/12 | 6/12 | 30° | 8/12 | 45° | 60° and over |
| Normal Pressure $P_n$ $P = 20$ | 11.5 | 13.1 | 14.9 | 16.0 | 17.0 | 18.9 | 20.0 |
| Normal Pressure $P_n$ $P = 30$ | 17.2 | 19.8 | 22.4 | 24.0 | 25.5 | 28.3 | 30.0 |

Table 19.—Equivalent Vertical Loads for Combined Wind and Snow Loads

| Location | Pitch... | | 1/2 | 1/3 | 1/4 | 1/5 | 1/6 |
|---|---|---|---|---|---|---|---|
| | Slope... | 60° | 45° | 8 in 12 | 6 in 12 | 4.8 in 12 | 4 in 12 |
| | | Lb. per Sq. Ft. of Roof Surface | | | | | |
| Southern States and Pacific Slope | | 30 | 30 | 25 | 25 | 22 | 20 |
| Central States | | 30 | 30 | 25 | 25 | 22 | 30 |
| Rocky Mountain States | | 30 | 30 | 25 | 25 | 27 | 35 |
| New England States | | 30 | 30 | 25 | 25 | 35 | 40 |
| Northwestern States | | 30 | 30 | 25 | 30 | 37 | 45 |

value plus dead load. The panel load is $A \times (D + C)$, where $A$ = area of panel, sq. ft., and $D$ and $C$ = dead load and combined load respectively, lb. per sq. ft. The estimated weight of the truss is distributed among the panel points, and the weight of the purlins added.

**PURLINS** are placed at panel points on top chords of roof trusses; sometimes one or more are placed between panel points. Each purlin supports a roof area of (panel length, *i.e.*, distance between purlins, × length of bay, *i.e.*, distance between trusses). With purlins between panel points, the top chord must be designed for bending combined with direct stress. The vertical load on a purlin may be resolved into components normal and parallel to the roof slope. If the roof deck gives lateral support, the parallel component may be assumed as carried by the deck; if the deck lacks rigidity for lateral support, $^5/_8$-in. or $^3/_4$-in. tie rods are run from purlin nearest the heel on one side of the truss, over the ridge, and down the other side. These should be placed at the center of $^1/_3$ points of purlin spans. Tie rods seldom are necessary on roofs of slope less than $^1/_4$. The component perpendicular to the roof is found graphically, and the purlin designed as a simple beam. See p. 17–47.

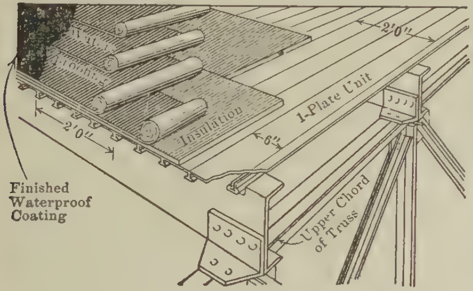


Fig. 20. Typical Steel Deck Construction (*Truscon Steel Co.*)

## Roof Decks

**STEEL ROOF DECKS.**—One type of deck uses hollow ribbed units over standard steel rafters and purlins. Its weight, including insulation and waterproofing, approximates 5 lb. per sq. ft. Sheets are end-lapped and attached to purlins by clips. For additional rigidity, arc-welding replaces clips.

Another type is No. 16 or 20 gage galvanized steel sheet with interlocking ribs 1 $^1/_2$ in. deep, spaced 6 in. apart, in lengths up to 12 ft. No. 18 gage is used on spans up to 8 ft.; the weight is 3 $^1/_4$ lb. per sq. ft. Steel roof decks may be covered with composition roofing or prepared for slag, slate or tile.

**GALVANIZED STEEL SHEATHING,** pressed into channels, is economical on a purlin spacing of about 4 ft. When rolled from No. 24 gage steel, the sections are 18 in. wide. This construction is suitable for flat roofs, when covered with insulation and membrane roofing.

**SLABS CAST IN PLACE,** supported directly by purlins have been used on better classes of work, but have given way to precast units.

**PRECAST SLABS** of light weight concrete, gypsum or similar materials are used for roof decks on fire-resistant buildings. The units are small, easily handled, and of sufficient structural strength to span between sub-purlins. They can be adapted to any type of roof surface.

Porete and Aerocrete are patented materials. Porete comes in slabs about 1 1/4 in. thick, 24 in. wide, 32 in. long. They are placed between T-shape sub-purlins, spaced about 2 ft. 9 in. centers. Larger units, up to 3 in. thick are suitable for a 6-ft. span. Slabs are reinforced with woven galvanized netting and fastened to the purlins by clips. Porete is highly porous, and is an excellent insulator. On flat or slightly sloped roofs, membrane waterproofing is laid on a 1/4-in. layer of cement and sand mixed with a small amount of nailing compound. For slate or tile the layer is 1/2 in. thick.

Porete slabs cast in channels may be used on a 6-ft. span; this assembly weighs 12 lb. per sq. ft. A combination of steel roof deck and Porete, comprising a ribbed steel unit 18 in. wide, ribs 6 in. apart, and a layer of light-weight nailing concrete, can be used on spans up to 10 ft. It is adapted to both flat and pitched roofs with any type of surface.

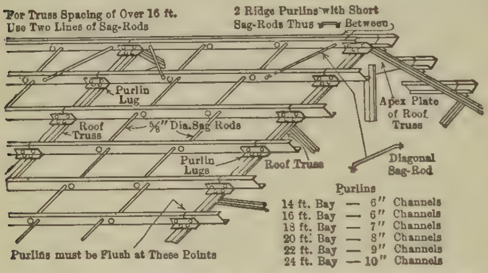


FIG. 21. Arrangement of Purlins Supporting Concrete Tile (*Am. Cement Tile Mfg. Co.*)

Standard dimensions of precast slabs of nailing Aerocrete are 24 × 32 in., 2 1/2 to

**Table 20.—Loads and Reinforcement for Aerocrete Short Span Roof Construction**

Assumptions: Slab Thickness = 4 in.; Unit tensile stress in steel = 18,000 lb.; Unit compressive stress in extreme fibers of concrete = 300 lb. (max.); Weight of Aerocrete = 60 lb. per cu. ft.

| Uniformly Distributed Dead Load + Live Load, lb. per sq. ft. | | | Span | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Moment Coefficient | | | | | | | | | |
| 1/8 | 1/10 | 1/12 | 5′ 0″ | 5′ 6″ | 6′ 0″ | 6′ 6″ | 7′ 0″ | 7′ 6″ | 8′ 0″ |
| | | | Steel Area per Foot Width of Slab, sq. in. | | | | | | |
| 53 | 67 | 80 | 0.039 | 0.047 | 0.056 | 0.068 | 0.078 | 0.09 | 0.104 |
| 66 | 83 | 100 | .049 | .059 | .071 | .084 | .098 | .114 | .131 |
| 80 | 100 | 120 | .059 | .072 | .086 | .103 | .12 | .139 | .16 |
| 93 | 117 | 140 | .069 | .084 | .102 | .121 | .141 | .164 | .189 |

**Table 21.—Safe Loads, Uniformly Distributed, for Gypsteel 3-in. Precast Roof Slabs**

Cables consist of 3 1/6-in. cold-drawn steel wire rods. Unit working stress 20,000 lb. per sq. in. Weight of 3-in. slab = 14 lb. per sq. ft.

| Slab Width, in. | Cable Spacing, in. | Cables per Slab | Distance Between Centers of Supports | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 4′ 0″ | 4′ 3″ | 4′ 6″ | 4′ 9″ | 5′ 0″ | 5′ 3″ | 5′ 6″ | 5′ 9″ | 6′ 0″ | 6′ 3″ | 6′ 6″ | 6′ 9″ | 7′ 0″ |
| | | | Safe Load, lb. per sq. ft. | | | | | | | | | | | | |
| 24...... | 4 | 6 | 136 | 122 | 108 | 98 | 88 | 80 | 72 | 66 | 61 | .. | .. | .. | .. |
| | 3.43 | 7 | 159 | 142 | 126 | 114 | 102 | 93 | 84 | 78 | 72 | .. | .. | .. | .. |
| | 3.00 | 8 | 184 | 162 | 144 | 130 | 117 | 106 | 97 | 89 | 82 | 76 | .. | .. | .. |
| 21...... | 4.2 | 5 | .. | .. | .. | .. | .. | .. | .. | .. | 59 | 54 | 50 | 46 | .. |
| | 3.5 | 6 | .. | .. | .. | .. | .. | .. | .. | .. | 70 | 64 | 59 | 55 | .. |
| | 3 | 7 | .. | .. | .. | .. | .. | .. | .. | .. | 82 | 76 | 70 | 65 | 60 |
| | 2.62 | 8 | .. | .. | .. | .. | .. | .. | .. | .. | 94 | 87 | 80 | 74 | 68 |
| 18...... | 3.6 | 5 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 54 | 50 |
| | 3 | 6 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 65 | 60 |
| | 2.57 | 7 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 76 | 70 |

3 in. thick. Structural Aerocrete, cast 3 to 4 in. thick may be used on spans up to 8 ft. It can be made to weigh 30 to 80 lb. per cu. ft.; strength increases with the weight.

Precast gypsum units on bulb T-sections make an excellent surface under tile or slate for short-span roofs. The Gypsteel system comes in several designs; the long-span suspension type is suitable for spans up to 7 ft. between purlins. The short-span system comprises 30-in. slabs resting on bottom flanges of sub-purlins. Another system comprises sheet rock forms on sub-purlins, spaced approximately 32 in. centers, over which gypsum stucco, containing 12 $^1/_2$% of wood fiber, by weight, is poured. Reinforcement is galvanized fabric. Weight is approximately 55 lb. per cu. ft. This design can be used for roofs pitched up to 30 deg.

**PRECAST SLABS** (Haydite) of light-weight aggregates made from burnt slag and reinforced with wire mesh, weigh about 10 lb. per sq. ft., when designed for spans up to 6 ft. For slate, tile or sheet metal, $^1/_4$ in. of nailing material is cast integrally over the top surface. Including surfacing, the slabs weigh 18 lb. per sq. ft. when adapted to 5 ft. purlin spacing.

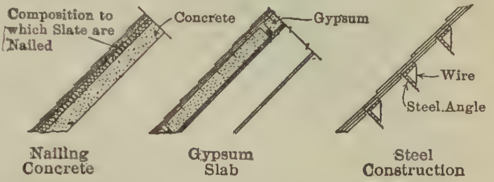


Fig. 22. Methods of Applying Slate over Various Types of Construction (*Natl. Slate Assoc.*)

For flat or steeply inclined roofs, gypsum slabs, 2 × 15 in., form economical decks for spans up to 7 ft. They weigh 11 lb. per sq. ft., are fire-resistant and good insulators.

## Roof Surfacing Materials

**CONCRETE TILE** is used for both flat and pitched roofs, particularly on industrial work; standard size of interlocking type is 24 × 52 in., 1 $^1/_8$ in. thick. Slabs hook over steel channel purlins, overlap and interlock, and are sealed with plastic cement on four sides. No surfacing is required. Purlins are spaced 4 ft. centers. Tile are made with Haydite aggregate and permanent colors on outer surfaces.

**CLAY TILE, SLATE AND ASBETOS SHINGLES** may be laid over any of the above roof decks, if nailing is provided. With stone or cinder concrete without other surfacing, 1 × 2 in. beveled wood strips are embedded in the slabs, over which a 30-lb. roofing felt is laid. A nailable surfacing is preferable to wood strips. Non-corrodible nails should be used with surfacing materials.

The length of slate usually is taken as (2 × exposure + head lap); on roofs of normal slope, head lap = 3 in. On steep roofs, as Mansard, 2-in. head lap is sufficient. Slopes less than 4 in. per ft. are treated as flat decks.

**SLATE.**—A formula derived from the recommendations of the National Slate Assoc. for estimating number of squares of slate required for a building is

$$S = K\{L \times a(R + 0.5) - {}^1/_2 M + D + H\} \div 100$$

where $S$ = squares of slate; $L$ = length of roof, ft.; $R$ = length of rafters, ft.; $M$ = (area of chimneys $c$ + area of dormers $d$), sq. ft.; if $(c + d) < 20$, $M = 0$; if $(c + d) > 80$, $M = (c + d) - 20$; $D$ = area of dormer roofs, sides of slated dormers, slated saddles and other slate surfaces, sq. ft.; $H$ = total length of hips and valleys; $a$ = constant = 1 for single slopes, and 2 for double slope roofs. $K$ = constant ranging from 1.02 to 1.15, depending on the number of intersections of other roofs, dormers, etc.

**TILE SLABS, QUARRY TILE** vary from 6 × 6 to 9 × 9 in., 1 in. thick. They are laid over membrane waterproofing on flat roofs, with elastic-filled expansion joints around all parapets, penthouses, etc.

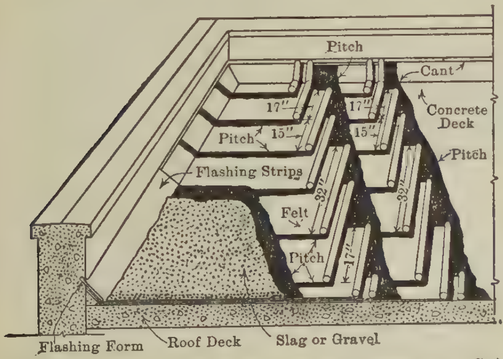


Fig. 23. Four-ply Membrane Roofing (*Barrett Mfg. Co.*)

**MEMBRANE ROOFING** in 3 to 5 layers, laid either with coal-tar pitch or asphalt, is standard roof waterproofing. Wearing surface on horizontal or slightly pitched roofs is slag or gravel; 200 lb. of pitch, 300 lb. of slag or 400 lb. of gravel are required per square.

**ASBESTOS SHINGLES,** made in a variety of sizes, colors and textures, are laid by two methods, hexagonal or French, and American. Hexagonal shingles usually are 12 or 16 in. square. The lower corner is cut off to give a honeycomb effect when laid. Average weight per square of $^{1}/_{8}$-in. thickness French laid is 275 lb.; average weight American laid is 565 lb. Asbestos corrugated roofing is used for pitched roofs of warehouses and industrial buildings.

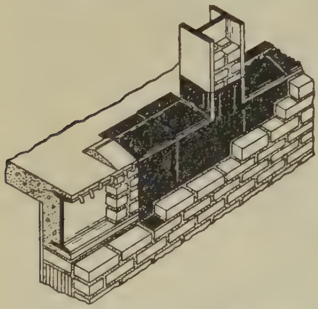


Fig. 24. Spandrel Beam Flashing (*Structural Waterproofing Co.*)

Three-ply roofing, guaranteed for 15 years, requires 175 lb. of pitch per square when laid on concrete. On wood roof decks, the first two plies are laid dry; a total of five plies then corresponds to and carries the same guarantee (20 yr.) as 4-ply work laid on concrete.

Roofs of industrial or commercial buildings used for recreation, use coal-tar pitch mixed at the factory with crushed limestone. It is laid cold, 1 $^{1}/_{2}$ in. thick, and later compacted to 1 in. by a 500-lb. roller.

**HEAVY FELTS,** impregnated with asphalt and surfaced with crushed minerals, are used as individual or strip shingles in a wide variety of colors and sizes. Standard sizes of individual shingles (American type) are 12 × 15 and 9 × 12 in., $^{3}/_{16}$ in. to $^{1}/_{4}$ in. thick. The 4-tab strips usually are 10 × 36 in.; when laid with a 2-in. head lap, weigh approximately 290 lb. per square.

**METAL ROOFING** for industrial buildings comprises corrugated iron sheets, tin and terne plate. See pp. 17–81 to 17–86.

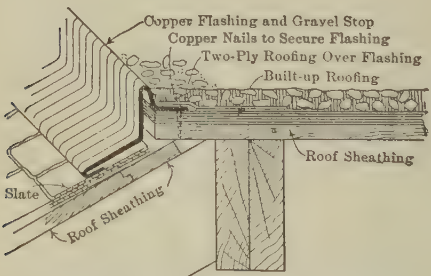
**Copper, Lead** and **Zinc** sheets also are used on both flat and pitched roofs. Copper costs more than tin, but is more lasting. It is laid over a felt base with flat-locked seams on roofs of pitch less than 1 in 3, and with ribbed- or standing-seams on sloping roofs.

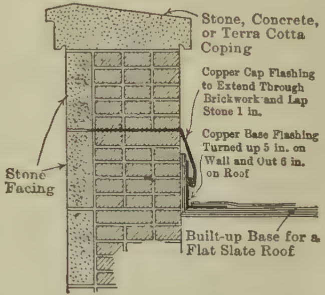
**Lead** is not used extensively, but lead-covered copper is occasionally used on monumental work. Zinc generally is cheaper than copper and has been used both as roofing and siding on industrial buildings. Corrugated sheets, 9 to 15 gage, weigh 79 and 187 lb., respectively, per square. No. 9 gage sheets with $^{7}/_{8}$-in. corrugations may be used with 30-in. purlin spacing; No. 15 gage, with a 78-in. purlin spacing.

**Aluminum** roofing in ordinary sheets, corrugated sheets and shingles, is laid by the same methods as other sheet metals. Thus, 24 × 96 in. No. 20 B. & S. gage sheets may be laid on wood battens 2 in. high.

All sheet metal roofs, particularly lead and zinc, must have provision for expansion. For additional information on weights of roofing materials, see p. 17–81.

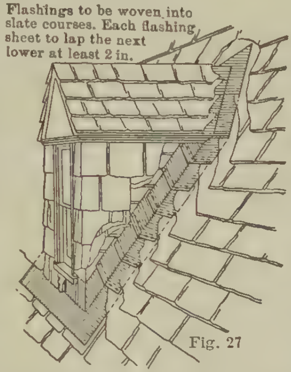
**FLASHINGS** are required where roof surfaces intersect dormers, chimneys, vents, parapets, penthouses; beneath copings and sills; over lintels and cornices; and along valleys, ridges or built-in gutters; and in steel-frame buildings, over spandrel beams at each floor level.

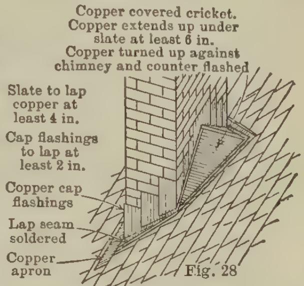

Copper Flashing and Gravel Stop
Copper Nails to Secure Flashing
Two-Ply Roofing Over Flashing
Built-up Roofing
Roof Sheathing
Slate
Roof Sheathing
Fig. 25.—Flashing at Edge of Membrane Roofing Above a Sloping Deck (*Natl. Slate Assoc.*)

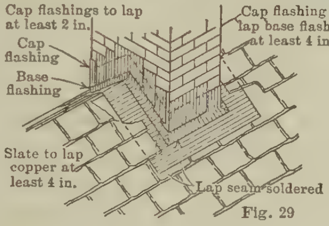

Stone, Concrete, or Terra Cotta Coping
Copper Cap Flashing to Extend Through Brickwork and Lap Stone 1 in.
Copper Base Flashing Turned up 5 in. on Wall and Out 6 in. on Roof
Stone Facing
Built-up Base for a Flat Slate Roof
Fig. 26. Parapet Flashing

The Copper and Brass Research Assoc. recommends 16-oz. soft, roofing temper, copper sheet, laid on rosin-sized paper or asbestos felt, and cornice temper copper for cornice work only. Large loose-lock joints should be used at roof intersections. Finished dimensions of seams are: standing, 1 in.; flat-locked, $1/2$ in.; lapped, 1 in.; double or copper-locked, $1/2$ in. Joints should be tinned thoroughly with enough tin to cover the entire surface, using rosin as a flux, and plenty of best composition half-and-half solder. Seams should be thoroughly heated before soldering. Fastenings should be only cleats or strips fastened with copper nails. Bends should not exceed 90°.

Metal flashings for short open valleys should be 20 in. wide; for long valleys, 24 in. Roofing is lapped 6 in. over the copper, flashing being exposed for 8 in. along bottom of valley. To flash closed valleys insert under every course of slate or other roofing, a trapezoidal sheet of copper approximately 16 × 10 × 10 in.; exact size depends on length of slate and pitch of adjoining slopes. Each sheet should overlap below at least 3 in., and extend at least 2 in. above top of the slate or tile on which it rests. Hips and ridges may be protected by metal molds, or covered by additional courses of shingles or slate without flashing. Special shapes are made for ridges and hips of clay and cement tile.


Flashings to be woven into slate courses. Each flashing sheet to lap the next lower at least 2 in.
Fig. 27


Copper covered cricket. Copper extends up under slate at least 6 in. Copper turned up against chimney and counter flashed
Slate to lap copper at least 4 in.
Cap flashings to lap at least 2 in.
Copper cap flashings
Lap seam soldered
Copper apron
Fig. 28


Cap flashings to lap at least 2 in.
Cap flashing
Base flashing
Cap flashing to lap base flashing at least 4 in.
Slate to lap copper at least 4 in.
Lap seam soldered
Fig. 29

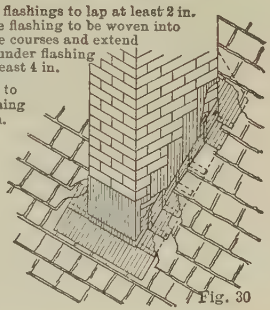

Cap flashings to lap at least 2 in. Base flashing to be woven into slate courses and extend up under flashing at least 4 in.
Fig. 30

Fig. 27. Flashing for Dormer Window
Fig. 28. Flashing and Cricket for Chimney on Roof Slope
Fig. 29. Chimney Flashing on Ridge
Fig. 30. Built-in Base Flashing for Chimney on Roof Slope

## 9. ROOF TRUSSES

**STRESSES IN ROOF TRUSSES** may be determined analytically or graphically. See p. 17–03.

**ALLOWABLE WORKING STRESSES,** when not controlled by a building code may be those in the Specifications for Structural Steel for Buildings. See p. 17–14.

**WOOD ROOF TRUSSES.**—The wood bow-string truss is economical for 25 to 100 ft. spans. Trusses usually are spaced 16 to 20 ft. centers. The required rise is about $1/8$ span. For a slow-burning classification, the size of web members is increased. If these trusses have sufficient fire-resistance, they usually are more economical than steel.

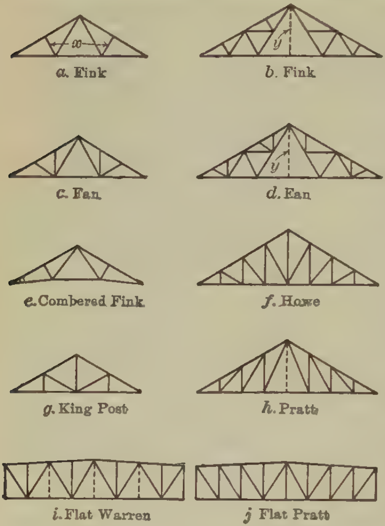
Many types of wood trusses are available for architectural designs. The best known are the hammer beam, scissors trusses and the crescent type; the Howe truss may be designed in wood with either a straight or curved lower chord.

**THE LAMELLA TRUSS** is a well-known patented design, built either of wood or steel. The sections of steel trusses are either pressed metal or standard structural shapes. For fireproofing, 2 in. of Gunite or precast gypsum tile can be used.

In wood designs, timber members range from 2 × 8 to 6 × 24 in., depending on span and load. They are assembled with standard bolts and washers. Four basically different structural types are: Segmental arch, tie rods taking thrust; segmental arch, heavy side walls or buttresses taking thrust; parabolic arch, buttresses taking thrust; gothic arch, thrust transmitted directly to foundations, which are supplemented by buttresses.

An advantage of the Lamella system is that the units are short and light weight. When of wood, they usually cost more than a steel truss. Lamella designs in steel cost approximately the same as steel trusses.

**STEEL ROOF TRUSSES.**—The **Fink Truss** and the **Fan Truss** are used for moderate spans. To lay out the Fink truss, Fig. 31*a*, draw a perpendicular *x* from center of top chord to lower chord; draw the other web members as indicated. In diagrams *b* and *d*, the lines *y* indicate sag rods, added for stiffness; theoretically they carry no stress. Panels in a half-truss may be increased to 4 by subdividing each panel, and to 8 by repeating the process. Fig. 31*c*, *d* shows construction of a fan truss.



*a.* Fink *b.* Fink *c.* Fan *d.* Fan *e.* Combered Fink *f.* Howe *g.* King Post *h.* Pratt *i.* Flat Warren *j* Flat Pratt

FIG. 31. Types of Roof Trusses

Fig. 31 shows other types of roof trusses. If trusses support purlins at panel points only, a good length for top-chord panels is 8 ft., corresponding to a maximum span of about 30 ft. for truss *a*, 40 ft. for *c*, 55 ft. for *b*, and 75 ft. for *d*. Warren and Pratt trusses may be used for long spans; the limit is the capacity of piers to carry end reactions.

Spacing of roof trusses depends on many factors. Arbitrary figures cannot be given. Bays of equal length are desirable in any one section of a building so that as many trusses as possible may be identical. For spans up to 60 ft., 15 to 18 ft. spacing usually is most economical for average loads; this spacing may be controlled by location of windows, piers, or architectural design. For spans over 80 ft., economical spacing may be 20 to 40 ft. To determine most economical spacing for the roof deck used, several designs should be made, including trusses, purlins and wind bracing.

**ARCH ROOF CONSTRUCTION,** usually of steel, is appropriate for spans of 100 ft. or more. The arches may be designed as hingeless, one-, two- or three-hinged arches; the two-hinged type predominates. Standard structural shapes and sizes are employed in a patented design with special connections. Spans from 100 to 200 ft. are the most common, but spans up to 400 ft. or more may be constructed with short lengths of the usual structural materials. The roof may be fireproof, fire-retardant or non-fireproof. Such designs are used for roofs of garages, exhibition halls, gymnasiums, armories and airplane hangars. Foregoing data were supplied by Arch Roof Construction Co.

**CONCRETE SHELL ROOF CONSTRUCTION,** identified as the Zeiss-Dywidag system, may be of the dome or barrel type, and can be applied to roofs of circular, rectangular or polygonal plan. Shell structures are essentially arched plates composed of reinforced concrete, and supported by the stiffening of their rims. They can be adapted with great economy to a variety of buildings where a large, unobstructed floor space is essential, such as planetariums, airplane hangars, market halls, train sheds, etc. By means of this system, shell domes spanning 500 ft., and shell vaults covering vast areas, without interior supporting columns, can be designed in reinforced concrete, which would otherwise be impossible, because of the dead load of the construction. Shell domes, approximately 250 ft. diameter, have been built with a shell thickness of only 3 $^{9}/_{16}$ in., and a barrel-shell concrete roof only 2 $^{3}/_{16}$ in. in thickness has been used on a span of 164 ft.

**GUASTAVINO TIMBREL ARCH CONSTRUCTION,** applicable to both domes and arches, comprises a shell composed of three or more layers of terra cotta tile, 1 in. thick, 6 in. wide, and 12 in. or 24 in. long. These are laid flat in cement mortar and bonded together to form a solid, homogeneous mass, which is reinforced with small steel rods where required. For ordinary 3-course work, the crown of the arch is about 4 in. thick. Domes may be constructed economically up to a span of 130 ft. or more. Hollow tile often are introduced within the thickness of the shell for insulation, and acoustical tile, for sound absorption, on the soffit of the arch or dome.

## 10. WATERPROOFING

**INTEGRAL WATERPROOFING** compounds are mixed with the concrete as liquids, pastes or powders. To some extent they lubricate the aggregate, producing a more workable mixture, reduce the capillarity of the hardened concrete and, in some cases more thoroughly hydrate the cement. Accredited compounds are of distinct value in producing watertight concrete and, if used according to the makers' directions, do not decrease strength. The first requisite is well-graded aggregate. Continuity at construction joints and avoidance of stone pockets or layers of laitance is necessary. Integral waterproofing will not cure poor materials, improperly graded aggregate or careless workmanship. Keys should be made at all construction joints; surfaces already cast should be roughened and slushed with rich mortar to insure a bond. A plastic mixture should be used, which can be thoroughly compacted, and sufficiently dry to avoid laitance forming on newly deposited surfaces.

**SURFACE COAT WATERPROOFING** may be classified as: 1. Plaster coat of cement mortar with or without waterproofing compounds; 2. Penetration surface treatments with inert liquids or liquids reacting chemically with the surface treated; 3. Plastic or liquid bituminous coatings; 4. Non-bituminous coatings forming a protective film on the surface. Classes 2 to 4 are used only for dampproofing. Class 1 consists in applying several coats of cement mortar (1 part cement, 2 parts well-graded sand with the addition of waterproofing compound). The base coat must be bonded to the concrete.

The process is: 1. Thoroughly roughen and wet the surface. 2. Work in a coat of neat cement with a brush or wooden float. 3. Before this grout is dry, apply a $3/8$- to $1/2$-in. coat of waterproof mortar, thoroughly trowelled and lightly scratch-brushed. 4. As soon as first coat is dry, apply a second coat, making a total thickness of $3/4$ in.; trowel to a smooth, hard finish with a steel trowel. For floors, total thickness of both coats is about 1 in. It is more practicable to carry waterproofing up the faces of the columns and partitions than under them. Recesses should be left at construction joints or other places susceptible to leakage, to be filled later with cement plaster reinforced with wire fabric to form a seal. The surface coat is low in cost, and has been applied to the inside surfaces of foundation walls of many large buildings.

**BITUMINOUS OR NON-BITUMINOUS SURFACE WASHES** are applied by a brush or mop. Two or three coats generally are used: a priming coat sufficiently liquid to penetrate the surface; the other coats are somewhat viscous. Such methods are successful for the exterior of basements where subdrainage precludes development of any hydrostatic head. They are not to be recommended where high water pressure is present.

**THE MASTIC METHOD** consists of applying either a sheet of mastic or a layer of brick laid in bituminous material. The brick are laid in two or more courses, one being completely surrounded by mastic. The outside courses often have a protective covering of concrete. This system is not widely used on building construction.

**THE MEMBRANOUS METHOD** uses a tar pitch or asphalt binder between layers of impregnated felt, jute, cotton drill or burlap. It is satisfactory, but often costly, as conditions usually make it impracticable on exteriors; if used on interiors, a concrete seal must hold the membrane in place. Membranes and sheet metals have the sole function of furnishing a more or less elastic waterproof filament and may be used where vibration precludes other methods. Before placing waterproof membrane, concrete surfaces should be pointed, fins removed, and subdrainage provided to insure dry surfaces for treatment. If dry surfaces cannot be obtained, waterproofing may be built over a layer of felt against the wall, although actual bond with the concrete is preferable.

Methods of laying membranes and the number of plies vary with conditions. Asphalt or pitch are satisfactory as binders; treated felt is cheapest, if elasticity is not required. For important work, particularly if elasticity is required, an open-mesh jute or cotton drill, $4\,1/2$ oz. per sq. yd., saturated in a bituminous mixture, is preferable. It should be used with an elastic binder, laid in 3-ft. squares, lapped a minimum of 2 in. on each edge. From 4 to 6 plies are used. Each layer of membrane should be carefully pressed into the binder and complete continuity obtained. Pipes passing through the membrane are fitted with copper or lead flanges extending into the waterproofing at least 1 ft. on all sides; for steam pipes a special detail is necessary to avoid melting the binder.

Two general schemes are used for membrane waterproofing: 1. Membrane is laid as a continuous mat over sub-floor, beneath columns, and carried up outside basement wall, where it is protected by a masonry seal. Below columns, or other members transmitting load, 20-oz. soft-rolled copper sheets extend 1 ft. on all sides into the membrane. 2. Membrane is not laid until columns and exterior walls are built. It is placed on the sub-floor and carried up column sides and exterior walls above the hydrostatic head, and protected with a masonry seal.

It is necessary in either case to build a seal over the basement floor strong enough to resist the pressure. For low heads a gravity seal may be used, but usually the floor is built as a horizontal slab, reinforced against uniform upward pressure and held in place by the columns and wall loads.

## 11. ESTIMATING

Accurate estimates of building costs should be made only from actual cost records on similar work. Quantities of materials for various types of construction can be obtained from handbooks, and material prices from local dealers. Labor costs, given in even the best handbooks, are practically of no value. The efficiency of all labor varies widely in different parts of the country, and standardization of man-hours for even the simplest operations is impossible.

**QUANTITIES.**—Accurate estimates must be based on a carefully made quantity survey. A clear idea of the job as a whole is obtained by visiting the site, reading specifications and studying drawings. Inconsistencies between drawings and specifications should be reconciled, and a definite plan of operation formulated for construction. Quantities are obtained by scaling drawings, in connection with specifications. All quantities are expressed in estimate units, as cu. yd. of excavation, cu. ft. of masonry, thousands of brick and sq. ft, of flooring.

Horizontal areas for excavation, floor construction, etc., and vertical areas for walls and partitions are determined from drawings and recorded. Columns are recorded in linear feet and their type and section noted. Inclined surfaces, as roofs, are computed on the superficial area; stairs and steps by number of treads and risers. The thickness of each element is obtained from drawings, and volumes computed where necessary. Window and door openings, stair and elevator wells, are deducted from gross wall and floor areas. These data are used later to schedule doors, windows or other frame-surrounded openings.

For example, consider concrete partitions. The lengths of all partitions, scaled from plans without deductions for small openings, and multiplied by heights read from the sections, give 1/2 the contact area of forms. Deducting from this quantity area of openings, and multiplying the difference by partition thickness, in decimals of a foot, gives the concrete volume. This same area multiplied by weight of reinforcement per square foot of partitions, is weight of steel in pounds. This area doubled, is the area in square feet for plastering or painting. Information already taken off gives number and size of doors and windows, which are subdivided by types. This last schedule gives linear feet of door frames and sills; from the list of doors and windows, hardware may be noted.

No great degree of accuracy is necessary in scaling quantities, but cumulative errors due to approximating quantities for units which repeat many times, should be avoided. Quantities should be tabulated on a quantity sheet under appropriate captions; these are the major divisions of work arranged in chronological sequence, as excavation, foundations, structural steel, floor arches, stone veneer, common brick, etc. After determining quantities and preparing a tentative plan of construction plant, a speed schedule is made.

**UNIT PRICES.**—Material prices should be carefully figured from quotations, to avoid error in this portion of the estimate. Labor prices must be obtained from previous experience on similar work. Material prices and probable cost of labor items, are multiplied by the quantities to obtain the cost of each item of the work.

**FINAL ESTIMATE** sheets should have separate columns for labor and material costs. Items of job overhead, as liability insurance, superintendent's salary, and plant rental charges are based either on time or percentages of direct labor and material costs. Subcontracts, which normally include heating and ventilating, plumbing, elevator installation, electrical work and sprinklers, may represent from 25% to 75% of total cost. Such items are recorded on the final estimate sheet following the tabulation of work performed by the builder's organization, with its accompanying overhead. The last item, profit or commission, usually a percentage of preceding items, completes the final estimate.

**ESTIMATING COST BY CUBICAL CONTENT.**—For approximate estimates, it may be useful to price a building on a cubic foot basis. The following procedure is based on recommendations of the Am. Inst. of Architects.

The cubic content (cube or cubage) of a building is actual space enclosed between outer surfaces of outside walls and outer surfaces of roof and 6 in. below finished surfaces of lowest floors. This definition includes cube of dormers, pent houses, vaults, pits, enclosed porches and other enclosed appendages as a part of cube of the building. It does not include cube of courts or light shafts, open at the top, or cube of outside steps, cornices, parapets, or open porches or loggias.

Items listed separately are: *a.* Cube of enclosed courts or light shafts open at the top, measured from outside face of enclosing walls, and from 6 in. below finished floor or paving to top of enclosing walls. *b.* Cube of open porches measured from outside face of wall, outside face of columns, finished floor and finished roof. *c.* Square foot area of all stoops, balconies and terraces. *d.* Memoranda, or brief description, of caissons, piling, special foundations, or features, if any.

As a guide, the following cubic prices represent total construction costs, exclusive of engineering or architectural services. These figures are of value *only for the purpose of making crude approximations.*

**Multi-story Industrial Buildings,** reinforced concrete, 20 to 25 cts. per cu. ft.

**Office Buildings, Fire-resistant Construction,** cheaply finished, 50 to 60 cts. per cu. ft. Office buildings, fire-resistant construction, expensively finished, 60 to 90 cts. per cu. ft.

**Public School Buildings,** 25 to 40 cts. per cu. ft.

**Frame Residences,** 25 to 35 cts. per cu. ft.

**Non-fireproof Masonry Residences,** 35 to 50 cts. per cu. ft.

**Fire-resistant, Elaborate Residences,** 50 to 90 cts. per cu. ft.

# PROPERTIES OF STRUCTURAL MATERIALS *

## ROLLED STRUCTURAL STEEL

The tables on pages 17–50 to 17–72 give the dimensions, properties, safe loads, etc., of the more commonly used forms of structural steel. These tables are considerably condensed from those issued by the manufacturers, but present a sufficient range of sizes and weights to permit the engineer to make, at least, preliminary designs. For more detailed information, consult the handbooks issued by the Bethlehem Steel Company, Carnegie-Illinois Steel Corp., and other manufacturers. For a more comprehensive treatment of the theory of beams and columns, see pages 7–17 to 7–34.

## 1. BEAMS AND CHANNELS

**TABLES OF SAFE LOADS.**—The tables of safe loads on the various shapes of structural sections are based on a fiber stress $f$ of 18,000 lb. per sq. in., when the section is used as a beam. Most building codes permit a value of $f = 18{,}000$ for quiescent loads; a few prescribe $f = 16{,}000$. For moving loads, $f = 12{,}000$. Where the beam is subject to impact the value of $f$ should be largely reduced, the formula being

$$f_i = f\{1 + \sqrt{(2H/y) + 1}\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where $f_i$ = extreme fiber stress, lb. per sq. in., due to a falling load; $f$ = extreme fiber stress due to the same load at rest; $y$ = deflection, in., due to the same load at rest; $H$ = height of fall, in.

If the load is concentrated at the center of the span, the safe load is one-half the value given in the tables. If the load is not uniformly distributed nor concentrated at the center of the span, the maximum bending moment must be used. The maximum stress, lb. per sq. in., in a beam is the quotient of the maximum bending moment, in inch-lb., divided by the section modulus. This quotient must not exceed the allowable fiber stress for the particular condition involved as explained above. The weight of the beam must be deducted from the loads given in the tables to obtain the net applied load.

**SHEARING VALUE OF WEB.**—In very short spans or with concentrated loads, a beam or channel may fail by shearing or buckling of the web. Maximum shear values for each size of section are given in the tables, and are based on a unit shearing stress $f_s$ of 12,000 lb. per sq. in. The following methods may be used to determine maximum safe loads, from the standpoint of web shear, the value of $f_s$ being taken on the gross area of the web. Let $V$ = maximum allowable web shear, lb.; $A$ = gross area of web, sq. in.; $t$ = web thickness, in.; $h$ = clear distance between flanges, in.; $f_s$ = unit shearing stress, lb. per sq. in. Then for $h \overset{=}{<} 60\,t$, $f_s = 12{,}000$; for $h > 60\,t$, where the web is not stiffened,

$$f_s = V/A \overset{=}{<} 18{,}000/\{1 + (h^2/7200\,t^2)\} \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

Safe shearing stress usually is taken as 12,000 lb. per sq. in., although some building codes limit it to 10,000 lb. per sq. in. Where loads produce shear exceeding the safe web crippling resistance, web stiffeners should be placed at points of support of concentrated loads and at suitable intervals.

**BUCKLING.**—Beams may fail from end crippling or buckling at the point where the beam rests on bearings of ordinary length, or from the application of concentrated loads. Let $B_c$ = $^1/_2$ distance over which concentrated load is applied, in.; $B_r$ = distance over which end reaction is applied, in.; $D$ = depth of beam, in.; $f_b$ = safe resistance of web to buckling, lb. per sq. in.; maximum value of $f_b$ should not exceed 15,000; $R$ = safe end reaction, lb.; $t$ = thickness of web, in.; $W$ = concentrated load, lb.

$$\text{If both flanges are fixed, } f_b = 18{,}000/[1 + \{(1/6000) \times (D/t)^2\}] \quad . \quad . \quad [3]$$

$$\text{If one flange is fixed, } f_b = 18{,}000/[1 + \{(1/3000) \times (D/t)^2\}] \quad . \quad . \quad [4]$$

$$R = f_b\,t\{B_r + (D/4)\} \quad . \quad . \quad . \quad [5] \qquad W = 2f_b\,t\{B_c + (D/4)\} \quad . \quad . \quad . \quad . \quad . \quad [6]$$

Formula [5] applies to any condition of loading. Formula [6] applies primarily to a single concentrated load, but it will apply to a system of concentrated loads if the sum of the distance $B_c < B_r$. Minimum distance between end of beam and point of application of concentrated load is $(B_c + B_r + D/2)$.

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* Staff contribution.

**DEFLECTION.**—The deflection of a beam uniformly loaded, supported at the ends, is based on the fundamental formula

$$y = (5/384) \times (Wl^3/EI) \quad [7]$$

where $y$ = deflection, in.; $W$ = total uniformly distributed load, lb.; $l$ = length of beam, in.; $E$ = modulus of elasticity = 29,000,000 for steel; $I$ = moment of inertia about neutral axis, in.$^4$ For a fiber stress of 18,000 lb. per sq. in., and a length $L$, ft. = 12 $l$, this formula becomes

$$y = 0.01862\, L^2/2n = C'/2n \quad [8]$$

where $n$ = distance of extreme fiber from neutral axis, and $C'$ = deflection coefficient whose values are given in Table 1. For floor beams carrying plastered ceilings

$$y < \text{span length}/360.$$

**Table 1.—Deflection Coefficients, $C'$, for Structural Beams**

Based on Unit Stress of 18,000 lb. per sq. in.

| Span, ft. | $C'$ | Span, ft. | $C'$ | Span, ft. | $C'$ | Span, ft. | $C'$ | Span, ft. | $C'$ | Span, ft. | $C'$ | Span, ft. | $C'$ | Span, ft. | $C'$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.019 | 11 | 2.253 | 21 | 8.212 | 31 | 17.894 | 41 | 31.301 | 51 | 48.432 | 61 | 69.288 | 71 | 93.867 |
| 2 | .074 | 12 | 2.681 | 22 | 9.012 | 32 | 19.068 | 42 | 32.847 | 52 | 50.350 | 62 | 71.578 | 72 | 96.530 |
| 3 | .168 | 13 | 3.147 | 23 | 9.850 | 33 | 20.278 | 43 | 34.430 | 53 | 52.306 | 63 | 73.906 | 73 | 99.230 |
| 4 | .298 | 14 | 3.650 | 24 | 10.726 | 34 | 21.526 | 44 | 36.050 | 54 | 54.298 | 64 | 76.270 | 74 | 101.967 |
| 5 | .466 | 15 | 4.190 | 25 | 11.638 | 35 | 22.810 | 45 | 37.707 | 55 | 56.328 | 65 | 78.672 | 75 | 104.741 |
| 6 | .670 | 16 | 4.767 | 26 | 12.588 | 36 | 24.132 | 46 | 39.401 | 56 | 58.395 | 66 | 81.112 | 76 | 107.553 |
| 7 | .912 | 17 | 5.381 | 27 | 13.574 | 37 | 25.492 | 47 | 41.133 | 57 | 60.499 | 67 | 83.588 | 77 | 110.402 |
| 8 | 1.192 | 18 | 6.033 | 28 | 14.599 | 38 | 26.888 | 48 | 42.902 | 58 | 62 640 | 68 | 86.102 | 78 | 113.288 |
| 9 | 1.508 | 19 | 6.722 | 29 | 15.660 | 39 | 28.322 | 49 | 44.708 | 59 | 64 819 | 69 | 88.653 | 79 | 116.212 |
| 10 | 1.862 | 20 | 7.448 | 30 | 16.759 | 40 | 29 793 | 50 | 46.552 | 60 | 67 035 | 70 | 91.241 | 80 | 119.172 |

The deflection due to impact of a moving load falling on a beam supported at the ends is

$$y_i = y(1 + \sqrt{(2H/y) + 1} \quad [9]$$

where $y_i$ = deflection, in. due to falling load; $y$ = deflection due to same load at rest; $H$ = height of fall, in. It should be noted that where the weight of the beam is a real factor, observed results do not agree with this formula, and practical tests give values much lower than the formulas.

**BEAMS LATERALLY UNSUPPORTED.**—The tables presuppose that beams are supported against side yielding of the compression flanges, by tie rods or other means at intervals of not more than 15 × flange width. If the distance between lateral supports exceeds this amount the tabular safe loads must be reduced in accordance with the values in Table 2. If $l$ = unsupported length of beam, and $b$ = width of flanges, both in inches, the maximum ratio of $l/b$ is 40. The A.I.S.C. formula for compressive stress in flexure is

$$s_c = 20{,}000/\{1 + l^2/(2000\, b^2)\} \quad [10]$$

with full stress limited to 18,000 lb. per sq. in. up to a ratio of $l/b$ = 15.

**Table 2.—Allowable Safe Loads on Unbraced Structural Sections Used as Beams**

Values are percentages of tabular safe loads (see pp. 17–55 to 17–57 and 17–60) for various ratios of length of span to width of beam, $l/b$.

(Am. Inst. of Steel Construction Code, 1931)

| $l/b$ | Percent | $l/b$ | Percent | $l/b$ | Percent | $l/b$ | Percent |
|---|---|---|---|---|---|---|---|
| ...... | ...... | 21 | 91.0 | 27.5 | 80.6 | 34 | 70.4 |
| 15 | 100.0 | 21.5 | 90.3 | 28 | 79.8 | 34.5 | 69.7 |
| 15.5 | 99.2 | 22 | 89.5 | 28.5 | 79.0 | 35 | 68.9 |
| 16 | 98.5 | 22.5 | 88.7 | 29 | 78.2 | 35.5 | 68.2 |
| 16.5 | 97.8 | 23 | 87.9 | 29.5 | 77.4 | 36 | 67.4 |
| 17 | 97.1 | 23.5 | 87.1 | 30 | 76.6 | 36.5 | 66.7 |
| 17.5 | 96.4 | 24 | 86.3 | 30.5 | 75.8 | 37 | 66.0 |
| 18 | 95.6 | 24.5 | 85.5 | 31 | 75.1 | 37.5 | 65.2 |
| 18.5 | 94.9 | 25 | 84.7 | 31.5 | 74.3 | 38 | 64.5 |
| 19 | 94.1 | 25.5 | 83.9 | 32 | 73.5 | 38.5 | 63.8 |
| 19.5 | 93.4 | 26 | 83.0 | 32.5 | 72.7 | 39 | 63.1 |
| 20 | 92.6 | 26.5 | 82.2 | 33 | 71.9 | 39.5 | 62.4 |
| 20.5 | 91.8 | 27 | 81.4 | 33.5 | 71.2 | 40 | 61.7 |

**BEAM CONNECTIONS.**—Figs. 1 and 2 and Table 7, p. 17–61, show standard forms of beam connections. Fig. 1 shows the construction for the usual run of work, providing shearing and bearing values are adequate. Fig. 2 shows the type of construction recommended where the web is thin, the span is short, or concentrated loads are applied. Under such conditions the riveting of the outstanding legs should be investigated and, if necessary, additional values should be provided by larger rivets in a single row or by double rows in the wider legs.

One-sided connections, that is connecting angle on one side of the beam only, never should be used if they possibly can be avoided. Such connections should be made with 6 × 6 × $^7/_{16}$-in. angles with a double row of rivets in each leg. Such connections should not be used on beams larger than 18 in.

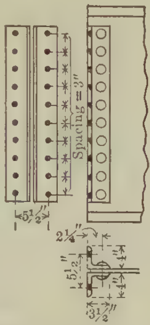


Fig. 1. Beam Connections, Usual Practice

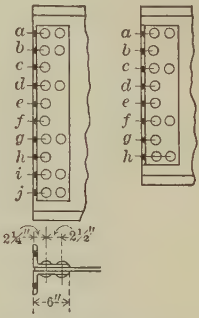


Fig. 2. Beam Connections for Short Span, Thin Web, or Concentrated Loads

**EXAMPLES IN BEAM DESIGN.**—The following examples show the use of the tables of properties and safe loads of standard rolled beams.

**1. Concentrated Loads.**—Required a beam, Fig. 3, of 20 ft. span to carry the following concentrated loads: $W_1$ = 6000 lb., 5 ft. from left support; $W_2$ = 8000 lb., 9 ft. from left support; $W_3$ = 4000 lb., 16 ft. from left support. Assume weight of beam at 50 lb. per ft. = $W_4$ = 50 × 20 = 1000 lb. The reactions are: $(R_1 \times 20) - (6000 \times 15) - (8000 \times 11) - (4000 \times 4) - (1000 \times 10) = 0$; whence $R_1$ = 10,200 lb.

$-(R_2 \times 20) + (6000 \times 5) + (8000 \times 9) + (4000 \times 16) + (1000 \times 10) = 0$; whence $R_2$ = 8800 lb.

Point of maximum bending moment or dangerous section is found as follows:

$(50 \times 5) + 6000 = 6250 < 10{,}200$; $(50 \times 9) + 6000 + 8000 = 14{,}450 > 10{,}200$; hence the dangerous section is at point of load $W_2$.

Maximum bending moment = $(10{,}200 \times 9) - (6000 \times 4) - (50 \times 9) \times 4.5 = 65{,}775$ ft.-lb.
or = $(8800 \times 11) - (4000 \times 7) - (50 \times 11) \times 5.5 = 65{,}775$ ft.-lb.

Based on a unit stress of 18,000 lb. per sq. in.,

Required section modulus = $(65{,}775 \times 12)/18{,}000 = 43.85$ in.$^3$

Referring to Table 5, the 12-in., 40.8-lb. beam has a section modulus of 44.8. It is satisfactory if the beam is braced against lateral deflection. If it is not feasible to brace laterally a heavier beam must be selected.

Three sections may be considered, viz., a 15 × 6-in., 60.8-lb. standard beam (Table 5), or a 12 × 8-in., 40-lb., or a 12 × 10-in., 53-lb. wide flange beam (Table 3).

Applying formula [10] to the 15 × 6 standard beam,

$$s_c = 20{,}000 / [1 + \{240^2/(2000 \times 6^2)\}] = 11{,}111 \text{ lb. per sq. in.}$$

Required section modulus $I/c$ = $(65{,}775 \times 12)/11{,}111 = 71.04$ (see p. 7–25.)

Similarly applying formula [10] to the other two beams:

For the 12 × 8-in. beam, $s_c$ = 13,793, and $I/c$ = 57.15; for the 12 × 10-in. beam, $s_c$ = 15,528 and $I/c$ = 50.48. The required section moduli are then compared with the actual section moduli as given in Tables 3 and 5, as follows:

| | 15 × 6-in. Beam | 12 × 8-in. Beam | 12 × 10-in. Beam |
|---|---|---|---|
| Required section modulus................ | 71.04 | 57.15 | 50.48 |
| Actual section modulus.................. | 81.2 | 51.9 | 70.7 |

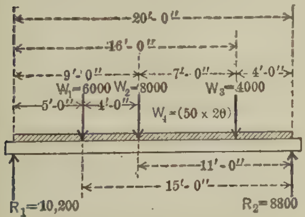


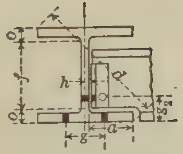
Fig. 3

It is evident that the 12 × 8-in. beam is too light, but that either of the other two are satisfactory.

**2. Uniform Load.**—Select a beam to carry a uniform load of 30,000 lb. on a span of 16 ft., beam simply supported. Select a beam from Table 6. Opposite 16 ft. span, find the safe load of a 12 × 5 $^1/_4$-in., 40.8-lb. standard beam to be 33,600 lb. As this exceeds the specified load, it will be satisfactory.

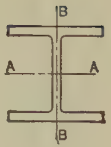
The beam also may be selected by finding the maximum bending moment. For a uniform load, this is at the midpoint of the beam, and is (see Table 2, p. 7–22) $M = Wl/8 = (30{,}000 \times 16 \times 12)/8 = 720{,}000$ in.-lb.

Based on an allowable stress of 18,000 lb. per sq. in. $S$ = 720,000/18,000 = 40 in.$^3$. From Table 5 the section modulus of a 12-in. 40.8-lb. standard beam is 44.8 in., and the maximum safe load is $W = {}^2/_3(fS/L) = {}^2/_3\{(18{,}000 \times 44.8)/16\} = 33{,}600$ lb. Weight of the beam is 16 × 40.8 = 652.8 lb., which must be added to the load; 30,000 + 652.8 = 30,653 < 33,600 lb. Hence, the beam will be satisfactory. By the same form of calculation, a wide flange beam, 8 × 8 in., 48 lb., is found to be satisfactory.



## Table 3.—Properties of American Standard Wide Flange Beams

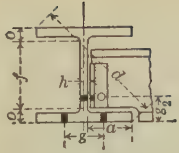
(Carnegie-Illinois Steel Corporation, Pittsburgh)

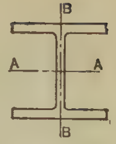


| Depth of Section, in. | Weight per Foot, lb. | Area of Section, sq. in. | Flange | | Web Thickness, in. | Elements of Sections | | | | | | Dimensions for Detailing | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Axis *A–A* | | | Axis *B–B* | | | | | | | | | |
| | | | Wide, in. | Thick, in. | | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | Moment of Inertia, $I'$, in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r'$, in. | $a$, in. | $f$, in. | $o$, in. | Diagonal, $d$, in. | $g_2$, min., in. | Clearance, $h$, in. | Gage, $g$, in. |
| **Section CB 362; 36 × 16 1/2 in.** | | | | | | | | | | | | | | | | | | |
| 36.72 | 300 | 88.17 | 16.655 | 1.680 | 0.945 | 20290.2 | 1105.1 | 15.17 | 1225.2 | 147.1 | 3.73 | 7 7/8 | 31 1/8 | 2 13/16 | 40 3/8 | 4 | 9/16 | 5 1/2 |
| 36.50 | 280 | 82.32 | 16.595 | 1.570 | .885 | 18819.3 | 1031.2 | 15.12 | 1127.5 | 135.9 | 3.70 | 7 7/8 | 31 1/8 | 2 11/16 | 40 1/8 | 4 | 1/2 | 5 1/2 |
| 36.24 | 260 | 76.56 | 16.555 | 1.440 | .845 | 17233.8 | 951.1 | 15.00 | 1020.6 | 123.3 | 3.65 | 7 7/8 | 31 1/8 | 2 9/16 | 39 7/8 | 3 3/4 | 1/2 | 5 1/2 |
| 36.12 | 250 | 73.49 | 16.525 | 1.380 | .815 | 16465.9 | 911.7 | 14.97 | 969.6 | 117.4 | 3.63 | 7 7/8 | 31 1/8 | 2 1/2 | 39 3/4 | 3 3/4 | 1/2 | 5 1/2 |
| 36.00 | 240 | 70.60 | 16.500 | 1.320 | .790 | 15724.0 | 873.6 | 14.92 | 920.1 | 111.5 | 3.61 | 7 7/8 | 31 1/8 | 2 7/16 | 39 5/8 | 3 3/4 | 1/2 | 5 1/2 |
| 35.88 | 230 | 67.73 | 16.475 | 1.260 | .765 | 14988.4 | 835.5 | 14.88 | 870.9 | 105.7 | 3.59 | 7 7/8 | 31 1/8 | 2 3/8 | 39 1/2 | 3 1/2 | 7/16 | 5 1/2 |
| **Section CB 361; 36 × 12 in. 5 weights rolled; alternate sizes listed** | | | | | | | | | | | | | | | | | | |
| 36.48 | 194 | 57.11 | 12.117 | 1.260 | 0.770 | 12103.4 | 663.6 | 14.56 | 355.4 | 58.7 | 2.49 | 5 5/8 | 32 1/4 | 2 1/8 | 38 1/2 | 3 1/4 | 7/16 | 5 1/2 |
| 36.16 | 170 | 49.98 | 12.027 | 1.100 | .680 | 10470.0 | 579.1 | 14.47 | 300.6 | 50.0 | 2.45 | 5 5/8 | 32 1/4 | 1 15/16 | 38 1/8 | 3 1/4 | 7/16 | 5 1/2 |
| 35.84 | 150 | 44.16 | 11.972 | 0.940 | .625 | 9012.1 | 502.9 | 14.29 | 250.4 | 41.8 | 2.38 | 5 5/8 | 32 1/4 | 1 13/16 | 37 7/8 | 3 | 3/8 | 5 1/2 |
| **Section CB 332; 33 × 15 3/4 in.** | | | | | | | | | | | | | | | | | | |
| 33.50 | 240 | 70.52 | 15.865 | 1.400 | 0.830 | 13585.1 | 811.1 | 13.88 | 874.3 | 110.2 | 3.52 | 7 1/2 | 28 5/8 | 2 7/16 | 37 1/8 | 3 3/4 | 1/2 | 5 1/2 |
| 33.25 | 220 | 64.73 | 15.810 | 1.275 | .775 | 12312.1 | 740.6 | 13.79 | 782.4 | 99.0 | 3.48 | 7 1/2 | 28 5/8 | 2 5/16 | 36 7/8 | 3 1/2 | 7/16 | 5 1/2 |
| 33.12 | 210 | 61.78 | 15.783 | 1.210 | .748 | 11664.5 | 704.4 | 13.74 | 735.6 | 93.2 | 3.45 | 7 1/2 | 28 5/8 | 2 1/4 | 36 3/4 | 3 1/2 | 7/16 | 5 1/2 |
| 33.00 | 200 | 58.79 | 15.750 | 1.150 | .715 | 11048.2 | 669.6 | 13.71 | 691.7 | 87.8 | 3.43 | 7 1/2 | 28 5/8 | 2 3/16 | 36 5/8 | 3 1/2 | 7/16 | 5 1/2 |
| **Section CB 331; 33 × 11 1/2 in. 4 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | | | |
| 33.50 | 152 | 44.71 | 11.565 | 1.055 | 0.635 | 8147.6 | 486.4 | 13.50 | 256.1 | 44.3 | 2.39 | 5 1/2 | 29 3/4 | 1 7/8 | 35 1/2 | 3 | 3/8 | 5 1/2 |
| 33.00 | 125 | 36.78 | 11.500 | 0.805 | .570 | 6354.7 | 385.1 | 13.14 | 188.2 | 32.7 | 2.26 | 5 1/2 | 29 3/4 | 1 5/8 | 35 | 2 3/4 | 3/8 | 5 1/2 |
| **Section CB 302; 30 × 15 in.** | | | | | | | | | | | | | | | | | | |
| 30.38 | 210 | 61.78 | 15.105 | 1.315 | 0.775 | 9872.4 | 649.9 | 12.64 | 707.9 | 93.7 | 3.38 | 7 1/8 | 25 3/4 | 2 5/16 | 34 | 3 1/2 | 7/16 | 5 1/2 |
| 30.25 | 200 | 58.76 | 15.070 | 1.250 | .740 | 9340.5 | 617.6 | 12.61 | 665.7 | 88.3 | 3.37 | 7 1/8 | 25 3/4 | 2 1/4 | 33 7/8 | 3 1/2 | 7/16 | 5 1/2 |
| 30.12 | 190 | 55.90 | 15.040 | 1.185 | .710 | 8825.9 | 586.1 | 12.57 | 624.6 | 83.1 | 3.34 | 7 1/8 | 25 3/4 | 2 3/16 | 33 3/4 | 3 1/2 | 7/16 | 5 1/2 |
| 30.00 | 180 | 52.89 | 15.000 | 1.125 | .670 | 8328.2 | 555.2 | 12.55 | 585.6 | 78.1 | 3.33 | 7 1/8 | 25 3/4 | 2 1/8 | 33 5/8 | 3 1/4 | 7/16 | 5 1/2 |
| 29.88 | 172 | 50.65 | 14.985 | 1.065 | .655 | 7891.5 | 528.2 | 12.48 | 550.1 | 73.4 | 3.30 | 7 1/8 | 25 3/4 | 2 1/16 | 33 1/2 | 3 1/4 | 3/8 | 5 1/2 |

| | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Section CB 301; 30 × 10 1/2 in.** 4 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | |
| 30.30 | 132 | 38.83 | 10.551 | 1.000 | 0.615 | 5753.1 | 379.7 | 12.17 | 185.0 | 35.1 | 2.18 | 5 | 26 7/8 | 1 11/16 | 32 1/8 | 3 | 3/8 | 5 1/2 |
| 29.82 | 108 | 31.77 | 10.484 | 0.760 | .548 | 4461.0 | 299.2 | 11.85 | 135.1 | 25.8 | 2.06 | 5 | 26 7/8 | 1 1/2 | 31 5/8 | 2 3/4 | 3/8 | 5 1/2 |
| **Section CB 272; 27 × 14 in.** | | | | | | | | | | | | | | | | | | |
| 27.31 | 177 | 52.10 | 14.090 | 1.190 | 0.725 | 6728.6 | 492.8 | 11.36 | 518.9 | 73.7 | 3.16 | 6 3/4 | 23 | 2 1/8 | 30 3/4 | 3 1/4 | 7/16 | 5 1/2 |
| 27.12 | 163 | 47.93 | 14.035 | 1.095 | .670 | 6141.5 | 452.9 | 11.32 | 468.7 | 66.8 | 3.13 | 6 3/4 | 23 | 2 1/16 | 30 5/8 | 3 1/4 | 7/16 | 5 1/2 |
| 27.00 | 154 | 45.30 | 14.000 | 1.035 | .635 | 5775.8 | 427.8 | 11.29 | 437.6 | 62.5 | 3.11 | 6 3/4 | 23 | 2 | 30 1/2 | 3 1/4 | 3/8 | 5 1/2 |
| 26.88 | 145 | 42.68 | 13.965 | 0.975 | .600 | 5414.3 | 402.9 | 11.26 | 406.9 | 58.3 | 3.09 | 6 3/4 | 23 | 1 15/16 | 30 3/8 | 3 1/4 | 3/8 | 5 1/2 |
| **Section CB 271; 27 × 10 in.** 4 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 27.28 | 114 | 33.53 | 10.070 | 0.932 | 0.570 | 4080.5 | 299.2 | 11.03 | 149.6 | 29.7 | 2.11 | 4 3/4 | 24 | 1 5/8 | 29 1/8 | 2 3/4 | 3/8 | 5 1/2 |
| 26.84 | 91 | 26.77 | 9.983 | .712 | .483 | 3129.2 | 233.2 | 10.81 | 109.0 | 21.8 | 2.02 | 4 3/4 | 24 | 1 7/16 | 28 3/4 | 2 3/4 | 5/16 | 5 1/2 |
| **Section CB 243; 24 × 14 in.** | | | | | | | | | | | | | | | | | | |
| 24.72 | 160 | 47.04 | 14.091 | 1.135 | 0.656 | 5110.3 | 413.5 | 10.42 | 492.6 | 69.9 | 3.23 | 6 3/4 | 20 3/4 | 2 | 28 1/2 | 3 1/4 | 3/8 | 5 1/2 |
| 24.56 | 150 | 44.10 | 14.063 | 1.055 | .628 | 4733.5 | 385.5 | 10.36 | 452.5 | 64.3 | 3.20 | 6 3/4 | 20 3/4 | 1 7/8 | 28 1/4 | 3 1/4 | 3/8 | 5 1/2 |
| 24.41 | 140 | 41.16 | 14.029 | 0.980 | .594 | 4376.1 | 358.6 | 10.31 | 414.5 | 59.1 | 3.17 | 6 3/4 | 20 3/4 | 1 13/16 | 28 1/8 | 3 | 3/8 | 5 1/2 |
| 24.25 | 130 | 38.21 | 14.000 | 0.900 | .565 | 4009.5 | 330.7 | 10.24 | 375.2 | 53.6 | 3.13 | 6 3/4 | 20 3/4 | 1 3/4 | 28 | 3 | 3/8 | 5 1/2 |
| **Section CB 242; 24 × 12 in.** 3 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 24.31 | 120 | 35.29 | 12.088 | 0.930 | 0.556 | 3635.3 | 299.1 | 10.15 | 254.0 | 42.0 | 2.68 | 5 3/4 | 20 7/8 | 1 11/16 | 27 1/8 | 3 | 3/8 | 5 1/2 |
| 24.00 | 100 | 29.43 | 12.000 | .775 | .468 | 2987.3 | 248.9 | 10.08 | 203.5 | 33.9 | 2.63 | 5 3/4 | 20 7/8 | 1 9/16 | 26 7/8 | 2 3/4 | 5/16 | 5 1/2 |
| **Section CB 241; 24 × 9 in.** 4 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 24.29 | 94 | 27.63 | 9.061 | 0.872 | 0.516 | 2683.0 | 220.9 | 9.85 | 102.2 | 22.6 | 1.92 | 4 1/4 | 21 3/8 | 1 7/16 | 25 7/8 | 2 3/4 | 5/16 | 5 1/2 |
| 23.87 | 74 | 21.77 | 8.975 | .662 | .430 | 2033.8 | 170.4 | 9.67 | 73.8 | 16.5 | 1.84 | 4 1/4 | 21 3/8 | 1 1/4 | 25 1/2 | 2 1/2 | 5/16 | 5 1/2 |
| **Section CB 213; 21 × 13 in.** | | | | | | | | | | | | | | | | | | |
| 21.46 | 142 | 41.76 | 13.132 | 1.095 | 0.659 | 3403.1 | 317.2 | 9.03 | 385.9 | 58.8 | 3.04 | 6 1/4 | 17 3/4 | 1 7/8 | 25 1/4 | 3 | 7/16 | 5 1/2 |
| 21.31 | 132 | 38.81 | 13.087 | 1.020 | .614 | 3141.6 | 294.8 | 9.00 | 353.8 | 54.1 | 3.02 | 6 1/4 | 17 3/4 | 1 3/4 | 25 | 3 | 3/8 | 5 1/2 |
| 21.16 | 122 | 35.85 | 13.040 | 0.945 | .567 | 2883.2 | 272.5 | 8.97 | 322.1 | 49.4 | 3.00 | 6 1/4 | 17 3/4 | 1 11/16 | 24 7/8 | 3 | 3/8 | 5 1/2 |
| 21.00 | 112 | 32.93 | 13.000 | 0.865 | .527 | 2620.6 | 249.6 | 8.92 | 289.7 | 44.6 | 2.96 | 6 1/4 | 17 3/4 | 1 5/8 | 24 3/4 | 3 | 5/16 | 5 1/2 |
| **Section CB 212; 21 × 9 in.** 4 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 21.29 | 103 | 30.27 | 9.071 | 1.010 | 0.608 | 2268.0 | 213.1 | 8.66 | 119.9 | 26.4 | 1.99 | 4 1/4 | 18 | 1 5/8 | 23 1/8 | 3 | 3/8 | 5 1/2 |
| 20.86 | 82 | 24.10 | 8.962 | 0.795 | .499 | 1752.4 | 168.0 | 8.53 | 89.6 | 20.0 | 1.93 | 4 1/4 | 18 | 1 7/16 | 22 3/4 | 2 3/4 | 5/16 | 5 1/2 |
| **Section CB 211; 21 × 8 1/4 in.** 4 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 21.24 | 73 | 21.46 | 8.295 | 0.740 | 0.455 | 1600.3 | 150.7 | 8.64 | 66.2 | 16.0 | 1.76 | 4 | 18 5/8 | 1 5/16 | 22 7/8 | 2 1/2 | 5/16 | 5 1/2 |
| 20.91 | 59 | 17.36 | 8.230 | .575 | .390 | 1246.8 | 119.3 | 8.47 | 49.2 | 12.0 | 1.68 | 4 | 18 5/8 | 1 1/8 | 22 1/2 | 2 1/2 | 1/4 | 5 1/2 |
| **Section CB 183; 18 × 11 3/4 in.** | | | | | | | | | | | | | | | | | | |
| 18.64 | 124 | 36.45 | 11.889 | 1.071 | 0.651 | 2227.1 | 239.0 | 7.82 | 281.9 | 47.4 | 2.78 | 5 5/8 | 15 1/8 | 1 3/4 | 22 1/8 | 3 | 3/8 | 5 1/2 |
| 18.48 | 114 | 33.51 | 11.833 | 0.991 | .595 | 2033.8 | 220.1 | 7.79 | 255.6 | 43.2 | 2.76 | 5 5/8 | 15 1/8 | 1 11/16 | 22 | 3 | 3/8 | 5 1/2 |
| 18.32 | 105 | 30.86 | 11.792 | .911 | .554 | 1852.5 | 202.2 | 7.75 | 231.0 | 39.2 | 2.73 | 5 5/8 | 15 1/8 | 1 5/8 | 21 7/8 | 2 3/4 | 3/8 | 5 1/2 |
| 18.16 | 96 | 28.22 | 11.750 | .831 | .512 | 1674.7 | 184.4 | 7.70 | 206.8 | 35.2 | 2.71 | 5 5/8 | 15 1/8 | 1 1/2 | 21 3/4 | 2 3/4 | 5/16 | 5 1/2 |
| **Section CB 182; 18 × 8 3/4 in.** 4 weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 18.32 | 85 | 24.97 | 8.838 | 0.911 | 0.526 | 1429.9 | 156.1 | 7.57 | 99.4 | 22.5 | 2.00 | 4 1/8 | 15 3/8 | 1 1/2 | 20 3/8 | 2 3/4 | 5/16 | 5 1/2 |
| 17.87 | 64 | 18.80 | 8.715 | .686 | .403 | 1045.8 | 117.0 | 7.46 | 70.3 | 16.1 | 1.93 | 4 1/8 | 15 3/8 | 1 1/4 | 20 | 2 1/2 | 1/4 | 5 1/2 |

*Table continued on page 17-52*





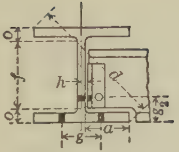
## Table 3.—Properties of American Standard Wide Flange Beams—*Continued*

(Carnegie-Illinois Steel Corporation, Pittsburgh)

| Depth of Section, in. | Weight per Foot, lb. | Area of Section, sq. in. | Flange Wide, in. | Flange Thick, in. | Web Thickness, in. | Axis A–A Moment of Inertia, $I$, in.$^4$ | Axis A–A Section Modulus, $S$, in.$^3$ | Axis A–A Radius of Gyration, $r$, in. | Axis B–B Moment of Inertia, $I'$, in.$^4$ | Axis B–B Section Modulus, $S'$, in.$^3$ | Axis B–B Radius of Gyration, $r'$, in. | $a$, in. | $f$, in. | $o$, in. | Diagonal, $d$, in. | $g_2$, min., in. | Clearance, $h$, in. | Gage, $g$, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Section CB 181; 18 × 7 1/2 in. 3 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | | | |
| 18.12 | 55 | 16.19 | 7.532 | 0.630 | 0.390 | 889.9 | 98.2 | 7.41 | 42.0 | 11.1 | 1.61 | 3 5/8 | 15 7/8 | 1 1/8 | 19 5/8 | 2 1/2 | 1/4 | 3 1/2 |
| 17.90 | 47 | 13.81 | 7.492 | .520 | .350 | 736.4 | 82.3 | 7.30 | 33.5 | 9.0 | 1.56 | 3 5/8 | 15 7/8 | 1 | 19 3/8 | 2 1/4 | 1/4 | 3 1/2 |
| **Section CB 163; 16 × 11 1/2 in.** | | | | | | | | | | | | | | | | | | |
| 16.64 | 114 | 33.51 | 11.629 | 1.035 | 0.631 | 1642.6 | 197.4 | 7.00 | 254.6 | 43.8 | 2.76 | 5 1/2 | 13 1/8 | 1 3/4 | 20 3/8 | 3 | 3/8 | 5 1/2 |
| 16.48 | 105 | 30.87 | 11.582 | 0.955 | .584 | 1497.5 | 181.7 | 6.96 | 230.7 | 39.8 | 2.73 | 5 1/2 | 13 1/8 | 1 11/16 | 20 1/4 | 3 | 3/8 | 5 1/2 |
| 16.32 | 96 | 28.22 | 11.533 | .875 | .535 | 1355.1 | 166.1 | 6.93 | 207.2 | 35.9 | 2.71 | 5 1/2 | 13 1/8 | 1 5/8 | 20 | 2 3/4 | 3/8 | 5 1/2 |
| 16.16 | 88 | 25.87 | 11.502 | .795 | .504 | 1222.6 | 151.3 | 6.87 | 185.2 | 32.2 | 2.67 | 5 1/2 | 13 1/8 | 1 1/2 | 19 7/8 | 2 3/4 | 5/16 | 5 1/2 |
| **Section CB 162; 16 × 8 1/2 in. 4 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | | | |
| 16.32 | 78 | 22.92 | 8.586 | 0.875 | 0.529 | 1042.6 | 127.8 | 6.74 | 87.5 | 20.4 | 1.95 | 4 | 13 3/8 | 1 1/2 | 18 1/2 | 2 3/4 | 5/16 | 5 1/2 |
| 15.86 | 58 | 17.04 | 8.464 | .645 | .407 | 746.4 | 94.1 | 6.62 | 60.5 | 14.3 | 1.88 | 4 | 13 3/8 | 1 1/4 | 18 | 2 1/2 | 5/16 | 5 1/2 |
| **Section CB 161; 16 × 7 in. 4 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | | | |
| 16.25 | 50 | 14.70 | 7.073 | 0.628 | 0.380 | 655.4 | 80.7 | 6.68 | 34.8 | 9.8 | 1.54 | 3 3/8 | 14 | 1 1/8 | 17 3/4 | 2 1/2 | 1/4 | 3 1/2 |
| 15.85 | 36 | 10.59 | 6.992 | .428 | .299 | 446.3 | 56.3 | 6.49 | 22.1 | 6.3 | 1.45 | 3 3/8 | 14 | 15/16 | 17 3/8 | 2 1/4 | 1/4 | 3 1/2 |
| **Section CB 146; 14 × 16 in. 28 weights rolled; alternate sizes listed** | | | | | | | | | | | | | | | | | | |
| 18.69 | 426 | 125.25 | 16.695 | 3.033 | 1.875 | 6610.3 | 707.4 | 7.26 | 2359.5 | 282.7 | 4.34 | 7 3/8 | 11 3/8 | 3 5/8 | 25 1/8 | 5 | 1 | 3—5 1/2—3 |
| 18.31 | 398 | 116.98 | 16.590 | 2.843 | 1.770 | 6013.7 | 656.9 | 7.17 | 2169.7 | 261.6 | 4.31 | 7 3/8 | 11 3/8 | 3 7/16 | 24 3/4 | 4 3/4 | 15/16 | |
| 17.94 | 370 | 108.78 | 16.475 | 2.658 | 1.655 | 5454.2 | 608.1 | 7.08 | 1986.0 | 241.1 | 4.27 | 7 3/8 | 11 3/8 | 3 1/4 | 24 3/8 | 4 1/2 | 7/8 | |
| 17.56 | 342 | 100.59 | 16.365 | 2.468 | 1.545 | 4911.5 | 559.4 | 6.99 | 1806.9 | 220.8 | 4.24 | 7 3/8 | 11 3/8 | 3 1/16 | 24 | 4 1/4 | 7/8 | |
| 17.19 | 314 | 92.30 | 16.235 | 2.283 | 1.415 | 4399.4 | 511.9 | 6.90 | 1631.4 | 201.0 | 4.20 | 7 3/8 | 11 3/8 | 2 7/8 | 23 3/4 | 4 1/4 | 13/16 | |
| 16.81 | 287 | 84.37 | 16.130 | 2.093 | 1.310 | 3912.1 | 465.5 | 6.81 | 1466.5 | 181.8 | 4.17 | 7 3/8 | 11 3/8 | 2 11/16 | 23 3/8 | 4 | 3/4 | |
| 16.50 | 264 | 77.63 | 16.025 | 1.938 | 1.205 | 3526.0 | 427.4 | 6.74 | 1331.2 | 166.1 | 4.14 | 7 3/8 | 11 3/8 | 2 9/16 | 23 | 3 3/4 | 11/16 | |
| 16.25 | 246 | 72.33 | 15.945 | 1.813 | 1.125 | 3228.9 | 397.4 | 6.68 | 1226.6 | 153.9 | 4.12 | 7 3/8 | 11 3/8 | 2 7/16 | 22 7/8 | 3 3/4 | 5/8 | |
| 16.00 | 228 | 67.06 | 15.865 | 1.688 | 1.045 | 2942.4 | 367.8 | 6.62 | 1124.8 | 141.8 | 4.10 | 7 3/8 | 11 3/8 | 2 5/16 | 22 5/8 | 3 1/2 | 5/8 | |
| 15.75 | 211 | 62.07 | 15.800 | 1.563 | 0.980 | 2671.4 | 339.2 | 6.56 | 1028.6 | 130.2 | 4.07 | 7 3/8 | 11 3/8 | 2 3/16 | 22 3/8 | 3 1/2 | 9/16 | |
| 15.50 | 193 | 56.73 | 15.710 | 1.438 | .890 | 2402.4 | 310.0 | 6.51 | 930.1 | 118.4 | 4.05 | 7 3/8 | 11 3/8 | 2 1/16 | 22 1/8 | 3 1/4 | 1/2 | |
| 15.25 | 176 | 51.73 | 15.640 | 1.313 | .820 | 2149.6 | 281.9 | 6.45 | 837.9 | 107.1 | 4.02 | 7 3/8 | 11 3/8 | 1 15/16 | 21 7/8 | 3 1/4 | 1/2 | |
| 15.00 | 158 | 46.47 | 15.550 | 1.188 | .730 | 1900.6 | 253.4 | 6.40 | 745.0 | 95.8 | 4.00 | 7 3/8 | 11 3/8 | 1 13/16 | 21 5/8 | 3 | 7/16 | |
| 14.75 | 142 | 41.85 | 15.500 | 1.063 | .680 | 1672.2 | 226.7 | 6.32 | 660.1 | 85.2 | 3.97 | 7 3/8 | 11 3/8 | 1 11/16 | 21 1/2 | 3 | 7/16 | |
| 16.81 | 320* | 94.12 | 16.710 | 2.093 | 1.890 | 4141.7 | 492.8 | 6.63 | 1635.1 | 195.7 | 4.17 | 7 3/8 | 11 3/8 | 2 11/16 | 23 3/4 | 4 | 1 | |

| | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Section CB 145; 14 × 14 1/2 in. 7 weights rolled; alternate sizes listed** | | | | | | | | | | | | | | | | |
| 14.75 | 136 | 39.98 | 14.740 | 1.063 | 0.660 | 1593.0 | 216.0 | 6.31 | 567.7 | 77.0 | 3.77 | 7 | 11 3/8 | 1 11/16 | 20 7/8 | 3 | 7/16 | 5 1/2 |
| 14.50 | 119 | 34.99 | 14.650 | 0.938 | .570 | 1373.1 | 189.4 | 6.26 | 491.8 | 67.1 | 3.75 | 7 | 11 3/8 | 1 9/16 | 20 5/8 | 2 3/4 | 3/8 | 5 1/2 |
| 14.25 | 103 | 30.26 | 14.575 | .813 | .495 | 1165.8 | 163.6 | 6.21 | 419.7 | 57.6 | 3.72 | 7 | 11 3/8 | 1 7/16 | 20 1/2 | 2 3/4 | 5/16 | 5 1/2 |
| 14.00 | 87 | 25.56 | 14.500 | .688 | .420 | 966.9 | 138.1 | 6.15 | 349.7 | 48.2 | 3.70 | 7 | 11 3/8 | 1 5/16 | 20 1/4 | 2 1/2 | 5/16 | 5 1/2 |
| **Section CB 144; 14 × 12 in.** | | | | | | | | | | | | | | | | |
| 14.18 | 84 | 24.71 | 12.023 | 0.778 | 0.451 | 928.4 | 130.9 | 6.13 | 225.5 | 37.5 | 3.02 | 5 3/4 | 11 3/8 | 1 3/8 | 18 5/8 | 2 3/4 | 5/16 | 5 1/2 |
| 14.06 | 78 | 22.94 | 12.000 | .718 | .428 | 851.2 | 121.1 | 6.09 | 206.9 | 34.5 | 3.00 | 5 3/4 | 11 3/8 | 1 5/16 | 18 1/2 | 2 1/2 | 5/16 | 5 1/2 |
| **Section CB 143; 14 × 10 in. 3 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | |
| 14.19 | 74 | 21.76 | 10.072 | 0.783 | 0.450 | 796.8 | 112.3 | 6.05 | 133.5 | 26.5 | 2.48 | 4 3/4 | 11 3/8 | 1 3/8 | 17 1/2 | 2 3/4 | 5/16 | 5 1/2 |
| 13.91 | 61 | 17.94 | 10.000 | .643 | .378 | 641.5 | 92.2 | 5.98 | 107.3 | 21.5 | 2.45 | 4 3/4 | 11 3/8 | 1 1/4 | 17 1/8 | 2 1/2 | 1/4 | 5 1/2 |
| **Section CB 142; 14 × 8 in. 4 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | |
| 14.06 | 58 | 17.06 | 8.098 | 0.718 | 0.406 | 597.9 | 85.0 | 5.92 | 63.7 | 15.7 | 1.93 | 3 7/8 | 11 3/8 | 1 5/16 | 16 1/4 | 2 1/2 | 1/4 | 5 1/2 |
| 13.68 | 43 | 12.65 | 8.000 | .528 | .308 | 429.0 | 62.7 | 5.82 | 45.1 | 11.3 | 1.89 | 3 7/8 | 11 3/8 | 1 1/8 | 15 7/8 | 2 1/2 | 1/4 | 5 1/2 |
| **Section CB 141; 14 × 6 3/4 in. 4 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | |
| 14.24 | 42 | 12.34 | 6.801 | 0.573 | 0.338 | 432.2 | 60.7 | 5.92 | 28.1 | 8.3 | 1.51 | 3 1/4 | 12 1/8 | 1 1/16 | 15 3/4 | 2 1/4 | 1/4 | 3 1/2 |
| 13.86 | 30 | 8.81 | 6.733 | .383 | .270 | 289.6 | 41.8 | 5.73 | 17.5 | 5.2 | 1.41 | 3 1/4 | 12 1/8 | 7/8 | 15 1/2 | 2 1/4 | 3/16 | 3 1/2 |
| **Section CB 124; 12 × 12 in. 13 weights rolled; alternate sizes listed** | | | | | | | | | | | | | | | | |
| 14.38 | 190 | 55.86 | 12.670 | 1.736 | 1.060 | 1892.5 | 263.2 | 5.82 | 589.7 | 93.1 | 3.25 | 5 3/4 | 9 3/4 | 2 5/16 | 19 1/4 | 3 3/4 | 5/8 | 5 1/2 |
| 13.88 | 161 | 47.38 | 12.515 | 1.486 | 0.905 | 1541.8 | 222.2 | 5.70 | 486.2 | 77.7 | 3.20 | 5 3/4 | 9 3/4 | 2 1/16 | 18 3/4 | 3 1/2 | 1/2 | 5 1/2 |
| 13.38 | 133 | 39.11 | 12.365 | 1.236 | .755 | 1221.2 | 182.5 | 5.59 | 389.9 | 63.1 | 3.16 | 5 3/4 | 9 3/4 | 1 13/16 | 18 1/4 | 3 1/4 | 7/16 | 5 1/2 |
| 12.88 | 106 | 31.19 | 12.230 | 0.986 | .620 | 930.7 | 144.5 | 5.46 | 300.9 | 49.2 | 3.11 | 5 3/4 | 9 3/4 | 1 9/16 | 17 7/8 | 3 | 3/8 | 5 1/2 |
| 12.62 | 92 | 27.06 | 12.155 | .856 | .545 | 788.9 | 125.0 | 5.40 | 256.4 | 42.2 | 3.08 | 5 3/4 | 9 3/4 | 1 7/16 | 17 1/2 | 2 3/4 | 3/8 | 5 1/2 |
| 12.38 | 79 | 23.22 | 12.080 | .736 | .470 | 663.0 | 107.1 | 5.34 | 216.4 | 35.8 | 3.05 | 5 3/4 | 9 3/4 | 1 5/16 | 17 3/8 | 2 3/4 | 5/16 | 5 1/2 |
| 12.12 | 65 | 19.11 | 12.000 | .606 | .390 | 533.4 | 88.0 | 5.28 | 174.6 | 29.1 | 3.02 | 5 3/4 | 9 3/4 | 1 3/16 | 17 1/8 | 2 1/2 | 1/4 | 5 1/2 |
| **Section CB 123; 12 × 10 in. 3 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | |
| 12.31 | 64 | 18.83 | 10.060 | 0.701 | 0.405 | 528.3 | 85.8 | 5.29 | 119.0 | 23.7 | 2.51 | 4 7/8 | 9 3/4 | 1 5/16 | 15 7/8 | 2 1/2 | 1/4 | 5 1/2 |
| 12.06 | 53 | 15.59 | 10.000 | .576 | .345 | 426.2 | 70.7 | 5.23 | 96.1 | 19.2 | 2.48 | 4 7/8 | 9 3/4 | 1 3/16 | 15 5/8 | 2 1/2 | 1/4 | 5 1/2 |
| **Section CB 122; 12 × 8 in. 3 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | |
| 12.19 | 50 | 14.71 | 8.077 | 0.641 | 0.371 | 394.5 | 64.7 | 5.18 | 56.4 | 14.0 | 1.96 | 3 7/8 | 9 3/4 | 1 1/4 | 14 5/8 | 2 1/2 | 1/4 | 5 1/2 |
| 11.94 | 40 | 11.77 | 8.000 | .516 | .294 | 310.1 | 51.9 | 5.13 | 44.1 | 11.0 | 1.94 | 3 7/8 | 9 3/4 | 1 1/8 | 14 3/8 | 2 1/2 | 1/4 | 5 1/2 |
| **Section CB 121; 12 × 6 1/2 in. 4 weights rolled; maximum and minimum sizes listed** | | | | | | | | | | | | | | | | |
| 12.24 | 36 | 10.59 | 6.565 | 0.540 | 0.305 | 280.8 | 45.9 | 5.15 | 23.7 | 7.2 | 1.50 | 3 1/8 | 10 3/8 | 15/16 | 14 | 2 1/4 | 1/4 | 3 1/2 |
| 11.87 | 25 | 7.39 | 6.500 | .355 | .240 | 183.4 | 30.9 | 4.98 | 14.5 | 4.5 | 1.40 | 3 1/8 | 10 3/8 | 3/4 | 13 5/8 | 2 1/4 | 3/16 | 3 1/2 |
| **Section CB 103; 10 × 10 in. 11 weights rolled; alternate sizes listed** | | | | | | | | | | | | | | | | |
| 11.88 | 136 | 40.03 | 10.575 | 1.498 | 0.915 | 917.2 | 154.4 | 4.79 | 295.9 | 56.0 | 2.72 | 4 7/8 | 7 7/8 | 2 | 16 | 3 1/4 | 9/16 | 5 1/2 |
| 11.38 | 112 | 32.92 | 10.415 | 1.248 | .755 | 718.7 | 126.3 | 4.67 | 235.4 | 45.2 | 2.67 | 4 7/8 | 7 7/8 | 1 3/4 | 15 1/2 | 3 | 7/16 | 5 1/2 |
| 10.88 | 89 | 26.19 | 10.275 | 0.998 | .615 | 542.4 | 99.7 | 4.55 | 180.6 | 35.2 | 2.63 | 4 7/8 | 7 7/8 | 1 1/2 | 15 | 2 3/4 | 3/8 | 5 1/2 |
| 10.50 | 72 | 21.18 | 10.170 | .808 | .510 | 420.7 | 80.1 | 4.46 | 141.8 | 27.9 | 2.59 | 4 7/8 | 7 7/8 | 1 5/16 | 14 5/8 | 2 3/4 | 5/16 | 5 1/2 |
| 10.25 | 60 | 17.66 | 10.075 | .683 | .415 | 343.7 | 67.1 | 4.41 | 116.5 | 23.1 | 2.57 | 4 7/8 | 7 7/8 | 1 3/16 | 14 3/8 | 2 1/2 | 5/16 | 5 1/2 |
| 10.00 | 49 | 14.40 | 10.000 | .558 | .340 | 272.9 | 54.6 | 4.35 | 93.0 | 18.6 | 2.54 | 4 7/8 | 7 7/8 | 1 1/16 | 14 1/8 | 2 1/2 | 1/4 | 5 1/2 |

*Table continued on page 17–54*



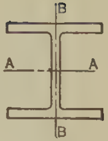


**Table 3.—Properties of American Standard Wide Flange Beams—*Continued***

(Carnegie-Illinois Steel Corporation, Pittsburgh)

| Depth of Section, in. | Weight per Foot, lb. | Area of Section, sq. in. | Flange | | Web Thickness, in. | Elements of Sections | | | | | | Dimensions for Detailing | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Axis A–A | | | Axis B–B | | | | | | | | | |
| | | | Wide, in. | Thick, in. | | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | Moment of Inertia, $I'$, in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r'$, in. | $a$, in. | $f$, in. | $o$, in. | Diagonal, $d$, in. | $g_2$, min., in. | Clearance, $h$, in. | Gage, $g$, in. |
| Section CB **102**; **10** × **8** in. **4** weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 10.12 | 45 | 13.24 | 8.022 | 0.618 | 0.350 | 248.6 | 49.1 | 4.33 | 53.2 | 13.3 | 2.00 | 3 7/8 | 7 7/8 | 1 1/8 | 13 | 2 1/2 | 1/4 | 5 1/2 |
| 9.75 | 33 | 9.71 | 7.964 | .433 | .292 | 170.9 | 35.0 | 4.20 | 36.5 | 9.2 | 1.94 | 3 7/8 | 7 7/8 | 15/16 | 12 5/8 | 2 1/4 | 1/4 | 5 1/2 |
| Section CB **101**; **10** × **5 3/4** in. **4** weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 10.22 | 29 | 8.53 | 5.799 | 0.500 | 0.289 | 157.3 | 30.8 | 4.29 | 15.2 | 5.2 | 1.34 | 2 3/4 | 8 1/2 | 7/8 | 11 3/4 | 2 1/4 | 1/4 | 2 3/4 |
| 9.90 | 21 | 6.19 | 5.750 | .340 | .240 | 106.3 | 21.5 | 4.14 | 9.7 | 3.4 | 1.25 | 2 3/4 | 8 1/2 | 11/16 | 11 1/2 | 2 | 3/16 | 2 3/4 |
| Section CB **83**; **8** × **8** in. **7** weights rolled; alternate sizes listed | | | | | | | | | | | | | | | | | | |
| 9.00 | 67 | 19.70 | 8.287 | 0.933 | 0.575 | 271.8 | 60.4 | 3.71 | 88.6 | 21.4 | 2.12 | 3 7/8 | 6 3/8 | 1 5/16 | 12 1/4 | 2 3/4 | 3/8 | 5 1/2 |
| 8.50 | 48 | 14.11 | 8.117 | .683 | .405 | 183.7 | 43.2 | 3.61 | 60.9 | 15.0 | 2.08 | 3 7/8 | 6 3/8 | 1 1/16 | 11 7/8 | 2 1/2 | 1/4 | 5 1/2 |
| 8.12 | 35 | 10.30 | 8.027 | .493 | .315 | 126.5 | 31.1 | 3.50 | 42.5 | 10.6 | 2.03 | 3 7/8 | 6 3/8 | 7/8 | 11 1/2 | 2 1/4 | 1/4 | 5 1/2 |
| 8.00 | 31 | 9.12 | 8.000 | .433 | .288 | 109.7 | 27.4 | 3.47 | 37.0 | 9.2 | 2.01 | 3 7/8 | 6 3/8 | 13/16 | 11 3/8 | 2 1/4 | 1/4 | 5 1/2 |
| Section CB **82**; **8** × **6 1/2** in. | | | | | | | | | | | | | | | | | | |
| 8.03 | 27 | 7.93 | 6.528 | 0.448 | 0.273 | 94.1 | 23.4 | 3.44 | 20.8 | 6.4 | 1.62 | 3 1/8 | 6 3/8 | 7/8 | 10 3/8 | 2 1/4 | 3/16 | 3 1/2 |
| 7.93 | 24 | 7.06 | 6.500 | .398 | .245 | 82.5 | 20.8 | 3.42 | 18.2 | 5.6 | 1.61 | 3 1/8 | 6 3/8 | 13/16 | 10 1/4 | 2 1/4 | 3/16 | 3 1/2 |
| Section CB **81**; **8** × **5 1/4** in. **3** weights rolled; maximum and minimum sizes listed | | | | | | | | | | | | | | | | | | |
| 8.19 | 21 | 6.18 | 5.272 | 0.403 | 0.252 | 73.8 | 18.0 | 3.45 | 9.13 | 3.5 | 1.22 | 2 1/2 | 6 3/4 | 3/4 | 9 3/4 | 2 1/4 | 3/16 | 2 3/4 |
| 8.00 | 17 | 5.00 | 5.250 | .308 | .230 | 56.4 | 14.1 | 3.36 | 6.72 | 2.6 | 1.16 | 2 1/2 | 6 3/4 | 5/8 | 9 5/8 | 2 1/4 | 3/16 | 2 3/4 |

## Table 4.—Allowable Uniform Load, in Kips, on Wide-flange Sections Used as Beams*

Maximum and minimum weights

| Span, ft. | CB 362, 36″×16 1/2″ | | CB 361, 36″×12″ | | CB 332, 33″×15 3/4″ | | CB 331, 33″×11 1/2″ | | CB 302, 30″×15″ | | CB 301, 30″×10 1/2″ | | CB 272, 27″×14″ | | CB 271, 27″×10″ | | CB 243, 24″×14″ | | CB 242, 24″×12″ | | CB 241, 24″×9″ | | Coefficient of Deflection |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 300 lb. | 230 lb. | 194 lb. | 150 lb. | 240 lb. | 200 lb. | 152 lb. | 125 lb. | 210 lb. | 172 lb. | 132 lb. | 108 lb. | 177 lb. | 145 lb. | 114 lb. | 91 lb. | 160 lb. | 130 lb. | 120 lb. | 100 lb. | 94 lb. | 74 lb. | |
| .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 311.1 | .... | .... | .... | .... | 300.8 | 246.3 | .... |
| 9 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 392.2 | .... | .... | 373.2 | 310.9 | .... | .... | .... | .... | 294.5 | 227.2 | 1.51 |
| 10 | .... | .... | .... | .... | .... | .... | .... | 451.4 | .... | .... | 447.2 | 359.0 | .... | .... | 359.0 | 279.8 | .... | .... | .... | .... | 265.1 | 204.5 | 1.86 |
| 11 | .... | .... | 674.2 | 537.6 | .... | .... | 510.5 | 420.1 | .... | .... | 414.2 | 326.4 | .... | .... | 326.4 | 254.4 | .... | .... | 324.4 | 269.6 | 241.0 | 185.9 | 2.25 |
| 12 | .... | .... | 663.6 | 502.9 | .... | .... | 486.4 | 385.1 | .... | .... | 379.7 | 299.2 | 475.2 | 387.1 | 299.2 | 233.2 | 389.2 | 328.8 | 299.1 | 248.9 | 220.9 | 170.4 | 2.68 |
| 13 | .... | .... | 612.6 | 464.2 | .... | .... | 449.0 | 355.5 | 565.1 | 469.7 | 350.5 | 276.2 | 454.9 | 371.9 | 276.2 | 215.3 | 381.7 | 305.3 | 276.1 | 229.8 | 203.9 | 157.3 | 3.15 |
| 14 | .... | .... | 568.8 | 431.1 | 667.3 | 566.3 | 416.9 | 330.1 | 557.1 | 452.7 | 325.5 | 256.5 | 422.4 | 345.3 | 256.5 | 199.9 | 354.4 | 283.5 | 256.4 | 213.3 | 189.3 | 146.1 | 3.65 |
| 15 | 832.8 | 658.8 | 530.9 | 402.3 | 648.9 | 535.7 | 389.1 | 308.1 | 519.9 | 422.6 | 303.8 | 239.4 | 394.2 | 322.3 | 239.4 | 186.6 | 330.8 | 264.6 | 239.3 | 199.1 | 176.7 | 136.3 | 4.19 |
| 16 | 828.8 | 626.6 | 497.7 | 377.2 | 608.3 | 502.2 | 364.8 | 288.8 | 487.4 | 396.2 | 284.8 | 224.4 | 369.6 | 302.2 | 224.4 | 174.9 | 310.1 | 248.0 | 224.3 | 186.7 | 165.7 | 127.8 | 4.77 |
| 17 | 780.1 | 589.8 | 468.4 | 355.0 | 572.5 | 472.7 | 343.3 | 271.8 | 458.8 | 372.8 | 268.0 | 211.2 | 347.9 | 284.4 | 211.2 | 164.6 | 291.9 | 233.4 | 211.1 | 175.7 | 155.9 | 120.3 | 5.38 |
| 18 | 736.7 | 557.0 | 442.4 | 335.3 | 540.7 | 446.4 | 324.3 | 256.7 | 433.3 | 352.1 | 253.1 | 199.5 | 328.5 | 268.6 | 199.5 | 155.5 | 275.7 | 220.5 | 199.4 | 165.9 | 147.3 | 113.6 | 6.03 |
| 19 | 698.0 | 527.7 | 419.1 | 317.6 | 512.3 | 422.9 | 307.2 | 243.2 | 410.5 | 333.6 | 239.8 | 189.0 | 311.2 | 254.5 | 189.0 | 147.3 | 261.2 | 208.9 | 188.9 | 157.2 | 139.5 | 107.6 | 6.72 |
| 20 | 663.1 | 501.3 | 398.2 | 301.7 | 486.7 | 401.8 | 291.8 | 231.1 | 389.9 | 316.9 | 227.8 | 179.5 | 295.7 | 241.7 | 179.5 | 139.9 | 248.1 | 198.4 | 179.5 | 149.3 | 132.5 | 102.2 | 7.45 |
| 22 | 602.8 | 455.7 | 362.0 | 274.3 | 442.4 | 365.2 | 265.3 | 210.1 | 354.5 | 288.1 | 207.1 | 163.2 | 268.8 | 219.8 | 163.2 | 127.2 | 225.5 | 180.4 | 163.1 | 135.8 | 120.5 | 92.9 | 9.01 |
| 24 | 552.6 | 417.8 | 331.8 | 251.5 | 405.6 | 334.8 | 243.2 | 192.6 | 325.0 | 264.1 | 189.9 | 149.6 | 246.4 | 201.5 | 149.6 | 116.6 | 206.8 | 165.4 | 149.6 | 124.5 | 110.5 | 85.2 | 10.73 |
| 26 | 510.0 | 385.6 | 306.3 | 232.1 | 374.4 | 309.0 | 224.5 | 177.7 | 300.0 | 243.8 | 175.2 | 138.1 | 227.4 | 186.0 | 138.1 | 107.6 | 190.8 | 152.6 | 138.0 | 114.9 | 102.0 | 78.6 | 12.59 |
| 28 | 473.6 | 358.1 | 284.4 | 215.5 | 347.6 | 287.0 | 208.5 | 165.0 | 278.5 | 226.4 | 162.7 | 128.2 | 211.2 | 172.7 | 128.2 | 99.9 | 177.2 | 141.7 | 128.2 | 106.7 | 94.7 | 73.0 | 14.60 |
| 30 | 442.0 | 334.2 | 265.4 | 201.2 | 324.4 | 267.8 | 194.6 | 154.0 | 260.0 | 211.3 | 151.9 | 119.7 | 197.1 | 161.2 | 119.7 | 93.3 | 165.4 | 132.3 | 119.6 | 99.6 | 88.4 | 68.2 | 16.76 |
| 32 | 414.4 | 313.3 | 248.9 | 188.6 | 304.2 | 251.1 | 182.4 | 144.4 | 243.7 | 198.1 | 142.4 | 112.2 | 184.8 | 151.1 | 112.2 | 87.5 | 155.1 | 124.0 | 112.2 | 93.3 | 82.8 | 63.9 | 19.07 |
| 34 | 390.0 | 294.9 | 234.2 | 177.5 | 286.3 | 236.3 | 171.7 | 135.9 | 229.4 | 186.4 | 134.0 | 105.6 | 173.9 | 142.2 | 105.6 | 82.3 | 145.9 | 116.7 | 105.6 | 87.8 | 78.0 | 60.1 | 21.53 |
| 36 | 368.4 | 278.5 | 221.2 | 167.6 | 270.4 | 223.2 | 162.1 | 128.4 | 216.6 | 176.1 | 126.6 | 99.7 | 164.3 | 134.3 | 99.7 | 77.7 | 137.8 | 110.2 | 99.7 | 83.0 | 73.6 | 56.8 | 24.13 |
| 38 | 349.0 | 263.8 | 209.6 | 158.8 | 256.1 | 211.5 | 153.6 | 121.6 | 205.2 | 166.8 | 119.9 | 94.5 | 155.6 | 127.2 | 94.5 | 73.6 | 130.6 | 104.4 | 94.5 | 78.6 | 69.8 | 53.8 | 26.89 |
| 40 | 331.5 | 250.7 | 199.1 | 150.9 | 243.3 | 200.9 | 145.9 | 115.5 | 195.0 | 158.5 | 113.9 | 89.8 | 147.8 | 120.9 | 89.8 | 70.0 | 124.1 | 99.2 | 89.7 | 74.7 | 66.3 | 51.1 | 29.79 |
| 42 | 315.7 | 238.7 | 189.6 | 143.7 | 231.7 | 191.3 | 139.0 | 110.0 | 185.7 | 150.9 | 108.5 | 85.5 | 140.8 | 115.1 | 85.5 | 66.6 | 118.1 | 94.5 | 85.5 | 71.1 | 63.1 | 48.7 | 32.85 |
| 44 | 301.4 | 227.9 | 181.0 | 137.2 | 221.2 | 182.6 | 132.7 | 105.0 | 177.2 | 144.1 | 103.6 | 81.6 | 134.4 | 109.9 | 81.6 | 63.6 | 112.8 | 90.2 | 81.6 | 67.9 | 60.2 | 46.5 | 36.05 |
| 46 | 288.3 | 218.0 | 173.1 | 131.2 | 211.6 | 174.7 | 126.9 | 100.5 | 169.5 | 137.8 | 99.1 | 78.1 | 128.6 | 105.1 | 78.1 | 60.8 | 107.9 | 86.3 | 78.0 | 64.9 | 57.6 | 44.5 | 39.40 |
| 48 | 276.3 | 208.9 | 165.9 | 125.7 | 202.8 | 167.4 | 121.6 | 96.3 | 162.5 | 132.0 | 94.9 | 74.8 | 123.2 | 100.7 | 74.8 | 58.3 | 103.4 | 82.7 | 74.8 | 62.2 | 55.2 | 42.6 | 42.90 |
| 50 | 265.2 | 200.5 | 159.3 | 120.7 | 194.7 | 160.7 | 116.7 | 92.4 | 156.0 | 126.8 | 91.1 | 71.8 | 118.3 | 96.7 | 71.8 | 56.0 | .... | .... | .... | .... | .... | .... | 46.55 |
| 52 | 255.0 | 192.8 | 153.1 | 116.1 | 187.2 | 154.5 | 112.2 | 88.9 | 150.0 | 121.9 | 87.6 | 69.0 | 113.7 | 93.0 | 69.0 | 53.8 | .... | .... | .... | .... | .... | .... | 50.35 |
| 54 | 245.6 | 185.7 | 147.5 | 111.8 | 180.2 | 148.8 | 108.1 | 85.6 | 144.4 | 117.4 | 84.4 | 66.5 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 54.30 |
| 56 | 236.8 | 179.0 | 142.2 | 107.8 | 173.8 | 143.5 | 104.2 | 82.5 | 139.3 | 113.2 | 81.4 | 64.1 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 58.40 |
| 58 | 228.6 | 172.9 | 137.3 | 104.0 | 167.8 | 138.5 | 100.6 | 79.7 | 134.5 | 109.3 | 78.6 | 61.9 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 62.64 |
| 60 | 221.0 | 167.1 | 132.7 | 100.6 | 162.2 | 133.9 | 97.3 | 77.0 | 130.0 | 105.6 | 75.9 | 59.8 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 67.04 |
| 62 | 213.9 | 161.7 | 128.4 | 97.3 | 157.0 | 129.6 | 94.1 | 74.5 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 71.58 |
| 64 | 207.2 | 156.7 | 124.4 | 94.3 | 152.1 | 125.6 | 91.2 | 72.2 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 76.27 |
| 66 | 200.9 | 151.9 | 120.7 | 91.4 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 81.11 |

* See footnotes at end of table.

(*Table continued on page 17–56*)

## Table 4.—Allowable Uniform Load, in Kips, on Wide-flange Sections Used as Beams*—*Continued*

Maximum and minimum weights

| Span, ft. | CB 213, 21″×13″ | | CB 212, 21″×9″ | | CB 211, 21″×8 1/4″ | | CB 183, 18″×11 3/4″ | | CB 182, 18″×8 3/4″ | | CB 181, 18″×7 1/2″ | | CB 163, 16″×11 1/2″ | | CB 162, 16″×8 1/2″ | | CB 161, 16″×7″ | | CB 145, 14″×14 1/2″ | | CB 144, 14″×12″ | | Coefficient of Deflection |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 142 lb. | 112 lb. | 103 lb. | 82 lb. | 73 lb. | 59 lb. | 124 lb. | 96 lb. | 85 lb. | 64 lb. | 55 lb. | 47 lb. | 114 lb. | 88 lb. | 78 lb. | 58 lb. | 50 lb. | 36 lb. | 119 lb. | 87 lb. | 84 lb. | 78 lb. | |
| ..... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 113.7 | ...... | ...... | ...... | ...... | ...... |
| 6 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 169.6 | 150.4 | ...... | ...... | ...... | ...... | 148.2 | 112.6 | ...... | ...... | ...... | ...... | 0.67 |
| 7 | ...... | ...... | ...... | ...... | 231.9 | 195.7 | ...... | ...... | ...... | ...... | 168.3 | 141.1 | ...... | ...... | 207.2 | 154.9 | 138.3 | 96.5 | ...... | ...... | ...... | ...... | 0.91 |
| 8 | ...... | ...... | 310.7 | 249.8 | 226.1 | 179.0 | ...... | ...... | 231.3 | 172.8 | 147.3 | 123.5 | ...... | ...... | 191.7 | 141.2 | 121.1 | 84.5 | ...... | ...... | ...... | ...... | 1.19 |
| 9 | ...... | ...... | 284.1 | 224.0 | 200.9 | 159.1 | 291.2 | 223.2 | 208.1 | 156.0 | 130.9 | 109.7 | 252.0 | 195.5 | 170.4 | 125.5 | 107.6 | 75.1 | ...... | ...... | ...... | ...... | 1.51 |
| 10 | ...... | ...... | 255.7 | 201.6 | 180.8 | 143.2 | 286.8 | 221.3 | 187.3 | 140.4 | 117.8 | 98.8 | 236.9 | 181.6 | 153.4 | 112.9 | 96.8 | 67.6 | ...... | ...... | 153.5 | 144.4 | 1.86 |
| 11 | 339.4 | 265.6 | 232.5 | 183.3 | 164.4 | 130.1 | 260.7 | 201.2 | 170.3 | 127.6 | 107.1 | 89.8 | 215.3 | 165.1 | 139.4 | 102.7 | 88.0 | 61.4 | 198.4 | 141.1 | 142.8 | 132.1 | 2.25 |
| 12 | 317.2 | 249.6 | 213.1 | 168.0 | 150.7 | 119.3 | 239.0 | 184.4 | 156.1 | 117.0 | 98.2 | 82.3 | 197.4 | 151.3 | 127.8 | 94.1 | 80.7 | 56.3 | 189.4 | 138.1 | 130.9 | 121.1 | 2.68 |
| 13 | 292.8 | 230.4 | 196.7 | 155.1 | 139.1 | 110.1 | 220.6 | 170.2 | 144.1 | 108.0 | 90.6 | 76.0 | 182.2 | 139.7 | 118.0 | 86.9 | 74.5 | 52.0 | 174.8 | 127.5 | 120.8 | 111.8 | 3.15 |
| 14 | 271.9 | 213.9 | 182.7 | 144.0 | 129.2 | 102.3 | 204.9 | 158.1 | 133.8 | 100.3 | 84.2 | 70.5 | 169.2 | 129.7 | 109.5 | 80.7 | 69.2 | 48.3 | 162.3 | 118.4 | 112.2 | 103.8 | 3.65 |
| 15 | 253.8 | 199.7 | 170.5 | 134.4 | 120.6 | 95.4 | 191.2 | 147.5 | 124.9 | 93.6 | 78.6 | 65.8 | 157.9 | 121.0 | 102.2 | 75.3 | 64.6 | 45.0 | 151.5 | 110.5 | 104.7 | 96.9 | 4.19 |
| 16 | 237.9 | 187.2 | 159.8 | 126.0 | 113.0 | 89.5 | 179.3 | 138.3 | 117.1 | 87.8 | 73.7 | 61.7 | 148.1 | 113.5 | 95.9 | 70.6 | 60.5 | 42.2 | 142.1 | 103.6 | 98.2 | 90.8 | 4.77 |
| 17 | 223.9 | 176.2 | 150.4 | 118.6 | 106.4 | 84.2 | 168.7 | 130.2 | 110.2 | 82.6 | 69.3 | 58.1 | 139.3 | 106.8 | 90.2 | 66.4 | 57.0 | 39.7 | 133.7 | 97.5 | 92.4 | 85.5 | 5.38 |
| 18 | 211.5 | 166.4 | 142.1 | 112.0 | 100.5 | 79.5 | 159.3 | 122.9 | 104.1 | 78.0 | 65.5 | 54.9 | 131.6 | 100.9 | 85.2 | 62.7 | 53.8 | 37.5 | 126.3 | 92.1 | 87.3 | 80.7 | 6.03 |
| 19 | 200.3 | 157.6 | 134.6 | 106.1 | 95.2 | 75.3 | 150.9 | 116.5 | 98.6 | 73.9 | 62.0 | 52.0 | 124.7 | 95.6 | 80.7 | 59.4 | 51.0 | 35.6 | 119.6 | 87.2 | 82.7 | 76.5 | 6.72 |
| 20 | 190.3 | 149.8 | 127.9 | 100.8 | 90.4 | 71.6 | 143.4 | 110.6 | 93.7 | 70.2 | 58.9 | 49.4 | 118.4 | 90.8 | 76.7 | 56.5 | 48.4 | 33.8 | 113.6 | 82.9 | 78.5 | 72.7 | 7.45 |
| 21 | 181.3 | 142.6 | 121.8 | 96.0 | 86.1 | 68.2 | 136.6 | 105.4 | 89.2 | 66.9 | 56.1 | 47.0 | 112.8 | 86.5 | 73.0 | 53.8 | 46.1 | 32.2 | 108.2 | 78.9 | 74.8 | 69.2 | 8.21 |
| 22 | 173.0 | 136.1 | 116.2 | 91.6 | 82.2 | 65.1 | 130.4 | 100.6 | 85.1 | 63.8 | 53.6 | 44.9 | 107.7 | 82.5 | 69.7 | 51.3 | 44.0 | 30.7 | 103.3 | 75.3 | 71.4 | 66.1 | 9.01 |
| 23 | 165.5 | 130.2 | 111.2 | 87.7 | 78.6 | 62.2 | 124.7 | 96.2 | 81.4 | 61.0 | 51.2 | 42.9 | 103.0 | 78.9 | 66.7 | 49.1 | 42.1 | 29.4 | 98.8 | 72.1 | 68.3 | 63.2 | 9.85 |
| 24 | 158.6 | 124.8 | 106.6 | 84.0 | 75.4 | 59.7 | 119.5 | 92.2 | 78.1 | 58.5 | 49.1 | 41.2 | 98.7 | 75.7 | 63.9 | 47.1 | 40.4 | 28.2 | 94.7 | 69.1 | 65.5 | 60.6 | 10.73 |
| 25 | 152.3 | 119.8 | 102.3 | 80.6 | 72.3 | 57.3 | 114.7 | 88.5 | 74.9 | 56.2 | 47.1 | 39.5 | 94.8 | 72.6 | 61.3 | 45.2 | 38.7 | 27.0 | 90.9 | 66.3 | 62.8 | 58.1 | 11.64 |
| 26 | 146.4 | 115.2 | 98.4 | 77.5 | 69.6 | 55.1 | 110.3 | 85.1 | 72.0 | 54.0 | 45.3 | 38.0 | 91.1 | 69.8 | 59.0 | 43.4 | 37.2 | 26.0 | 87.4 | 63.7 | 60.4 | 55.9 | 12.59 |
| 27 | 141.0 | 110.9 | 94.7 | 74.7 | 67.0 | 53.0 | 106.2 | 82.0 | 69.4 | 52.0 | 43.6 | 36.6 | 87.7 | 67.2 | 56.8 | 41.8 | 35.9 | 25.0 | 84.2 | 61.4 | 58.2 | 53.8 | 13.57 |
| 28 | 135.9 | 107.0 | 91.3 | 72.0 | 64.6 | 51.1 | 102.4 | 79.0 | 66.9 | 50.1 | 42.1 | 35.3 | 84.6 | 64.8 | 54.8 | 40.3 | 34.6 | 24.1 | 81.2 | 59.2 | 56.1 | 51.9 | 14.60 |
| 29 | 131.3 | 103.3 | 88.2 | 69.5 | 62.4 | 49.4 | 98.9 | 76.3 | 64.6 | 48.4 | 40.6 | 34.1 | 81.7 | 62.6 | 52.9 | 38.9 | 33.4 | 23.3 | ...... | ...... | ...... | ...... | 15.66 |
| 30 | 126.9 | 99.8 | 85.2 | 67.2 | 60.3 | 47.7 | 95.6 | 73.8 | 62.4 | 46.8 | 39.3 | 32.9 | 79.0 | 60.5 | 51.1 | 37.6 | 32.3 | 22.5 | ...... | ...... | ...... | ...... | 16.76 |
| 32 | 119.0 | 93.6 | 79.9 | 63.0 | 56.5 | 44.7 | 89.6 | 69.2 | 58.5 | 43.9 | 36.8 | 30.9 | ...... | 56.7 | 47.9 | ...... | 30.3 | ...... | ...... | ...... | ...... | ...... | 19.07 |
| 34 | 112.0 | 88.1 | 75.2 | 59.3 | 53.2 | 42.1 | 84.4 | 65.1 | 55.1 | 41.3 | 34.7 | 29.0 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 21.53 |
| 36 | 105.7 | 83.2 | 71.0 | 56.0 | 50.2 | 39.8 | 79.7 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 24.13 |
| 38 | 100.2 | 78.8 | 67.3 | 53.1 | 47.6 | 37.7 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 26.89 |
| 40 | 95.2 | 74.9 | 63.9 | 50.4 | 45.2 | 35.8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 29.79 |
| 42 | 90.6 | 71.3 | 60.9 | ...... | 43.1 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 32.85 |

* See footnotes at end of table.

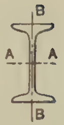
(*Table continued on page 17–57*)

### Table 4.—Allowable Uniform Load, in Kips, on Wide-flange Sections Used as Beams*—*Continued*

Maximum and minimum weights

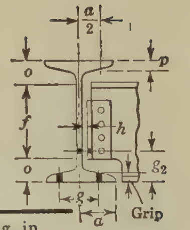
| Span, ft. | CB 143, 14″×10″ | | CB 142, 14″×8″ | | CB 141, 14″×6 3/4″ | | CB 124, 12″×12″ | | CB 123, 12″×10″ | | CB 122, 12″×8″ | | CB 121, 12″×6 1/2″ | | CB 103, 10″×10″ | | CB 102, 10″×8″ | | CB 101, 10″×5 3/4″ | | CB 83, 8″×8″ | | CB 82, 8″×6 1/2″ | | CB 81, 8″×5 1/4″ | | Coefficient of Deflection |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 74 lb. | 61 lb. | 58 lb. | 43 lb. | 42 lb. | 30 lb. | 85 lb. | 65 lb. | 64 lb. | 53 lb. | 50 lb. | 40 lb. | 36 lb. | 25 lb. | 66 lb. | 49 lb. | 45 lb. | 33 lb. | 29 lb. | 21 lb. | 35 lb. | 31 lb. | 27 lb. | 24 lb. | 21 lb. | 17 lb. | |
| ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 44.2 | ...... |
| 4 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 57.0 | ...... | ...... | ...... | ...... | 49.5 | 42.3 | 0.30 |
| 5 | ...... | ...... | ...... | ...... | ...... | 89.8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 68.4 | ...... | ...... | ...... | ...... | 70.9 | 51.6 | ...... | 55.3 | 52.6 | 46.6 | 43.2 | 33.8 | .47 |
| 6 | ...... | ...... | ...... | ...... | 115.5 | 83.6 | ...... | ...... | ...... | ...... | ...... | ...... | 89.6 | 61.8 | ...... | ...... | 85.0 | 68.3 | 61.6 | 43.0 | 61.4 | 54.8 | 46.8 | 41.6 | 36.0 | 28.2 | .67 |
| 7 | ...... | ...... | 137.0 | 101.1 | 104.1 | 71.7 | ...... | ...... | ...... | ...... | 108.5 | 84.2 | 78.7 | 53.0 | 113.8 | ...... | 84.2 | 60.0 | 52.8 | 36.9 | 53.3 | 47.0 | 40.1 | 35.7 | 30.9 | 24.2 | .91 |
| 8 | 153.3 | 126.2 | 127.5 | 94.1 | 91.1 | 62.7 | ...... | ...... | 119.7 | 99.9 | 97.1 | 77.9 | 68.9 | 46.4 | 110.6 | 81.6 | 73.7 | 52.5 | 46.2 | 32.3 | 46.7 | 41.1 | 35.1 | 31.2 | 27.0 | 21.2 | 1.19 |
| 9 | 149.7 | 122.9 | 113.3 | 83.6 | 80.9 | 55.7 | 148.5 | 113.4 | 114.4 | 94.3 | 86.3 | 69.2 | 61.2 | 41.2 | 98.3 | 72.8 | 65.5 | 46.7 | 41.1 | 28.7 | 41.5 | 36.5 | 31.2 | 27.7 | 24.0 | 18.8 | 1.51 |
| 10 | 134.8 | 110.6 | 102.0 | 75.2 | 72.8 | 50.2 | 138.8 | 105.6 | 103.0 | 84.8 | 77.6 | 62.3 | 55.1 | 37.1 | 88.4 | 65.5 | 58.9 | 42.0 | 37.0 | 25.8 | 37.3 | 32.9 | 28.1 | 25.0 | 21.6 | 16.9 | 1.86 |
| 11 | 122.5 | 100.6 | 92.7 | 68.4 | 66.2 | 45.6 | 126.2 | 96.0 | 93.6 | 77.1 | 70.6 | 56.6 | 50.1 | 33.7 | 80.4 | 59.6 | 53.6 | 38.2 | 33.6 | 23.5 | 33.9 | 29.9 | 25.5 | 22.7 | 19.6 | 15.4 | 2.25 |
| 12 | 112.3 | 92.2 | 85.0 | 62.7 | 60.7 | 41.8 | 115.7 | 88.0 | 85.8 | 70.7 | 64.7 | 51.9 | 45.9 | 30.9 | 73.7 | 54.6 | 49.1 | 35.0 | 30.8 | 21.5 | 31.1 | 27.4 | 23.4 | 20.8 | 18.0 | 14.1 | 2.68 |
| 13 | 103.7 | 85.1 | 78.5 | 57.9 | 56.0 | 38.6 | 106.8 | 81.2 | 79.2 | 65.3 | 59.7 | 47.9 | 42.4 | 28.5 | 68.0 | 50.4 | 45.3 | 32.3 | 28.4 | 19.8 | 28.7 | 25.3 | 21.6 | 19.2 | 16.6 | 13.0 | 3.15 |
| 14 | 96.3 | 79.0 | 72.9 | 53.7 | 52.0 | 35.8 | 99.2 | 75.4 | 73.5 | 60.6 | 55.5 | 44.5 | 39.3 | 26.5 | 63.2 | 46.8 | 42.1 | 30.0 | 26.4 | 18.4 | 26.7 | 23.5 | 20.1 | 17.8 | 15.4 | 12.1 | 3.65 |
| 15 | 89.8 | 73.8 | 68.0 | 50.2 | 48.6 | 33.4 | 92.6 | 70.4 | 68.6 | 56.6 | 51.8 | 41.5 | 36.7 | 24.7 | 59.0 | 43.7 | 39.3 | 28.0 | 24.6 | 17.2 | 24.9 | 21.9 | 18.7 | 16.6 | 14.4 | 11.3 | 4.19 |
| 16 | 84.2 | 69.2 | 63.8 | 47.0 | 45.5 | 31.4 | 86.8 | 66.0 | 64.4 | 53.0 | 48.5 | 38.9 | 34.4 | 23.2 | 55.3 | 41.0 | 36.8 | 26.3 | 23.1 | 16.1 | 23.3 | 20.6 | 17.6 | 15.6 | 13.5 | 10.6 | 4.77 |
| 17 | 79.3 | 65.1 | 60.0 | 44.3 | 42.8 | 29.5 | 81.7 | 62.1 | 60.6 | 49.9 | 45.7 | 36.6 | 32.4 | 21.8 | 52.0 | 38.5 | 34.7 | 24.7 | 21.7 | 15.2 | 22.0 | 19.3 | 16.5 | 14.7 | 12.7 | 10.0 | 5.38 |
| 18 | 74.9 | 61.5 | 56.7 | 41.8 | 40.5 | 27.9 | 77.1 | 58.7 | 57.2 | 47.1 | 43.1 | 34.6 | 30.6 | 20.6 | 49.1 | 36.4 | 32.7 | 23.3 | 20.5 | 14.3 | ...... | ...... | ...... | ...... | ...... | ...... | 6.03 |
| 19 | 70.9 | 58.2 | 53.7 | 39.6 | 38.3 | 26.4 | 73.1 | 55.6 | 54.2 | 44.7 | 40.9 | 32.8 | 29.0 | 19.5 | 46.5 | 34.5 | 31.0 | 22.1 | 19.5 | 13.6 | ...... | ...... | ...... | ...... | ...... | ...... | 6.72 |
| 20 | 67.4 | 55.3 | 51.0 | 37.6 | 36.4 | 25.1 | 69.4 | 52.8 | 51.5 | 42.4 | 38.8 | 31.1 | 27.5 | 18.5 | 44.2 | 32.8 | 29.5 | 21.0 | 18.5 | 12.9 | ...... | ...... | ...... | ...... | ...... | ...... | 7.45 |
| 21 | 64.2 | 52.7 | 48.6 | 35.8 | 34.7 | 23.9 | 66.1 | 50.3 | 49.0 | 40.4 | 37.0 | 29.7 | 26.2 | 17.7 | 42.1 | ...... | 28.1 | ...... | 17.6 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 8.21 |
| 22 | 61.3 | 50.3 | 46.4 | 34.2 | 33.1 | 22.8 | 63.1 | 48.0 | 46.8 | 38.6 | 35.3 | 28.3 | 25.0 | 16.9 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 9.01 |
| 23 | 58.6 | 48.1 | 44.3 | 32.7 | 31.7 | 21.8 | 60.4 | 45.9 | 44.8 | 36.9 | 33.8 | 27.1 | 23.9 | 16.1 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 9.85 |
| 24 | 56.2 | 46.1 | 42.5 | 31.4 | 30.4 | 20.9 | 57.9 | 44.0 | 42.9 | 35.4 | 32.4 | 26.0 | 23.0 | 15.5 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 10.73 |
| 25 | 53.9 | 44.3 | 40.8 | 30.1 | 29.1 | 20.1 | 55.5 | ...... | 41.2 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 11.64 |
| 26 | 51.8 | 42.6 | 39.2 | 28.9 | 28.0 | 19.3 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 12.59 |
| 27 | 49.9 | 41.0 | 37.8 | 27.9 | 27.0 | 18.6 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 13.57 |
| 28 | 48.1 | ...... | 36.4 | ...... | 26.0 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 14.60 |

* Tabular values are for sections braced against lateral deflection. For unbraced sections, loads must be reduced. See p. 17–48.
Maximum bending stress, 18,000 lb. per sq. in.
Loads above upper horizontal lines will produce maximum allowable shear in webs.
Loads below lower horizontal lines will produce excessive deflections.
1 Kip = 1000 lb.



## Table 5.—Properties of American Standard Steel Beams

(Carnegie-Illinois Steel Corporation, Pittsburgh)



| Depth of Beam, in. | Weight per Foot, lb. | Area of Section, sq. in. | Flange | | Web Thickness, in. | Elements of Sections | | | | | | Dimensions for Detailing, in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Axis A–A | | | Axis B–B | | | | | | | | | | |
| | | | Wide, in. | Thick, p, in. | | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | Moment of Inertia, $I'$, in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r$, in. | a | f | o | * $g_2$, min. | Clearance, h | Gage, g | Grip | Max. Flange Rivet. |
| Section No. B18—24 × 7 7/8 in. | | | | | | | | | | | | | | | | | | | |
| 24 | 120.0 | 35.13 | 8.048 | 1 1/8 | 0.798 | 3010.8 | 250.9 | 9.26 | 84.9 | 21.1 | 1.56 | 3 5/8 | 20 1/8 | 1 15/16 | 3 1/4 | 1/2 | 4 | 1 1/8 | 1 |
| | 115.0 | 33.67 | 7.987 | 1 1/8 | .737 | 2940.5 | 245.0 | 9.35 | 82.8 | 20.7 | 1.57 | 3 5/8 | 20 1/8 | 1 15/16 | 3 1/4 | 7/16 | 4 | 1 1/8 | 1 |
| | 110.0 | 32.18 | 7.925 | 1 1/8 | .675 | 2869.1 | 239.1 | 9.44 | 80.6 | 20.3 | 1.58 | 3 5/8 | 20 1/8 | 1 15/16 | 3 1/4 | 7/16 | 4 | 1 1/8 | 1 |
| | 105.9 | 30.98 | 7.875 | 1 1/8 | .625 | 2811.5 | 234.3 | 9.53 | 78.9 | 20.0 | 1.60 | 3 5/8 | 20 1/8 | 1 15/16 | 3 1/4 | 3/8 | 4 | 1 1/8 | 1 |
| Section No. B1—24 × 7 in. | | | | | | | | | | | | | | | | | | | |
| 24 | 100.0 | 29.25 | 7.247 | 7/8 | 0.747 | 2371.8 | 197.6 | 9.05 | 48.4 | 13.4 | 1.29 | 3 1/4 | 20 3/4 | 1 5/8 | 3 | 7/16 | 4 | 7/8 | 1 |
| | 95.0 | 27.79 | 7.186 | 7/8 | .686 | 2301.5 | 191.8 | 9.08 | 47.0 | 13.0 | 1.30 | 3 1/4 | 20 3/4 | 1 5/8 | 3 | 7/16 | 4 | 7/8 | 1 |
| | 90.0 | 26.30 | 7.124 | 7/8 | .624 | 2230.1 | 185.8 | 9.21 | 45.5 | 12.8 | 1.32 | 3 1/4 | 20 3/4 | 1 5/8 | 3 | 3/8 | 4 | 7/8 | 1 |
| | 85.0 | 24.84 | 7.063 | 7/8 | .563 | 2159.8 | 180.0 | 9.33 | 44.2 | 12.5 | 1.33 | 3 1/4 | 20 3/4 | 1 5/8 | 3 | 3/8 | 4 | 7/8 | 1 |
| | 79.9 | 23.33 | 7.000 | 7/8 | .500 | 2087.2 | 173.9 | 9.46 | 42.9 | 12.2 | 1.36 | 3 1/4 | 20 3/4 | 1 5/8 | 3 | 5/16 | 4 | 7/8 | 1 |
| Section No. B2—20 × 7 in. | | | | | | | | | | | | | | | | | | | |
| 20 | 100.0 | 29.20 | 7.273 | 15/16 | 0.873 | 1648.3 | 164.8 | 7.51 | 52.4 | 14.4 | 1.34 | 3 1/4 | 16 1/2 | 1 3/4 | 3 1/4 | 1/2 | 4 | 15/16 | 1 |
| | 95.0 | 27.74 | 7.200 | 15/16 | .800 | 1599.7 | 160.0 | 7.59 | 50.5 | 14.0 | 1.35 | 3 1/4 | 16 1/2 | 1 3/4 | 3 1/4 | 1/2 | 4 | 15/16 | 1 |
| | 90.0 | 26.26 | 7.126 | 15/16 | .726 | 1550.3 | 155.0 | 7.68 | 48.7 | 13.7 | 1.36 | 3 1/4 | 16 1/2 | 1 3/4 | 3 1/4 | 7/16 | 4 | 15/16 | 1 |
| | 85.0 | 24.80 | 7.053 | 15/16 | .653 | 1501.7 | 150.2 | 7.78 | 47.0 | 13.3 | 1.38 | 3 1/4 | 16 1/2 | 1 3/4 | 3 1/4 | 3/8 | 4 | 7/8 | 1 |
| | 81.4 | 23.74 | 7.000 | 15/16 | .600 | 1466.3 | 146.6 | 7.86 | 45.8 | 13.1 | 1.39 | 3 1/4 | 16 1/2 | 1 3/4 | 3 1/4 | 3/8 | 4 | 7/8 | 1 |
| Section No. B3—20 × 6 1/4 in. | | | | | | | | | | | | | | | | | | | |
| 20 | 75.0 | 21.90 | 6.391 | 13/16 | 0.641 | 1263.5 | 126.3 | 7.60 | 30.1 | 9.4 | 1.17 | 2 7/8 | 16 7/8 | 1 9/16 | 3 | 3/8 | 3 1/2 | 13/16 | 7/8 |
| | 70.0 | 20.42 | 6.317 | 13/16 | .567 | 1214.2 | 121.4 | 7.71 | 28.9 | 9.2 | 1.19 | 2 7/8 | 16 7/8 | 1 9/16 | 3 | 3/8 | 3 1/2 | 13/16 | 7/8 |
| | 65.4 | 19.08 | 6.250 | 13/16 | .500 | 1169.5 | 116.9 | 7.83 | 27.9 | 8.9 | 1.21 | 2 7/8 | 16 7/8 | 1 9/16 | 3 | 5/16 | 3 1/2 | 3/4 | 7/8 |
| Section No. B4—18 × 6 in. | | | | | | | | | | | | | | | | | | | |
| 18 | 70.0 | 20.46 | 6.251 | 11/16 | 0.711 | 917.5 | 101.9 | 6.70 | 24.5 | 7.8 | 1.09 | 2 3/4 | 15 1/4 | 1 3/8 | 2 3/4 | 7/16 | 3 1/2 | 11/16 | 7/8 |
| | 65.0 | 18.98 | 6.169 | 11/16 | .629 | 877.7 | 97.5 | 6.80 | 23.4 | 7.6 | 1.11 | 2 3/4 | 15 1/4 | 1 3/8 | 2 3/4 | 7/16 | 3 1/2 | 11/16 | 7/8 |
| | 60.0 | 17.50 | 6.087 | 11/16 | .547 | 837.8 | 93.1 | 6.92 | 22.3 | 7.3 | 1.13 | 2 3/4 | 15 1/4 | 1 3/8 | 2 3/4 | 3/8 | 3 1/2 | 11/16 | 7/8 |
| | 54.7 | 15.94 | 6.000 | 11/16 | .460 | 795.5 | 88.4 | 7.07 | 21.2 | 7.1 | 1.15 | 2 3/4 | 15 1/4 | 1 3/8 | 2 3/4 | 5/16 | 3 1/2 | 11/16 | 7/8 |
| Section No. B6—15 × 6 in. | | | | | | | | | | | | | | | | | | | |
| 15 | 75.0 | 21.85 | 6.278 | 13/16 | 0.868 | 687.2 | 91.6 | 5.61 | 30.6 | 9.8 | 1.18 | 2 3/4 | 11 3/4 | 1 5/8 | 3 | 1/2 | 3 1/2 | 13/16 | 7/8 |
| | 70.0 | 20.38 | 6.180 | 13/16 | .770 | 659.6 | 87.9 | 5.69 | 28.8 | 9.3 | 1.19 | 2 3/4 | 11 3/4 | 1 5/8 | 3 | 7/16 | 3 1/2 | 13/16 | 7/8 |
| | 65.0 | 18.91 | 6.082 | 13/16 | .672 | 632.1 | 84.3 | 5.78 | 27.2 | 8.9 | 1.20 | 2 3/4 | 11 3/4 | 1 5/8 | 3 | 7/16 | 3 1/2 | 13/16 | 7/8 |
| | 60.8 | 17.68 | 6.000 | 13/16 | .590 | 609.0 | 81.2 | 5.87 | 26.0 | 8.7 | 1.21 | 2 3/4 | 11 3/4 | 1 5/8 | 3 | 3/8 | 3 1/2 | 13/16 | 7/8 |

| | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Section No. B7—15 × 5 1/2 in.** | | | | | | | | | | | | | | | | | | | |
| 15 | 55.0 | 16.06 | 5.738 | 5/8 | 0.648 | 508.7 | 67.8 | 5.63 | 17.0 | 5.9 | 1.03 | 2 1/2 | 12 1/2 | 1 1/4 | 2 3/4 | 3/8 | 3 1/2 | 5/8 | 3/4 |
| | 50.0 | 14.59 | 5.640 | 5/8 | .550 | 481.1 | 64.2 | 5.74 | 16.0 | 5.7 | 1.05 | 2 1/2 | 12 1/2 | 1 1/4 | 2 3/4 | 3/8 | 3 1/2 | 9/16 | 3/4 |
| | 45.0 | 13.12 | 5.542 | 5/8 | .452 | 453.6 | 60.5 | 5.88 | 15.0 | 5.4 | 1.07 | 2 1/2 | 12 1/2 | 1 1/4 | 2 3/4 | 5/16 | 3 1/2 | 9/16 | 3/4 |
| | 42.9 | 12.49 | 5.500 | 5/8 | .410 | 441.8 | 58.9 | 5.95 | 14.6 | 5.3 | 1.08 | 2 1/2 | 12 1/2 | 1 1/4 | 2 3/4 | 5/16 | 3 1/2 | 9/16 | 3/4 |
| **Section No. B8—12 × 5 1/4 in.** | | | | | | | | | | | | | | | | | | | |
| 12 | 55.0 | 16.04 | 5.600 | 11/16 | 0.810 | 319.3 | 53.2 | 4.46 | 17.3 | 6.2 | 1.04 | 2 3/8 | 9 3/8 | 1 5/16 | 2 3/4 | 1/2 | 3 | 5/8 | 3/4 |
| | 50.0 | 14.57 | 5.477 | 11/16 | .687 | 301.6 | 50.3 | 4.55 | 16.0 | 5.8 | 1.05 | 2 3/8 | 9 3/8 | 1 5/16 | 2 3/4 | 7/16 | 3 | 5/8 | 3/4 |
| | 45.0 | 13.10 | 5.355 | 11/16 | .565 | 284.1 | 47.3 | 4.66 | 14.8 | 5.5 | 1.06 | 2 3/8 | 9 3/8 | 1 5/16 | 2 3/4 | 3/8 | 3 | 5/8 | 3/4 |
| | 40.8 | 11.84 | 5.250 | 11/16 | .460 | 268.9 | 44.8 | 4.77 | 13.8 | 5.3 | 1.08 | 2 3/8 | 9 3/8 | 1 5/16 | 2 3/4 | 5/16 | 3 | 5/8 | 3/4 |
| **Section No. B9—12 × 5 in.** | | | | | | | | | | | | | | | | | | | |
| 12 | 35.0 | 10.20 | 5.078 | 9/16 | 0.428 | 227.0 | 37.8 | 4.72 | 10.0 | 3.9 | 0.99 | 2 3/8 | 9 3/4 | 1 1/8 | 2 1/2 | 5/16 | 3 | 1/2 | 3/4 |
| | 31.8 | 9.26 | 5.000 | 9/16 | .350 | 215.8 | 36.0 | 4.83 | 9.5 | 3.8 | 1.01 | 2 3/8 | 9 3/4 | 1 1/8 | 2 1/2 | 1/4 | 3 | 1/2 | 3/4 |
| **Section No. B10—10 × 4 5/8 in.** | | | | | | | | | | | | | | | | | | | |
| 10 | 40.0 | 11.69 | 5.091 | 1/2 | 0.741 | 158.0 | 31.6 | 3.68 | 9.4 | 3.7 | 0.90 | 2 1/8 | 8 | 1 | 2 1/2 | 7/16 | 2 3/4 | 1/2 | 3/4 |
| | 35.0 | 10.22 | 4.944 | 1/2 | .594 | 145.8 | 29.2 | 3.78 | 8.5 | 3.4 | 0.91 | 2 1/8 | 8 | 1 | 2 1/2 | 3/8 | 2 3/4 | 1/2 | 3/4 |
| | 30.0 | 8.75 | 4.797 | 1/2 | .447 | 133.5 | 26.7 | 3.91 | 7.6 | 3.2 | 0.93 | 2 1/8 | 8 | 1 | 2 1/2 | 5/16 | 2 3/4 | 1/2 | 3/4 |
| | 25.4 | 7.38 | 4.660 | 1/2 | .310 | 122.1 | 24.4 | 4.07 | 6.9 | 3.0 | 0.97 | 2 1/8 | 8 | 1 | 2 1/2 | 1/4 | 2 3/4 | 1/2 | 3/4 |
| **Section No. B12—8 × 4 in.** | | | | | | | | | | | | | | | | | | | |
| 8 | 25.5 | 7.43 | 4.262 | 7/16 | 0.532 | 68.1 | 17.0 | 3.03 | 4.7 | 2.2 | 0.80 | 1 7/8 | 6 1/4 | 7/8 | 2 1/4 | 5/16 | 2 1/4 | 7/16 | 3/4 |
| | 23.0 | 6.71 | 4.171 | 7/16 | .441 | 64.2 | 16.0 | 3.09 | 4.4 | 2.1 | 0.81 | 1 7/8 | 6 1/4 | 7/8 | 2 1/4 | 5/16 | 2 1/4 | 7/16 | 3/4 |
| | 20.5 | 5.97 | 4.079 | 7/16 | .349 | 60.2 | 15.1 | 3.18 | 4.0 | 2.0 | 0.82 | 1 7/8 | 6 1/4 | 7/8 | 2 1/4 | 1/4 | 2 1/4 | 7/16 | 3/4 |
| | 18.4 | 5.34 | 4.000 | 7/16 | .270 | 56.9 | 14.2 | 3.26 | 3.8 | 1.9 | 0.84 | 1 7/8 | 6 1/4 | 7/8 | 2 1/4 | 3/16 | 2 1/4 | 7/16 | 3/4 |
| **Section No. B13—7 × 3 5/8 in.** | | | | | | | | | | | | | | | | | | | |
| 7 | 20.0 | 5.83 | 3.860 | 3/8 | 0.450 | 41.9 | 12.0 | 2.68 | 3.1 | 1.6 | 0.74 | 1 3/4 | 5 3/8 | 13/16 | 2 | 5/16 | 2 1/4 | 3/8 | 5/8 |
| | 17.5 | 5.09 | 3.755 | 3/8 | .345 | 38.9 | 11.1 | 2.77 | 2.9 | 1.6 | 0.76 | 1 3/4 | 5 3/8 | 13/16 | 2 | 1/4 | 2 1/4 | 3/8 | 5/8 |
| | 15.3 | 4.43 | 3.660 | 3/8 | .250 | 36.2 | 10.4 | 2.86 | 2.7 | 1.5 | 0.78 | 1 3/4 | 5 3/8 | 13/16 | 2 | 3/16 | 2 1/4 | 3/8 | 5/8 |
| **Section No. B14—6 × 3 3/8 in.** | | | | | | | | | | | | | | | | | | | |
| 6 | 17.25 | 5.02 | 3.565 | 3/8 | 0.465 | 26.0 | 8.7 | 2.28 | 2.3 | 1.3 | 0.68 | 1 1/2 | 4 1/2 | 3/4 | 2 | 5/16 | 2 | 3/8 | 5/8 |
| | 14.75 | 4.29 | 3.443 | 3/8 | .343 | 23.8 | 7.9 | 2.36 | 2.1 | 1.2 | 0.69 | 1 1/2 | 4 1/2 | 3/4 | 2 | 1/4 | 2 | 3/8 | 5/8 |
| | 12.5 | 3.61 | 3.330 | 3/8 | .230 | 21.8 | 7.3 | 2.46 | 1.8 | 1.1 | 0.72 | 1 1/2 | 4 1/2 | 3/4 | 2 | 3/16 | 2 | 5/16 | 5/8 |
| **Section No. B15—5 × 3 in.** | | | | | | | | | | | | | | | | | | | |
| 5 | 14.75 | 4.29 | 3.284 | 5/16 | 0.494 | 15.0 | 6.0 | 1.87 | 1.7 | 1.0 | 0.63 | 1 3/8 | 3 5/8 | 11/16 | 2 | 5/16 | 1 3/4 | 5/16 | 1/2 |
| | 12.25 | 3.56 | 3.137 | 5/16 | .347 | 13.5 | 5.4 | 1.95 | 1.4 | 0.91 | .63 | 1 3/8 | 3 5/8 | 11/16 | 2 | 1/4 | 1 3/4 | 5/16 | 1/2 |
| | 10.0 | 2.87 | 3.000 | 5/16 | .210 | 12.1 | 4.8 | 2.05 | 1.2 | .82 | .65 | 1 3/8 | 3 5/8 | 11/16 | 2 | 3/16 | 1 3/4 | 5/16 | 1/2 |
| **Section No. B16—4 × 2 5/8 in.** | | | | | | | | | | | | | | | | | | | |
| 4 | 10.5 | 3.05 | 2.870 | 5/16 | 0.400 | 7.1 | 3.5 | 1.52 | 1.0 | 0.70 | 0.57 | 1 1/4 | 2 3/4 | 5/8 | 2 | 1/4 | 1 1/2 | 5/16 | 1/2 |
| | 9.5 | 2.76 | 2.796 | 5/16 | .326 | 6.7 | 3.3 | 1.56 | 0.91 | .65 | .58 | 1 1/4 | 2 3/4 | 5/8 | 2 | 1/4 | 1 1/2 | 5/16 | 1/2 |
| | 8.5 | 2.46 | 2.723 | 5/16 | .253 | 6.3 | 3.2 | 1.60 | .83 | .61 | .58 | 1 1/4 | 2 3/4 | 5/8 | 2 | 3/16 | 1 1/2 | 5/16 | 1/2 |
| | 7.7 | 2.21 | 2.660 | 5/16 | .190 | 6.0 | 3.0 | 1.64 | .77 | .58 | .59 | 1 1/4 | 2 3/4 | 5/8 | 2 | 3/16 | 1 1/2 | 5/16 | 1/2 |
| **Section No. B17—3 × 2 3/8 in.** | | | | | | | | | | | | | | | | | | | |
| 3 | 7.5 | 2.17 | 2.509 | 1/4 | 0.349 | 2.9 | 1.9 | 1.15 | 0.59 | 0.47 | 0.52 | 1 1/8 | 1 7/8 | 9/16 | .... | 1/4 | 1 1/2 | 1/4 | 3/8 |
| | 6.5 | 1.88 | 2.411 | 1/4 | .251 | 2.7 | 1.8 | 1.19 | .51 | .43 | .52 | 1 1/8 | 1 7/8 | 9/16 | .... | 3/16 | 1 1/2 | 1/4 | 3/8 |
| | 5.7 | 1.64 | 2.330 | 1/4 | .170 | 2.5 | 1.7 | 1.23 | .46 | .40 | .53 | 1 1/8 | 1 7/8 | 9/16 | .... | 3/16 | 1 1/2 | 1/4 | 3/8 |

* Gages $g_2$ are based on 1 1/4 in. edge distance (7/8-in. maximum rivet.)

### Table 6.—Safe Uniform Loads on American Standard Beams

(Carnegie-Illinois Steel Corporation, Pittsburgh)

Applicable only when sections are braced against lateral deflection. For unbraced sections, see p. 17–48. Maximum and minimum weights only of each section are given. For intermediate weights, use the formulas on p. 7–22.

| Span, ft. | Section No., Nominal Depth and Flange Width and Weight per Foot | | | | | | | | | | | | | | Coefficient of Deflection |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | B18, 24 × 7 7/8 in. | | B1, 24 × 7 in. | | B2, 20 × 7 in. | | B3, 20 × 6 1/4 in. | | B4, 18 × 6 in. | | B6, 15 × 6 in. | | B7, 15 × 5 1/2 in. | | |
| | 120 lb. | 105.9 lb. | 100 lb. | 79.9 lb. | 100 lb. | 81.4 lb. | 75 lb. | 65.4 lb. | 70 lb. | 54.7 lb. | 75 lb. | 60.8 lb. | 55 lb. | 42.9 lb. | |
| | Safe Load, Kips* | | | | | | | | | | | | | | |
| ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 307.2 | ..... | 312.5 | ..... | 233.3 | ..... | ..... |
| 4 | ..... | ..... | ..... | ..... | 419.0 | ..... | 307.7 | ..... | 305.8 | ..... | 274.9 | 212.4 | 203.5 | 147.6 | 0.30 |
| 5 | ..... | ..... | 430.3 | ..... | 395.6 | ..... | 303.2 | 240.0 | 244.7 | 198.7 | 219.9 | 194.9 | 162.8 | 141.4 | .47 |
| 6 | 495.6 | ..... | 395.3 | ..... | 329.7 | 288.0 | 252.7 | 233.9 | 203.9 | 176.8 | 183.3 | 162.4 | 135.7 | 117.8 | .67 |
| 7 | 430.1 | 360.0 | 338.8 | 288.0 | 282.6 | 251.4 | 216.6 | 200.5 | 174.8 | 151.5 | 157.1 | 139.2 | 116.3 | 101.0 | .91 |
| 8 | 376.4 | 351.4 | 296.5 | 260.9 | 247.2 | 219.9 | 189.5 | 175.4 | 152.9 | 132.6 | 137.4 | 121.8 | 101.7 | 88.4 | 1.19 |
| 9 | 334.5 | 312.4 | 263.5 | 231.9 | 219.8 | 195.5 | 168.5 | 155.9 | 135.9 | 117.9 | 122.2 | 108.3 | 90.4 | 78.5 | 1.51 |
| 10 | 301.1 | 281.2 | 237.2 | 208.7 | 197.8 | 176.0 | 151.6 | 140.3 | 122.3 | 106.1 | 110.0 | 97.4 | 81.4 | 70.7 | 1.86 |
| 11 | 273.7 | 255.6 | 215.6 | 189.7 | 179.8 | 160.0 | 137.8 | 127.6 | 111.2 | 96.4 | 100.0 | 88.6 | 74.0 | 64.3 | 2.25 |
| 12 | 250.9 | 234.3 | 197.7 | 173.9 | 164.8 | 146.6 | 126.3 | 116.9 | 101.9 | 88.4 | 91.6 | 81.2 | 67.8 | 58.9 | 2.68 |
| 13 | 231.6 | 216.3 | 182.4 | 160.6 | 152.2 | 135.3 | 116.6 | 108.0 | 94.1 | 81.6 | 84.6 | 75.0 | 62.6 | 54.4 | 3.15 |
| 14 | 215.1 | 200.8 | 169.4 | 149.1 | 141.3 | 125.7 | 108.3 | 100.2 | 87.4 | 75.8 | 78.5 | 69.6 | 58.1 | 50.5 | 3.65 |
| 15 | 200.7 | 187.4 | 158.1 | 139.1 | 131.9 | 117.3 | 101.1 | 93.6 | 81.6 | 70.7 | 73.3 | 65.0 | 54.3 | 47.1 | 4.19 |
| 16 | 188.2 | 175.7 | 148.2 | 130.5 | 123.6 | 110.0 | 94.8 | 87.7 | 76.5 | 66.3 | 68.7 | 60.9 | 50.9 | 44.2 | 4.77 |
| 17 | 177.1 | 165.4 | 139.5 | 122.8 | 116.4 | 103.5 | 89.2 | 82.6 | 72.0 | 62.4 | 64.7 | 57.3 | 47.9 | 41.6 | 5.38 |
| 18 | 167.3 | 156.2 | 131.8 | 116.0 | 109.9 | 97.8 | 84.2 | 78.0 | 68.0 | 58.9 | 61.1 | 54.1 | 45.2 | 39.3 | 6.03 |
| 19 | 158.5 | 148.0 | 124.8 | 109.9 | 104.1 | 92.6 | 79.8 | 73.9 | 64.4 | 55.8 | 57.9 | 51.3 | 42.8 | 37.2 | 6.72 |
| 20 | 150.5 | 140.6 | 118.6 | 104.4 | 98.9 | 88.0 | 75.8 | 70.2 | 61.2 | 53.0 | 55.0 | 48.7 | 40.7 | 35.3 | 7.45 |
| 21 | 143.4 | 133.9 | 112.9 | 99.4 | 94.2 | 83.8 | 72.2 | 66.8 | 58.3 | 50.5 | 52.4 | 46.4 | 38.8 | 33.7 | 8.21 |
| 22 | 136.9 | 127.8 | 107.8 | 94.9 | 89.9 | 80.0 | 68.9 | 63.8 | 55.6 | 48.2 | 50.0 | 44.3 | 37.0 | 32.1 | 9.01 |
| 23 | 130.9 | 122.2 | 103.1 | 90.7 | 86.0 | 76.5 | 65.9 | 61.0 | 53.2 | 46.1 | 47.8 | 42.4 | 35.4 | 30.7 | 9.85 |
| 24 | 125.5 | 117.1 | 98.8 | 87.0 | 82.4 | 73.3 | 63.2 | 58.5 | 51.0 | 44.2 | 45.8 | 40.6 | 33.9 | 29.5 | 10.73 |
| 25 | 120.4 | 112.5 | 94.9 | 83.5 | 79.1 | 70.4 | 60.6 | 56.1 | 48.9 | 42.4 | 44.0 | 39.0 | 32.6 | 28.3 | 11.64 |
| 26 | 115.8 | 108.1 | 91.2 | 80.3 | 76.1 | 67.7 | 58.3 | 54.0 | 47.1 | 40.8 | 42.3 | 37.5 | 31.3 | 27.2 | 12.59 |
| 27 | 111.5 | 104.1 | 87.8 | 77.3 | 73.3 | 65.2 | 56.2 | 52.0 | 45.3 | 39.3 | 40.7 | 36.1 | 30.1 | 26.2 | 13.57 |
| 28 | 107.5 | 100.4 | 84.7 | 74.5 | 70.6 | 62.8 | 54.2 | 50.1 | 43.7 | 37.9 | 39.3 | 34.8 | 29.1 | 25.2 | 14.60 |
| 29 | 103.8 | 96.9 | 81.8 | 72.0 | 68.2 | 60.7 | 52.3 | 48.4 | 42.2 | 36.6 | 37.9 | 33.6 | 28.1 | 24.4 | 15.66 |
| 30 | 100.4 | 93.7 | 79.1 | 69.6 | 65.9 | 58.7 | 50.5 | 46.8 | 40.8 | 35.4 | ..... | ..... | ..... | ..... | 16.76 |
| 31 | 97.1 | 90.7 | 76.5 | 67.3 | 63.8 | 56.8 | 48.9 | 45.3 | 39.5 | 34.2 | ..... | ..... | ..... | ..... | 17.89 |
| 32 | 94.1 | 87.9 | 74.1 | 65.2 | 61.8 | 55.0 | 47.4 | 43.9 | 38.2 | 33.1 | ..... | ..... | ..... | ..... | 19.07 |
| 33 | 91.2 | 85.2 | 71.9 | 63.3 | 59.9 | 53.3 | 45.9 | 42.5 | 37.1 | 32.1 | ..... | ..... | ..... | ..... | 20.28 |
| 34 | 88.6 | 82.7 | 69.8 | 61.4 | 58.2 | 51.8 | 44.6 | 41.3 | 36.0 | 31.2 | ..... | ..... | ..... | ..... | 21.53 |
| 35 | 86.0 | 80.3 | 67.8 | 59.6 | 56.5 | 50.3 | 43.3 | 40.1 | 35.0 | 30.3 | ..... | ..... | ..... | ..... | 22.81 |
| 36 | 83.6 | 78.1 | 65.9 | 58.0 | 54.9 | 48.9 | 42.1 | 39.0 | ..... | ..... | ..... | ..... | ..... | ..... | 24.13 |
| 37 | 81.4 | 76.0 | 64.1 | 56.4 | 53.5 | 47.6 | 41.0 | 37.9 | ..... | ..... | ..... | ..... | ..... | ..... | 25.49 |
| 38 | 79.2 | 74.0 | 62.4 | 54.9 | 52.1 | 46.3 | 39.9 | 36.9 | ..... | ..... | ..... | ..... | ..... | ..... | 26.89 |
| 39 | 77.2 | 72.1 | 60.8 | 53.5 | ..... | ..... | ..... | | | | | | | | 28.32 |
| 40 | 75.3 | 70.3 | 59.3 | 52.2 | ..... | ..... | ..... | | | | | | | | 29.79 |
| 42 | 71.7 | 66.9 | 56.5 | 49.7 | ..... | ..... | ..... | | | | | | | | 32.85 |
| 44 | 68.4 | 63.9 | 53.9 | 47.4 | ..... | ..... | ..... | | | | | | | | 36.05 |
| 46 | 65.5 | 61.1 | 51.6 | 45.4 | ..... | ..... | ..... | | | | | | | | 39.40 |
| 48 | 62.7 | 58.6 | 49.4 | 43.5 | ..... | ..... | ..... | | | | | | | | 42.90 |

Loads above upper horizontal lines will produce maximum allowable shear in webs.

Loads below lower horizontal lines will produce excessive deflections.

* 1 Kip = 1000 lb.

(*Table continued on following page*)

### Table 6.—Safe Uniform Loads on American Standard Beams—*Continued*

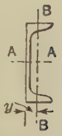
| Span, ft. | Section No., Nominal Depth and Flange Width and Weight per Foot | | | | | | | | | | | | | | | | | | Coefficient of Deflection |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | B8, 12 × 5 1/4 in. | | B9, 12 × 5 in. | | B10, 10 × 4 5/8 in. | | B12, 8 × 4 in. | | B13, 7 × 3 5/8 in. | | B14, 6 × 3 3/8 in. | | B15, 5 × 3 in. | | B16, 4 × 2 5/8 in. | | B17, 3 × 2 3/8 in. | | |
| | 55 lb. | 40.8 lb. | 35 lb. | 31.8 lb. | 40 lb. | 25.4 lb. | 25.5 lb. | 18.4 lb. | 20 lb. | 15.3 lb. | 17.25 lb. | 12.5 lb. | 14.75 lb. | 10 lb. | 10.5 lb. | 7.7 lb. | 7.5 lb. | 5.7 lb. | |
| ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .... | .... | .... | ..... | ..... | ..... | .... | .... | .... | 25.1 | .... | ..... |
| 1 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | .... | 75.6 | .... | 67.0 | ..... | 59.3 | .... | 38.4 | 18.2 | 23.1 | 12.2 | 0.02 |
| 2 | 233.3 | ..... | ..... | ..... | 177.8 | ..... | 102.1 | .... | 71.9 | 42.0 | 52.0 | 33.1 | 36.1 | 25.2 | 21.3 | 17.9 | 11.5 | 9.9 | .07 |
| 3 | 212.9 | ..... | 123.3 | ..... | 126.4 | 74.4 | 68.1 | 51.8 | 47.9 | 41.4 | 34.7 | 29.0 | 24.1 | 19.3 | 14.2 | 11.9 | 7.7 | 6.6 | .17 |
| 4 | 159.7 | 132.5 | 113.5 | 100.8 | 94.8 | 73.3 | 51.0 | 42.7 | 36.0 | 31.1 | 26.0 | 21.8 | 18.1 | 14.5 | 10.6 | 8.9 | 5.8 | 5.0 | .30 |
| 5 | 127.7 | 107.6 | 90.8 | 86.3 | 75.8 | 58.6 | 40.8 | 34.1 | 28.8 | 24.8 | 20.8 | 17.4 | 14.4 | 11.6 | 8.5 | 7.2 | 4.6 | 4.0 | .47 |
| 6 | 106.5 | 89.7 | 75.7 | 71.9 | 63.2 | 48.8 | 34.0 | 28.4 | 24.0 | 20.7 | 17.3 | 14.5 | 12.0 | 9.7 | 7.1 | 6.0 | 3.9 | 3.3 | .67 |
| 7 | 91.3 | 76.8 | 64.9 | 61.7 | 54.2 | 41.9 | 29.2 | 24.4 | 20.5 | 17.7 | 14.9 | 12.4 | 10.3 | 8.3 | 6.1 | 5.1 | 3.3 | 2.8 | .91 |
| 8 | 79.8 | 67.2 | 56.8 | 54.0 | 47.4 | 36.6 | 25.5 | 21.3 | 18.0 | 15.5 | 13.0 | 10.9 | 9.0 | 7.3 | 5.3 | 4.5 | 2.9 | 2.5 | 1.19 |
| 9 | 71.0 | 59.8 | 50.5 | 48.0 | 42.1 | 32.6 | 22.7 | 19.0 | 16.0 | 13.8 | 11.6 | 9.7 | 8.0 | 6.4 | 4.7 | 4.0 | .... | .... | 1.51 |
| 10 | 63.9 | 53.8 | 45.4 | 43.2 | 37.9 | 29.3 | 20.4 | 17.1 | 14.4 | 12.4 | 10.4 | 8.7 | 7.2 | 5.8 | 4.3 | 3.6 | .... | .... | 1.86 |
| 11 | 58.1 | 48.9 | 41.3 | 39.2 | 34.5 | 26.6 | 18.6 | 15.5 | 13.1 | 11.3 | 9.5 | 7.9 | 6.6 | 5.3 | .... | .... | .... | .... | 2.25 |
| 12 | 53.2 | 44.8 | 37.8 | 36.0 | 31.6 | 24.4 | 17.0 | 14.2 | 12.0 | 10.4 | 8.7 | 7.3 | ..... | .... | .... | .... | .... | .... | 2.68 |
| 13 | 49.1 | 41.4 | 34.9 | 33.2 | 29.2 | 22.5 | 15.7 | 13.1 | 11.1 | 9.6 | 8.0 | 6.7 | ..... | .... | .... | .... | .... | .... | 3.15 |
| 14 | 45.6 | 38.4 | 32.4 | 30.8 | 27.1 | 20.9 | 14.6 | 12.2 | 10.3 | 8.9 | ..... | ..... | ..... | .... | .... | .... | .... | .... | 3.65 |
| 15 | 42.6 | 35.9 | 30.3 | 28.8 | 25.3 | 19.5 | 13.6 | 11.4 | 9.6 | 8.3 | ..... | ..... | ..... | .... | .... | .... | .... | .... | 4.19 |
| 16 | 39.9 | 33.6 | 28.4 | 27.0 | 23.7 | 18.3 | 12.8 | 10.7 | .... | .... | ..... | ..... | ..... | .... | .... | .... | .... | .... | 4.77 |
| 17 | 37.6 | 31.6 | 26.7 | 25.4 | 22.3 | 17.2 | 12.0 | 10.0 | .... | .... | ..... | ..... | ..... | .... | .... | .... | .... | .... | 5.38 |
| 18 | 35.5 | 29.9 | 25.2 | 24.0 | 21.1 | 16.3 | ..... | .... | .... | .... | ..... | ..... | ..... | .... | .... | .... | .... | .... | 6.03 |
| 19 | 33.6 | 28.3 | 23.9 | 22.7 | 20.0 | 15.4 | ..... | .... | .... | | | | | | | | | | 6.72 |
| 20 | 31.9 | 26.9 | 22.7 | 21.6 | 19.0 | 14.7 | ..... | .... | .... | | | | | | | | | | 7.45 |
| 21 | 30.4 | 25.6 | 21.6 | 20.6 | ..... | ..... | ..... | .... | .... | | | | | | | | | | 8.21 |
| 22 | 29.0 | 24.4 | 20.6 | 19.6 | ..... | ..... | ..... | .... | .... | | | | | | | | | | 9.01 |
| 23 | 27.8 | 23.4 | 19.7 | 18.8 | ..... | ..... | ..... | .... | .... | | | | | | | | | | 9.85 |
| 24 | 26.6 | 22.4 | 18.9 | 18.0 | ..... | ..... | ..... | .... | .... | | | | | | | | | | 10.73 |

Loads above upper horizontal lines will produce maximum allowable shear in webs.

Loads below lower horizontal lines will produce excessive deflections.

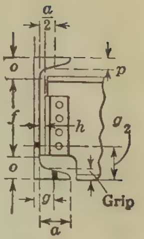
### Table 7.—Connections for Beams and Channels

| Depth of Beam or Channel, in. | Size of Angles, in. | Length of Angle, in. | Usual Construction See Fig. 1, p. 17–49 | | | Additional Riveting for Short Span, Thin Web or Concentrated Load |
|---|---|---|---|---|---|---|
| | | | No. of Rows of Rivets in Leg Riveted to Beam | No. of Rivets in Each Leg | Spacing of Holes, in. | Location of Rivets in Second Row See Fig. 2, p. 17–49 |
| 36 | 4 × 3 1/2 × 7/16 | 29 1/2 | 1 | 10 | 3 | *a, b, d, g, i, j* |
| 33 | 4 × 3 1/2 × 7/16 | 26 1/2 | 1 | 9 | 3 | *a, c, e, g, i* |
| 30 | 4 × 3 1/2 × 7/16 | 23 1/2 | 1 | 8 | 3 | *a, c, f, h* |
| 27 | 4 × 3 1/2 × 7/16 | 20 1/2 | 1 | 7 | 3 | *a, c, e, g* |
| 24 | 4 × 3 1/2 × 7/16 | 17 1/2 | 1 | 6 | 3 | *a, b, e, f* |
| 21 | 4 × 3 1/2 × 7/16 | 14 1/2 | 1 | 5 | 3 | *a, c, e* |
| 20 | 4 × 3 1/2 × 7/16 | 14 1/2 | 1 | 5 | 3 | *a, c, e* |
| 18 | 4 × 3 1/2 × 3/8 | 11 1/2 | 1 | 4 | 3 | *a, b, c, d* |
| 15 | 4 × 3 1/2 × 3/8 | 11 1/2 | 1 | 4 | 3 | *a, b, c, d* |
| 14 | 4 × 3 1/2 × 3/8 | 8 1/2 | 1 | 3 | 3 | *a, b, c* |
| 12 | 4 × 3 1/2 × 3/8 | 8 1/2 | 1 | 3 | 3 | *a, b, e* |
| 10 | 6 × 4 × 3/8 | 5 1/2 | 2 | 2 | 3 | ..... |
| 8 | 6 × 4 × 3/8 | 5 1/2 | 2 | 2 | 3 | ..... |
| 7 | 6 × 4 × 3/8 | 3 | 2 | 1 | ... | ..... |
| 5 | 6 × 4 × 3/8 | 3 | 2 | 1 | ... | ..... |



## Table 8.—Properties of American Standard Channels

(Carnegie-Illinois Steel Corporation, Pittsburgh)



| Depth of Channel, in. | Elements of Sections | | | | | | | | | | | | Dimensions for Detailing, in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Wt. per ft., lb. | Area of Section, sq. in. | Flange | | Web Thickness, in. | Axis A-A | | | Axis B-B | | | $y$ | $a$ | $f$ | $o$ | $g_2$, min. | $h$ | Gage $g$ | Grip | Max. Flange Rivet |
| | | | Width, in. | Thickness, $P$, in. | | Moment of Inertia, $I$, in.⁴ | Section Modulus, $S$, in.³ | Radius of Gyration, $r$, in. | Moment of Inertia, $I'$, in.⁴ | Section Modulus, $S'$, in.³ | Radius of Gyration, $r'$, in. | | | | | | | | | |
| **Section C1—15 × 3 3/8 in.** | | | | | | | | | | | | | | | | | | | | |
| 15 | 55.0 | 16.11 | 3.814 | 5/8 | 0.814 | 429.0 | 57.2 | 5.16 | 12.1 | 4.1 | 0.87 | 0.82 | 3 | 12 3/8 | 1 5/16 | 2 3/4 | 7/8 | 2 1/4 | 11/16 | 1 |
| | 50.0 | 14.64 | 3.716 | 5/8 | .716 | 401.4 | 53.6 | 5.24 | 11.2 | 3.8 | .87 | .80 | 3 | 12 3/8 | 1 5/16 | 2 3/4 | 13/16 | 2 1/4 | 5/8 | 1 |
| | 45.0 | 13.17 | 3.618 | 5/8 | .618 | 373.9 | 49.8 | 5.33 | 10.3 | 3.6 | .88 | .79 | 3 | 12 3/8 | 1 5/16 | 2 3/4 | 11/16 | 2 1/4 | 5/8 | 1 |
| | 40.0 | 11.70 | 3.520 | 5/8 | .520 | 346.3 | 46.2 | 5.44 | 9.3 | 3.4 | .89 | .78 | 3 | 12 3/8 | 1 5/16 | 2 3/4 | 5/8 | 2 | 5/8 | 1 |
| | 35.0 | 10.23 | 3.422 | 5/8 | .422 | 318.7 | 42.5 | 5.58 | 8.4 | 3.2 | .91 | .79 | 3 | 12 3/8 | 1 5/16 | 2 3/4 | 1/2 | 2 | 5/8 | 1 |
| | 33.9 | 9.90 | 3.400 | 5/8 | .400 | 312.6 | 41.7 | 5.62 | 8.2 | 3.2 | .91 | .79 | 3 | 12 3/8 | 1 5/16 | 2 3/4 | 1/2 | 2 | 5/8 | 1 |
| **Section C2—12 × 3 in.** | | | | | | | | | | | | | | | | | | | | |
| 12 | 40.0 | 11.73 | 3.415 | 1/2 | 0.755 | 196.5 | 32.8 | 4.09 | 6.6 | 2.5 | 0.75 | 0.72 | 2 5/8 | 9 7/8 | 1 1/16 | 2 1/2 | 13/16 | 2 | 1/2 | 7/8 |
| | 35.0 | 10.26 | 3.292 | 1/2 | .632 | 178.8 | 29.8 | 4.18 | 5.9 | 2.3 | .76 | .69 | 2 5/8 | 9 7/8 | 1 1/16 | 2 1/2 | 11/16 | 2 | 1/2 | 7/8 |
| | 30.0 | 8.79 | 3.170 | 1/2 | .510 | 161.2 | 26.9 | 4.28 | 5.2 | 2.1 | .77 | .68 | 2 5/8 | 9 7/8 | 1 1/16 | 2 1/2 | 9/16 | 1 3/4 | 1/2 | 7/8 |
| | 25.0 | 7.32 | 3.047 | 1/2 | .387 | 143.5 | 23.9 | 4.43 | 4.5 | 1.9 | .79 | .68 | 2 5/8 | 9 7/8 | 1 1/16 | 2 1/2 | 7/16 | 1 3/4 | 1/2 | 7/8 |
| | 20.7 | 6.03 | 2.940 | 1/2 | .280 | 128.1 | 21.4 | 4.61 | 3.9 | 1.7 | .81 | .70 | 2 5/8 | 9 7/8 | 1 1/16 | 2 1/2 | 3/8 | 1 3/4 | 1/2 | 7/8 |
| **Section C3—10 × 2 5/8 in.** | | | | | | | | | | | | | | | | | | | | |
| 10 | 35.0 | 10.27 | 3.180 | 7/16 | 0.820 | 115.2 | 23.0 | 3.34 | 4.6 | 1.9 | 0.67 | 0.69 | 2 3/8 | 8 1/8 | 15/16 | 2 1/2 | 7/8 | 1 3/4 | 1/2 | 3/4 |
| | 30.0 | 8.80 | 3.033 | 7/16 | .673 | 103.0 | 20.6 | 3.42 | 4.0 | 1.7 | .67 | .65 | 2 3/8 | 8 1/8 | 15/16 | 2 1/2 | 3/4 | 1 3/4 | 7/16 | 3/4 |
| | 25.0 | 7.33 | 2.886 | 7/16 | .526 | 90.7 | 18.1 | 3.52 | 3.4 | 1.5 | .68 | .62 | 2 3/8 | 8 1/8 | 15/16 | 2 1/2 | 5/8 | 1 3/4 | 7/16 | 3/4 |
| | 20.0 | 5.86 | 2.739 | 7/16 | .379 | 78.5 | 15.7 | 3.66 | 2.8 | 1.3 | .70 | .61 | 2 3/8 | 8 1/8 | 15/16 | 2 1/2 | 7/16 | 1 1/2 | 7/16 | 3/4 |
| | 15.3 | 4.47 | 2.600 | 7/16 | .240 | 66.9 | 13.4 | 3.87 | 2.3 | 1.2 | .72 | .64 | 2 3/8 | 8 1/8 | 15/16 | 2 1/2 | 5/16 | 1 1/2 | 7/16 | 3/4 |
| **Section C4—9 × 2 1/2 in.** | | | | | | | | | | | | | | | | | | | | |
| 9 | 25.0 | 7.33 | 2.812 | 7/16 | 0.612 | 70.5 | 15.7 | 3.10 | 3.0 | 1.4 | 0.64 | 0.61 | 2 1/4 | 7 1/4 | 7/8 | 2 1/2 | 11/16 | 1 1/2 | 7/16 | 3/4 |
| | 20.0 | 5.86 | 2.648 | 7/16 | .448 | 60.6 | 13.5 | 3.22 | 2.4 | 1.2 | .65 | .59 | 2 1/4 | 7 1/4 | 7/8 | 2 1/2 | 1/2 | 1 1/2 | 7/16 | 3/4 |
| | 15.0 | 4.39 | 2.485 | 7/16 | .285 | 50.7 | 11.3 | 3.40 | 1.9 | 1.0 | .67 | .59 | 2 1/4 | 7 1/4 | 7/8 | 2 1/2 | 3/8 | 1 3/8 | 7/16 | 3/4 |
| | 13.4 | 3.89 | 2.430 | 7/16 | .230 | 47.3 | 10.5 | 3.49 | 1.8 | 0.97 | .67 | .61 | 2 1/4 | 7 1/4 | 7/8 | 2 1/2 | 5/16 | 1 3/8 | 3/8 | 3/4 |

Section C5—8 × 2 1/4 in.

| | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 8 | 21.25 | 6.23 | 2.619 | 3/8 | 0.579 | 47.6 | 11.9 | 2.77 | 2.2 | 1.1 | 0.60 | 0.59 | 2 | 6 3/8 | 13/16 | 2 1/4 | 11/16 | 1 1/2 | 3/8 | 3/4 |
| | 18.75 | 5.49 | 2.527 | 3/8 | .487 | 43.7 | 10.9 | 2.82 | 2.0 | 1.0 | .60 | .57 | 2 | 6 3/8 | 13/16 | 2 1/4 | 9/16 | 1 1/2 | 3/8 | 3/4 |
| | 16.25 | 4.76 | 2.435 | 3/8 | .395 | 39.8 | 9.9 | 2.89 | 1.8 | 0.94 | .61 | .56 | 2 | 6 3/8 | 13/16 | 2 1/4 | 1/2 | 1 1/2 | 3/8 | 3/4 |
| | 13.75 | 4.02 | 2.343 | 3/8 | .303 | 35.8 | 9.0 | 2.99 | 1.5 | .86 | .62 | .56 | 2 | 6 3/8 | 13/16 | 2 1/4 | 3/8 | 1 3/8 | 3/8 | 3/4 |
| | 11.5 | 3.36 | 2.260 | 3/8 | .220 | 32.3 | 8.1 | 3.10 | 1.3 | .79 | .63 | .58 | 2 | 6 3/8 | 13/16 | 2 1/4 | 5/16 | 1 3/8 | 3/8 | 3/4 |

Section C6—7 × 2 in.

| | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7 | 19.75 | 5.79 | 2.509 | 3/8 | 0.629 | 33.1 | 9.4 | 2.39 | 1.8 | 0.96 | 0.56 | 0.58 | 1 7/8 | 5 3/8 | 13/16 | 2 | 11/16 | 1 1/2 | 3/8 | 5/8 |
| | 17.25 | 5.05 | 2.404 | 3/8 | .524 | 30.1 | 8.6 | 2.44 | 1.6 | .86 | .56 | .55 | 1 7/8 | 5 3/8 | 13/16 | 2 | 5/8 | 1 1/2 | 3/8 | 5/8 |
| | 14.75 | 4.32 | 2.299 | 3/8 | .419 | 27.1 | 7.7 | 2.51 | 1.4 | .79 | .57 | .53 | 1 7/8 | 5 3/8 | 13/16 | 2 | 1/2 | 1 1/4 | 3/8 | 5/8 |
| | 12.25 | 3.58 | 2.194 | 3/8 | .314 | 24.1 | 6.9 | 2.59 | 1.2 | .71 | .58 | .53 | 1 7/8 | 5 3/8 | 13/16 | 2 | 3/8 | 1 1/4 | 3/8 | 5/8 |
| | 9.8 | 2.85 | 2.090 | 3/8 | .210 | 21.1 | 6.0 | 2.72 | 0.98 | .63 | .59 | .55 | 1 7/8 | 5 3/8 | 13/16 | 2 | 5/16 | 1 1/4 | 3/8 | 5/8 |

Section C7—6 × 2 in.

| | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 15.5 | 4.54 | 2.279 | 3/8 | 0.559 | 19.5 | 6.5 | 2.07 | 1.3 | 0.73 | 0.53 | 0.55 | 1 3/4 | 4 1/2 | 3/4 | 2 | 5/8 | 1 3/8 | 3/8 | 5/8 |
| | 13.0 | 3.81 | 2.157 | 3/8 | .437 | 17.3 | 5.8 | 2.13 | 1.1 | .65 | .53 | .52 | 1 3/4 | 4 1/2 | 3/4 | 2 | 1/2 | 1 3/8 | 5/16 | 5/8 |
| | 10.5 | 3.07 | 2.034 | 3/8 | .314 | 15.1 | 5.0 | 2.22 | 0.87 | .57 | .53 | .50 | 1 3/4 | 4 1/2 | 3/4 | 2 | 3/8 | 1 1/8 | 3/8 | 5/8 |
| | 8.2 | 2.39 | 1.920 | 3/8 | .200 | 13.0 | 4.3 | 2.34 | .70 | .50 | .54 | .52 | 1 3/4 | 4 1/2 | 3/4 | 2 | 1/4 | 1 1/8 | 5/16 | 5/8 |

Section C8—5 × 1 3/4 in.

| | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 11.5 | 3.36 | 2.032 | 5/16 | 0.472 | 10.4 | 4.1 | 1.76 | 0.82 | 0.54 | 0.49 | 0.51 | 1 1/2 | 3 5/8 | 11/16 | 2 | 9/16 | 1 1/8 | 5/16 | 1/2 |
| | 9.0 | 2.63 | 1.885 | 5/16 | .325 | 8.8 | 3.5 | 1.83 | .64 | .45 | .49 | .48 | 1 1/2 | 3 5/8 | 11/16 | 2 | 3/8 | 1 1/8 | 5/16 | 1/2 |
| | 6.7 | 1.95 | 1.750 | 5/16 | .190 | 7.4 | 3.0 | 1.95 | .48 | .38 | .50 | .49 | 1 1/2 | 3 5/8 | 11/16 | 2 | 1/4 | 1 1/8 | 5/16 | 1/2 |

Section C9—4 × 1 5/8 in.

| | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 4 | 7.25 | 2.12 | 1.720 | 5/16 | 0.320 | 4.5 | 2.3 | 1.47 | 0.44 | 0.35 | 0.46 | 0.46 | 1 3/8 | 2 3/4 | 5/8 | 2 | 3/8 | 1 | 5/16 | 1/2 |
| | 6.25 | 1.82 | 1.647 | 5/16 | .247 | 4.1 | 2.1 | 1.50 | .38 | .32 | .45 | .46 | 1 3/8 | 2 3/4 | 5/8 | 2 | 5/16 | 1 | 5/16 | 1/2 |
| | 5.4 | 1.56 | 1.580 | 5/16 | .180 | 3.8 | 1.9 | 1.56 | .32 | .29 | .45 | .46 | 1 3/8 | 2 3/4 | 5/8 | 2 | 1/4 | 1 | 1/4 | 1/2 |

Section C10—3 × 1 1/2 in.

| | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3 | 6.0 | 1.75 | 1.596 | 1/4 | 0.356 | 2.1 | 1.4 | 1.08 | 0.31 | 0.27 | 0.42 | 0.46 | 1 1/4 | 1 3/4 | 5/8 | ..... | 7/16 | 7/8 | 5/16 | 1/2 |
| | 5.0 | 1.46 | 1.498 | 1/4 | .258 | 1.8 | 1.2 | 1.12 | .25 | .24 | .41 | .44 | 1 1/4 | 1 3/4 | 5/8 | ..... | 5/16 | 7/8 | 1/4 | 1/2 |
| | 4.1 | 1.19 | 1.410 | 1/4 | .170 | 1.6 | 1.1 | 1.17 | .20 | .21 | .41 | .44 | 1 1/4 | 1 3/4 | 5/8 | ..... | 1/4 | 7/8 | 1/4 | 1/2 |

## Table 9.—Safe Uniform Loads on American Standard Channels

(Carnegie-Illinois Steel Corp., Pittsburgh)

Applicable only when sections are braced against lateral deflection. Maximum bending stress, 18,000 lb. per sq. in. For safe loads for unbraced sections, see p. 17–48. Maximum and minimum weights of each section only are given. For safe loads of intermediate sections, apply the formulas on p. 7–22.

| Span, ft. | Section No., Nominal Depth and Flange Width and Weight per Foot | | | | | | | | | | | | | | | | | | | | | | | | Coefficient of Deflection |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C60 *, 18 × 4 in. | | C1, 15 × 3 3/8 in. | | C20, 13 × 4 in. | | C2, 12 × 3 in. | | C3, 10 × 2 5/8 in. | | C4, 9 × 2 1/2 in. | | C5, 8 × 2 1/4 in. | | C6, 7 × 2 1/8 in. | | C7, 6 × 1 7/8 in. | | C8, 5 × 1 3/4 in. | | C9, 4 × 1 5/8 in. | | C10, 3 × 1 3/8 in. | | |
| | 58.0 lb. | 42.7 lb. | 55 lb. | 33.9 lb. | 50 lb. | 31.8 lb. | 40 lb. | 20.7 lb. | 35 lb. | 15.3 lb. | 25 lb. | 13.4 lb. | 21.25 lb. | 11.5 lb. | 19.75 lb. | 9.8 lb. | 15.5 lb. | 8.2 lb. | 11.5 lb. | 6.7 lb. | 7.25 lb. | 5.4 lb. | 6 lb. | 4.1 lb. | |
| | Safe Load, Kips † | | | | | | | | | | | | | | | | | | | | | | | | |
| | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 80.5 | .... | 56.6 | .... | 30.7 | .... | 25.6 | .... | .... |
| 1 | .... | .... | .... | .... | .... | .... | 217.4 | .... | 196.8 | .... | 132.2 | .... | 111.2 | .... | 105.7 | .... | 77.8 | 28.8 | 49.7 | 22.8 | 27.3 | 17.3 | 16.4 | 12.2 | 0.02 |
| 2 | 302.4 | .... | 293.0 | .... | 245.5 | .... | 196.5 | .... | 138.3 | 57.6 | 94.0 | 49.7 | 71.4 | 42.2 | 56.7 | 35.3 | 38.9 | 26.0 | 24.8 | 17.8 | 13.6 | 11.4 | 8.2 | 6.5 | .07 |
| 3 | 298.1 | 194.4 | 228.8 | 144.0 | 192.6 | 117.0 | 131.0 | 80.6 | 92.2 | 53.5 | 62.7 | 42.1 | 47.6 | 32.3 | 37.8 | 24.1 | 25.9 | 17.3 | 16.6 | 11.9 | 9.1 | 7.6 | 5.5 | 4.4 | .17 |
| 4 | 223.6 | 183.1 | 171.6 | 125.0 | 144.4 | 109.6 | 98.3 | 64.1 | 69.1 | 40.1 | 47.0 | 31.5 | 35.7 | 24.2 | 28.3 | 18.1 | 19.5 | 13.0 | 12.4 | 8.9 | 6.8 | 5.7 | 4.1 | 3.3 | .30 |
| 5 | 178.9 | 146.5 | 137.3 | 100.0 | 115.5 | 87.7 | 78.6 | 51.2 | 55.3 | 32.1 | 37.6 | 25.2 | 28.6 | 19.4 | 22.7 | 14.5 | 15.6 | 10.4 | 9.9 | 7.1 | 5.5 | 4.6 | 3.3 | 2.6 | .47 |
| 6 | 149.1 | 122.1 | 114.4 | 83.4 | 96.3 | 73.1 | 65.5 | 42.7 | 46.1 | 26.8 | 31.4 | 21.0 | 23.8 | 16.2 | 18.9 | 12.0 | 13.0 | 8.7 | 8.3 | 5.9 | 4.5 | 3.8 | 2.7 | 2.2 | .67 |
| 7 | 127.8 | 104.6 | 98.1 | 71.4 | 82.5 | 62.6 | 56.2 | 36.6 | 39.5 | 22.9 | 26.9 | 18.0 | 20.4 | 13.8 | 16.2 | 10.3 | 11.1 | 7.4 | 7.1 | 5.1 | 3.9 | 3.3 | 2.3 | 1.9 | .91 |
| 8 | 111.8 | 91.5 | 85.8 | 62.5 | 72.2 | 54.8 | 49.1 | 32.0 | 34.6 | 20.1 | 23.5 | 15.8 | 17.9 | 12.1 | 14.2 | 9.0 | 9.7 | 6.5 | 6.2 | 4.4 | 3.4 | 2.8 | 2.1 | 1.6 | 1.19 |
| 9 | 99.4 | 81.4 | 76.3 | 55.6 | 64.2 | 48.7 | 43.7 | 28.5 | 30.7 | 17.8 | 20.9 | 14.0 | 15.9 | 10.8 | 12.6 | 8.0 | 8.6 | 5.8 | 5.5 | 4.0 | 3.0 | 2.5 | .... | .... | 1.51 |
| 10 | 89.4 | 73.2 | 68.6 | 50.0 | 57.8 | 43.8 | 39.3 | 25.6 | 27.7 | 16.1 | 18.8 | 12.6 | 14.3 | 9.7 | 11.3 | 7.2 | 7.8 | 5.2 | 5.0 | 3.6 | 2.7 | 2.3 | .... | .... | 1.86 |
| 11 | 81.3 | 66.6 | 62.4 | 45.5 | 52.5 | 39.9 | 35.7 | 23.3 | 25.1 | 14.6 | 17.1 | 11.5 | 13.0 | 8.8 | 10.3 | 6.6 | 7.1 | 4.7 | 4.5 | 3.2 | .... | .... | .... | .... | 2.25 |
| 12 | 74.5 | 61.0 | 57.2 | 41.7 | 48.1 | 36.5 | 32.8 | 21.4 | 23.0 | 13.4 | 15.7 | 10.5 | 11.9 | 8.1 | 9.4 | 6.0 | 6.5 | 4.3 | .... | .... | .... | .... | .... | .... | 2.68 |
| 13 | 68.8 | 56.3 | 52.8 | 38.5 | 44.4 | 33.7 | 30.2 | 19.7 | 21.3 | 12.3 | 14.5 | 9.7 | 11.0 | 7.5 | 8.7 | 5.6 | 6.0 | 4.0 | .... | .... | .... | .... | .... | .... | 3.15 |
| 14 | 63.9 | 52.3 | 49.0 | 35.7 | 41.3 | 31.3 | 28.1 | 18.3 | 19.8 | 11.5 | 13.4 | 9.0 | 10.2 | 6.9 | 8.1 | 5.2 | .... | .... | .... | .... | .... | .... | .... | .... | 3.65 |
| 15 | 59.6 | 48.8 | 45.8 | 33.3 | 38.5 | 29.2 | 26.2 | 17.1 | 18.4 | 10.7 | 12.5 | 8.4 | 9.5 | 6.5 | 7.6 | 4.8 | .... | .... | .... | .... | .... | .... | .... | .... | 4.19 |
| 16 | 55.9 | 45.8 | 42.9 | 31.3 | 36.1 | 27.4 | 24.6 | 16.0 | 17.3 | 10.0 | 11.8 | 7.9 | 8.9 | 6.1 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 4.77 |
| 17 | 52.6 | 43.1 | 40.4 | 29.4 | 34.0 | 25.8 | 23.1 | 15.1 | 16.3 | 9.4 | 11.1 | 7.4 | 8.4 | 5.7 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 5.38 |
| 18 | 49.7 | 40.7 | 38.1 | 27.8 | 32.1 | 24.4 | 21.8 | 14.2 | 15.4 | 8.9 | 10.4 | 7.0 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 6.03 |
| 19 | 47.1 | 38.5 | 36.1 | 26.3 | 30.4 | 23.1 | 20.7 | 13.5 | 14.6 | 8.4 | 9.9 | 6.6 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 6.72 |
| 20 | 44.7 | 36.6 | 34.3 | 25.0 | 28.9 | 21.9 | 19.7 | 12.8 | 13.8 | 8.0 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 7.45 |
| 21 | 42.6 | 34.9 | 32.7 | 23.8 | 27.5 | 20.9 | 18.7 | 12.2 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 8.21 |
| 22 | 40.7 | 33.3 | 31.2 | 22.7 | 26.3 | 19.9 | 17.9 | 11.6 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 9.01 |
| 23 | 38.9 | 31.8 | 29.8 | 21.7 | 25.1 | 19.1 | 17.1 | 11.1 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 9.85 |
| 24 | 37.3 | 30.5 | 28.6 | 20.8 | 24.1 | 18.3 | 16.4 | 10.7 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 10.73 |
| 25 | 35.8 | 29.3 | 27.5 | 20.0 | 23.1 | 17.5 | .... | .... | .... | .... | .... | .... | | | | | | | .... | .... | .... | .... | .... | .... | 11.64 |
| 26 | 34.4 | 28.2 | 26.4 | 19.2 | 22.2 | 16.9 | .... | .... | .... | .... | .... | .... | | | | | | | .... | .... | .... | .... | .... | .... | 12.59 |
| 27 | 33.1 | 27.1 | 25.4 | 18.5 | .... | .... | .... | .... | .... | .... | .... | .... | | | | | | | .... | .... | .... | .... | .... | .... | 13.57 |
| 28 | 31.9 | 26.2 | 24.5 | 17.9 | .... | .... | .... | .... | .... | .... | .... | .... | | | | | | | .... | .... | .... | .... | .... | .... | 14.60 |
| 29 | 30.8 | 25.3 | 23.7 | 17.2 | .... | .... | .... | .... | .... | .... | .... | .... | | | | | | | .... | .... | .... | .... | .... | .... | 15.66 |
| 30 | 29.8 | 24.4 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 16.76 |
| 31 | 28.8 | 23.6 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 17.89 |
| 32 | 28.0 | 22.9 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 19.07 |
| 33 | 27.1 | 22.2 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 20.28 |
| 34 | 26.3 | 21.5 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 21.53 |
| 35 | 25.5 | 20.9 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 22.81 |

Loads above upper horizontal lines will produce maximum allowable shear in webs.

Loads below lower horizontal lines will produce excessive deflections.

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* C60, 18 in. channel is a ship building channel, not American Standard.

† 1 kip = 1000 lb.

## 2. ANGLES, ZEES AND TEES

The properties of unequal and equal angles are given in Tables 12 and 13 respectively. Safe loads on angles used as beams are given in Tables 10 and 14.

**RADIUS OF GYRATION OF TWO ANGLES.**—The values of Table 16 for radii of gyration of two angles are employed for computing the safe resistance to compressive stress of

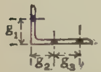
### Table 10.—Safe Uniform Load on Unequal Steel Angles Used as Beams.

Maximum, Intermediate and Minimum Weights
(Carnegie Illinois Steel Corporation, Pittsburgh)
Applicable only when sections are rigidly secured against lateral deflection
Maximum bending stress, 18,000 lb. per sq. in.

| Size, in. | Thickness, in. | Neutral Axis Parallel to Short Leg | | | Neutral Axis Parallel to Long Leg | | |
|---|---|---|---|---|---|---|---|
| | | 1 Foot Span | Maximum Span, 360 × Deflection | | 1 Foot Span | Maximum Span, 360 × Deflection | |
| | | Safe Load, Kips * | Safe Load, Kips * | Length, ft. | Safe Load, Kips * | Safe Load, Kips * | Length, ft. |
| 8 × 6 | 1 1/8 | 201.60 | 10.61 | 19.0 | 118.80 | 7.71 | 15.4 |
| | 3/4 | 140.04 | 7.19 | 19.5 | 83.16 | 5.23 | 15.9 |
| | 7/16 | 84.84 | 4.27 | 19.9 | 50.76 | 3.12 | 16.3 |
| 8 × 4 | 1 | 169.20 | 9.56 | 17.7 | 46.80 | 4.42 | 10.6 |
| | 3/4 | 130.80 | 7.23 | 18.1 | 37.20 | 3.41 | 10.9 |
| | 7/16 | 79.20 | 4.28 | 18.5 | 22.80 | 2.02 | 11.3 |
| 7 × 4 | 1 | 129.60 | 8.20 | 15.8 | 46.80 | 4.50 | 10.4 |
| | 11/16 | 93.60 | 5.81 | 16.1 | 33.60 | 3.11 | 10.8 |
| | 3/8 | 52.80 | 3.18 | 16.6 | 19.20 | 1.71 | 11.2 |
| 6 × 4 | 1 | 96.24 | 7.02 | 13.7 | 45.48 | 4.49 | 10.1 |
| | 11/16 | 69.36 | 4.92 | 14.1 | 33.12 | 3.14 | 10.5 |
| | 3/8 | 39.84 | 2.74 | 14.5 | 19.20 | 1.75 | 11.0 |
| 6 × 3 1/2 | 1 | 93.96 | 7.02 | 13.4 | 34.80 | 3.91 | 8.9 |
| | 5/8 | 62.28 | 4.49 | 13.9 | 23.28 | 2.48 | 9.4 |
| | 5/16 | 32.88 | 2.30 | 14.3 | 12.48 | 1.27 | 9.8 |
| 5 × 3 1/2 | 7/8 | 58.56 | 5.10 | 11.5 | 30.24 | 3.43 | 8.8 |
| | 5/8 | 43.80 | 3.71 | 11.8 | 22.80 | 2.50 | 9.1 |
| | 5/16 | 23.28 | 1.91 | 12.2 | 12.24 | 1.29 | 9.5 |
| 5 × 3 | 13/16 | 53.40 | 4.75 | 11.2 | 20.88 | 2.73 | 7.7 |
| | 9/16 | 38.76 | 3.35 | 11.6 | 15.24 | 1.91 | 8.0 |
| | 5/16 | 22.68 | 1.91 | 11.9 | 9.00 | 1.08 | 8.3 |
| 4 × 3 1/2 | 13/16 | 35.04 | 3.71 | 9.5 | 27.60 | 3.22 | 8.6 |
| | 9/16 | 25.80 | 2.64 | 9.8 | 20.16 | 2.27 | 8.9 |
| | 5/16 | 15.12 | 1.50 | 10.1 | 12.00 | 1.31 | 9.2 |
| 4 × 3 | 13/16 | 34.44 | 3.76 | 9.2 | 20.16 | 2.74 | 7.4 |
| | 1/2 | 22.68 | 2.37 | 9.6 | 13.32 | 1.71 | 7.8 |
| | 1/4 | 12.00 | 1.22 | 9.9 | 7.20 | 0.89 | 8.1 |
| 3 1/2 × 3 | 13/16 | 26.40 | 3.25 | 8.1 | 19.80 | 2.74 | 7.2 |
| | 1/2 | 17.40 | 2.05 | 8.5 | 13.20 | 1.74 | 7.6 |
| | 1/4 | 9.36 | 1.06 | 8.8 | 6.96 | 0.88 | 7.9 |

| Size, in. | Thickness, in. | Neutral Axis Parallel to Short Leg | | | Neutral Axis Parallel to Long Leg | | |
|---|---|---|---|---|---|---|---|
| | | 1 Foot Span | Maximum Span, 360 × Deflection | | 1 Foot Span | Maximum Span, 360 × Deflection | |
| | | Safe Load, Kips* | Safe Load, Kips * | Length, ft. | Safe Load, Kips * | Safe Load, Kips * | Length, ft. |
| 3 1/2 × 2 1/2 | 11/16 | 22.20 | 2.78 | 8.0 | 11.88 | 1.92 | 6.2 |
| | 1/2 | 16.92 | 2.06 | 8.2 | 9.12 | 1.42 | 6.4 |
| | 1/4 | 9.00 | 1.05 | 8.6 | 4.92 | 0.73 | 6.8 |
| 3 × 2 1/2 | 9/16 | 13.80 | 1.95 | 7.1 | 9.84 | 1.59 | 6.2 |
| | 7/16 | 11.16 | 1.54 | 7.2 | 7.92 | 1.25 | 6.3 |
| | 1/4 | 6.72 | 0.90 | 7.5 | 4.80 | 0.73 | 6.6 |
| 3 × 2 | 1/2 | 12.00 | 1.75 | 6.9 | 5.64 | 1.11 | 5.1 |
| | 3/8 | 9.36 | 1.33 | 7.0 | 4.44 | 0.85 | 5.2 |
| | 1/4 | 6.48 | 0.90 | 7.2 | 3.00 | 0.58 | 5.4 |
| 2 1/2 × 2 | 1/2 | 8.40 | 1.45 | 5.8 | 5.52 | 1.12 | 4.9 |
| | 5/16 | 5.64 | 0.93 | 6.1 | 3.72 | 0.72 | 5.2 |
| | 1/8 | 2.40 | .38 | 6.3 | 1.62 | .30 | 5.4 |
| 2 1/2 × 1 1/2 | 5/16 | 5.28 | .92 | 5.7 | 2.04 | .52 | 3.9 |
| | 1/4 | 4.32 | .75 | 5.8 | 1.68 | .41 | 4.1 |
| | 3/16 | 3.36 | .57 | 5.9 | 1.32 | .32 | 4.1 |
| 2 1/4 × 1 1/2 | 1/2 | 6.48 | 1.30 | 5.0 | 3.12 | .85 | 3.7 |
| | 5/16 | 4.32 | 0.83 | 5.2 | 2.04 | .53 | 3.9 |
| | 3/16 | 2.76 | .51 | 5.4 | 1.32 | .33 | 4.0 |
| 2 × 1 1/2 | 3/8 | 4.08 | .88 | 4.6 | 2.40 | .65 | 3.7 |
| | 1/4 | 2.88 | .60 | 4.8 | 1.68 | .43 | 3.9 |
| | 1/8 | 1.50 | .30 | 4.9 | 0.90 | .22 | 4.0 |
| 2 × 1 1/4 | 1/4 | 2.75 | .60 | 4.6 | 1.16 | .35 | 3.3 |
| | 3/16 | 2.12 | .45 | 4.7 | 0.90 | .27 | 3.4 |
| 1 3/4 × 1 1/4 | 1/4 | 2.11 | .51 | 4.1 | 1.14 | .35 | 3.2 |
| | 3/16 | 1.64 | .39 | 4.2 | 0.90 | .27 | 3.3 |
| | 1/8 | 1.13 | .27 | 4.3 | 0.62 | .19 | 3.4 |
| 1 1/2 × 1 1/4 | 5/16 | 1.92 | .55 | 3.5 | 1.36 | .45 | 3.0 |
| | 1/4 | 1.56 | .44 | 3.6 | 1.12 | .36 | 3.1 |
| | 3/16 | 1.21 | .33 | 3.7 | 0.88 | .27 | 3.2 |

* 1 Kip = 1000 lb.



### Table 11.—Gages for Angles

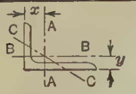
(American Bridge Co. Standard)

| Gage | Width of Leg, in. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3/4 | 1 | 1 1/4 | 1 3/8 | 1 1/2 | 1 3/4 | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 5 | 6 | 7 | 8 |
| | Dimension, in. | | | | | | | | | | | | | | |
| $g_1$ | 1/2 | 5/8 | 3/4 | 7/8 | 7/8 | 1 | 1 1/8 | 1 3/8 | 1 3/4 | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 |
| $g_2$ | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 2 | 2 1/4 | 2 1/2 | 3 |
| $g_3$ | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 1 3/4 | 2 1/2 | 3 | 3 |
| Max. rivet. Diam., in. | 1/4 | 1/4 | 3/8 | 3/8 | 3/8 | 1/2 | 5/8 | 3/4 | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 | 1 | 1 1/8 |

### Table 12.—Properties of Rolled Steel Unequal Angles. Elements of Sections.

Maximum, Intermediate and Minimum Thicknesses and Weights

(Carnegie–Illinois Steel Corporation, Pittsburgh)

|  | | Wt. per Foot, lb. | Area of Section, sq. in. | Axis A–A | | | | Axis B–B | | | | Axis C–C |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Size, in. | Thickness, in. | | | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | $x$ | Moment of Inertia, $I'$, in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r'$, in. | $y$ | Least Radius of Gyration, $r''$, in. |
| 8×6 | 1 1/8 | 49.3 | 14.48 | 88.9 | 16.8 | 2.48 | 2.70 | 42.5 | 9.9 | 1.71 | 1.70 | 1.28 |
| | 3/4 | 33.8 | 9.94 | 63.4 | 11.7 | 2.53 | 2.56 | 30.7 | 6.9 | 1.76 | 1.56 | 1.29 |
| | 7/16 | 20.2 | 5.93 | 39.2 | 7.1 | 2.57 | 2.45 | 19.3 | 4.2 | 1.80 | 1.45 | 1.30 |
| 8×4 | 1 | 37.4 | 11.00 | 69.6 | 14.1 | 2.52 | 3.05 | 11.6 | 3.9 | 1.03 | 1.05 | 0.85 |
| | 3/4 | 28.7 | 8.44 | 54.9 | 10.9 | 2.55 | 2.95 | 9.4 | 3.1 | 1.05 | 0.95 | .85 |
| | 7/16 | 17.2 | 5.06 | 34.1 | 6.6 | 2.60 | 2.83 | 6.0 | 1.9 | 1.09 | 0.83 | .87 |
| 7×4 | 1 | 34.0 | 10.00 | 47.7 | 10.8 | 2.18 | 2.60 | 11.2 | 3.9 | 1.06 | 1.10 | .85 |
| | 11/16 | 24.2 | 7.09 | 35.1 | 7.8 | 2.23 | 2.49 | 8.5 | 2.8 | 1.09 | 0.99 | .86 |
| | 3/8 | 13.6 | 3.99 | 20.6 | 4.4 | 2.27 | 2.37 | 5.1 | 1.6 | 1.13 | 0.87 | .88 |
| 6×4 | 1 | 30.6 | 9.00 | 30.8 | 8.0 | 1.85 | 2.17 | 10.8 | 3.8 | 1.09 | 1.17 | .85 |
| | 11/16 | 21.8 | 6.40 | 22.8 | 5.8 | 1.89 | 2.06 | 8.1 | 2.8 | 1.13 | 1.06 | .86 |
| | 3/8 | 12.3 | 3.61 | 13.5 | 3.3 | 1.93 | 1.94 | 4.9 | 1.6 | 1.17 | 0.94 | .88 |
| 6×3 1/2 | 1 | 28.9 | 8.50 | 29.2 | 7.8 | 1.85 | 2.26 | 7.2 | 2.9 | 0.92 | 1.01 | .74 |
| | 11/16 | 20.6 | 6.06 | 21.7 | 5.6 | 1.89 | 2.15 | 5.5 | 2.1 | 0.95 | 0.90 | .75 |
| | 5/16 | 9.8 | 2.87 | 10.9 | 2.7 | 1.95 | 2.01 | 2.9 | 1.0 | 1.00 | 0.76 | .77 |
| 5×3 1/2 | 7/8 | 22.7 | 6.67 | 15.7 | 4.9 | 1.53 | 1.79 | 6.2 | 2.5 | 0.96 | 1.04 | .75 |
| | 5/8 | 16.8 | 4.92 | 12.0 | 3.7 | 1.56 | 1.70 | 4.8 | 1.9 | 0.99 | 0.95 | .75 |
| | 5/16 | 8.7 | 2.56 | 6.6 | 1.9 | 1.61 | 1.59 | 2.7 | 1.0 | 1.03 | .84 | .76 |
| 5×3 | 13/16 | 19.9 | 5.84 | 14.0 | 4.5 | 1.55 | 1.86 | 3.7 | 1.7 | 0.80 | .86 | .64 |
| | 9/16 | 14.3 | 4.18 | 10.4 | 3.2 | 1.58 | 1.77 | 2.8 | 1.3 | 0.82 | .77 | .65 |
| | 5/16 | 8.2 | 2.40 | 6.3 | 1.9 | 1.61 | 1.68 | 1.8 | 0.75 | 0.85 | .68 | .66 |
| 4×3 1/2 | 13/16 | 18.5 | 5.43 | 7.8 | 2.9 | 1.19 | 1.36 | 5.5 | 2.3 | 1.01 | 1.11 | .72 |
| | 9/16 | 13.3 | 3.90 | 5.9 | 2.1 | 1.23 | 1.27 | 4.2 | 1.7 | 1.03 | 1.02 | .72 |
| | 5/16 | 7.7 | 2.25 | 3.6 | 1.3 | 1.26 | 1.18 | 2.6 | 1.0 | 1.07 | 0.93 | .73 |
| 4×3 | 13/16 | 17.1 | 5.03 | 7.3 | 2.9 | 1.21 | 1.44 | 3.5 | 1.7 | 0.83 | .94 | .64 |
| | 1/2 | 11.1 | 3.25 | 5.0 | 1.9 | 1.25 | 1.33 | 2.4 | 1.1 | .86 | .83 | .64 |
| | 1/4 | 5.8 | 1.69 | 2.8 | 1.0 | 1.28 | 1.24 | 1.4 | 0.60 | .89 | .74 | .65 |
| 3 1/2×3 | 13/16 | 15.8 | 4.62 | 5.0 | 2.2 | 1.04 | 1.23 | 3.3 | 1.7 | .85 | .98 | .62 |
| | 1/2 | 10.2 | 3.00 | 3.5 | 1.5 | 1.07 | 1.13 | 2.3 | 1.1 | .88 | .88 | .62 |
| | 1/4 | 5.4 | 1.56 | 1.9 | 0.78 | 1.11 | 1.04 | 1.3 | 0.58 | .91 | .79 | .63 |
| 3 1/2×2 1/2 | 11/16 | 12.5 | 3.65 | 4.1 | 1.9 | 1.06 | 1.27 | 1.7 | .99 | .69 | .77 | .53 |
| | 1/2 | 9.4 | 2.75 | 3.2 | 1.4 | 1.09 | 1.20 | 1.4 | .76 | .70 | .70 | .53 |
| | 1/4 | 4.9 | 1.44 | 1.8 | 0.75 | 1.12 | 1.11 | 0.78 | .41 | .74 | .61 | .54 |
| 3×2 1/2 | 9/16 | 9.5 | 2.78 | 2.3 | 1.2 | 0.91 | 1.02 | 1.4 | .82 | .72 | .77 | .52 |
| | 7/16 | 7.6 | 2.21 | 1.9 | 0.93 | .92 | 0.98 | 1.2 | .66 | .73 | .73 | .52 |
| | 1/4 | 4.5 | 1.31 | 1.2 | 0.56 | .95 | 0.91 | 0.74 | .40 | .75 | .66 | .53 |
| 3×2 | 1/2 | 7.7 | 2.25 | 1.9 | 1.0 | .92 | 1.08 | .67 | .47 | .55 | .58 | .43 |
| | 3/8 | 5.9 | 1.73 | 1.5 | 0.78 | .94 | 1.04 | .54 | .37 | .56 | .54 | .43 |
| | 1/4 | 4.1 | 1.19 | 1.1 | .54 | .95 | 0.99 | .39 | .26 | .57 | .49 | .43 |
| 2 1/2×2 | 1/2 | 6.8 | 2.00 | 1.1 | .70 | .75 | .88 | .64 | .46 | .56 | .63 | .42 |
| | 5/16 | 4.5 | 1.31 | 0.79 | .47 | .78 | .81 | .45 | .31 | .58 | .56 | .42 |
| | 1/8 | 1.86 | 0.55 | .35 | .20 | .80 | .74 | .20 | .13 | .61 | .49 | .43 |
| 2 1/2×1 1/2 | 5/16 | 3.92 | 1.15 | .71 | .44 | .79 | .90 | .19 | .17 | .41 | .40 | .32 |
| | 1/4 | 3.19 | 0.94 | .59 | .36 | .79 | .88 | .16 | .14 | .41 | .38 | .32 |
| | 3/16 | 2.44 | 0.72 | .46 | .28 | .80 | .85 | .13 | .11 | .42 | .35 | .33 |
| 2×1 1/2 | 3/8 | 3.99 | 1.17 | .43 | .34 | .61 | .71 | .21 | .20 | .42 | .46 | .32 |
| | 1/4 | 2.77 | 0.81 | .32 | .24 | .62 | .66 | .15 | .14 | .43 | .41 | .32 |
| | 1/8 | 1.44 | .42 | .17 | .13 | .64 | .62 | .09 | .08 | .45 | .37 | .33 |
| 1 3/4×1 1/4 | 1/4 | 2.34 | .69 | .20 | .18 | .54 | .60 | .09 | .10 | .35 | .35 | .27 |
| | 3/16 | 1.80 | .53 | .16 | .14 | .55 | .58 | .07 | .08 | .36 | .33 | .27 |
| | 1/8 | 1.23 | .36 | .11 | .09 | .56 | .56 | .05 | .05 | .37 | .31 | .27 |
| 1 3/8×7/8 | 3/16 | 1.32 | .39 | .071 | .081 | .43 | .49 | .022 | .035 | .24 | .24 | .19 |
| | 1/8 | 0.91 | .27 | .051 | .056 | .44 | .47 | .017 | .026 | .25 | .22 | .20 |

two angles placed back to back, used as a strut or the compression chord of a truss. The angles must be riveted or otherwise fastened together so that they act as a unit. The allowed stress per sq. in. corresponding to the ratio of $l/r$ of the section is obtained from the compression formula used, $r$ being taken from the table. The stress value so found is multiplied by the area of the two angles, the result being the allowable compressive stress.

Example.—Required the safe load in compression on a strut of two angles, 4 × 4 × 1/4 in., placed back to back; unsupported length, 9 ft. From Table 16, area of section = 3.88; least radius = 1.25; $l/r = 9 \times 12 \div 1.25 = 86.4$. Compression formula used,

$$f = 18{,}000 \div [1 + \{(l/r)^2 \div 18{,}000\}].$$

$$18{,}000 \div [1 + \{(86.4)^2 + 18{,}000\}] = 12{,}273 \text{ lb. per sq. in.}$$

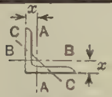
Safe load = 3.88 × 12,273 = 49,365 lb.

**GAGES FOR ANGLES.**—Table 11 gives gages for punching or drilling angles for riveting, based on the standards of the American Bridge Co.

**Table 13.—Properties of Rolled Steel Equal Angles. Elements of Sections.**

Maximum, Intermediate and Minimum Thicknesses and Weights

(Carnegie-Illinois Steel Corporation, Pittsburgh)

|  Size, in. | Thickness, in. | Weight per foot, lb. | Area of Section, sq. in. | Axis A–A and Axis B–B | | | | Axis C–C |
|---|---|---|---|---|---|---|---|---|
| | | | | Moment of Inertia, $I$, in⁴. | Section Modulus, $S$, in³. | Radius of Gyration, $r$, in. | $x$ | Min. Radius of Gyration, $r'$, in. |
| 8 × 8 | 1 1/8 | 56.9 | 16.73 | 98.0 | 17.5 | 2.42 | 2.41 | 1.55 |
| | 13/16 | 42.0 | 12.34 | 74.7 | 13.1 | 2.46 | 2.30 | 1.57 |
| | 1/2 | 26.4 | 7.75 | 48.6 | 8.4 | 2.51 | 2.19 | 1.58 |
| 6 × 6 | 1 1/16 | 39.6 | 11.62 | 37.2 | 9.0 | 1.79 | 1.89 | 1.16 |
| | 11/16 | 26.5 | 7.78 | 26.2 | 6.2 | 1.83 | 1.75 | 1.17 |
| | 3/8 | 14.9 | 4.36 | 15.4 | 3.5 | 1.88 | 1.64 | 1.19 |
| 5 × 5 | 1 | 30.6 | 9.00 | 19.6 | 5.8 | 1.48 | 1.61 | 0.96 |
| | 11/16 | 21.8 | 6.40 | 14.7 | 4.2 | 1.51 | 1.50 | 0.97 |
| | 3/8 | 12.3 | 3.61 | 8.7 | 2.4 | 1.56 | 1.39 | 0.99 |
| 4 × 4 | 13/16 | 19.9 | 5.84 | 8.1 | 3.0 | 1.18 | 1.29 | 0.77 |
| | 9/16 | 14.3 | 4.18 | 6.1 | 2.2 | 1.21 | 1.21 | 0.78 |
| | 1/4 | 6.6 | 1.94 | 3.0 | 1.0 | 1.25 | 1.09 | 0.79 |
| 3 1/2 × 3 1/2 | 13/16 | 17.1 | 5.03 | 5.3 | 2.3 | 1.02 | 1.17 | 0.67 |
| | 9/16 | 12.4 | 3.62 | 4.0 | 1.6 | 1.05 | 1.08 | 0.68 |
| | 1/4 | 5.8 | 1.69 | 2.0 | 0.79 | 1.09 | 0.97 | 0.69 |
| 3 × 3 | 5/8 | 11.5 | 3.36 | 2.6 | 1.3 | 0.88 | 0.98 | 0.57 |
| | 7/16 | 8.3 | 2.43 | 2.0 | 0.95 | 0.91 | 0.91 | 0.58 |
| | 1/4 | 4.9 | 1.44 | 1.2 | 0.58 | 0.93 | 0.84 | 0.59 |
| 2 1/2 × 2 1/2 | 1/2 | 7.7 | 2.25 | 1.2 | 0.73 | 0.74 | 0.81 | 0.47 |
| | 5/16 | 5.0 | 1.47 | 0.85 | 0.48 | 0.76 | 0.74 | 0.49 |
| | 1/8 | 2.08 | 0.61 | 0.38 | 0.20 | 0.79 | 0.67 | 0.50 |
| 2 × 2 | 7/16 | 5.3 | 1.56 | 0.54 | 0.40 | 0.59 | 0.66 | 0.39 |
| | 1/4 | 3.19 | 0.94 | 0.35 | 0.25 | 0.61 | 0.59 | 0.39 |
| | 1/8 | 1.65 | 0.48 | 0.19 | 0.13 | 0.63 | 0.55 | 0.40 |
| 1 3/4 × 1 3/4 | 5/16 | 3.39 | 1.00 | 0.27 | 0.23 | 0.52 | 0.55 | 0.34 |
| | 3/16 | 2.12 | 0.62 | 0.18 | 0.14 | 0.54 | 0.51 | 0.35 |
| | 1/8 | 1.44 | 0.42 | 0.13 | 0.10 | 0.55 | 0.48 | 0.35 |
| 1 1/2 × 1 1/2 | 3/8 | 3.35 | 0.98 | 0.19 | 0.19 | 0.44 | 0.51 | 0.29 |
| | 1/4 | 2.34 | 0.69 | 0.14 | 0.13 | 0.45 | 0.47 | 0.29 |
| | 1/8 | 1.23 | 0.36 | 0.08 | 0.07 | 0.46 | 0.42 | 0.30 |
| 1 1/4 × 1 1/4 | 5/16 | 2.33 | 0.68 | 0.09 | 0.11 | 0.36 | 0.42 | 0.24 |
| | 3/16 | 1.48 | 0.43 | 0.06 | 0.07 | 0.38 | 0.38 | 0.24 |
| | 1/8 | 1.01 | 0.30 | 0.04 | 0.05 | 0.38 | 0.35 | 0.25 |
| 1 × 1 | 1/4 | 1.49 | 0.44 | 0.04 | 0.06 | 0.29 | 0.34 | 0.19 |
| | 3/16 | 1.16 | 0.34 | 0.03 | 0.04 | 0.30 | 0.32 | 0.19 |
| | 1/8 | 0.80 | 0.23 | 0.02 | 0.03 | 0.31 | 0.30 | 0.19 |

### Table 14.—Safe Uniform Load on Equal Steel Angles Used as Beams

Maximum, Intermediate and Minimum Weights
(Carnegie-Illinois Steel Corporation, Pittsburgh)
Applicable only when sections are rigidly secured against lateral deflection. Maximum bending stress, 18,000 kips * per sq. in.

| Size of Angle, in. | Thickness, in. | 1 Foot Span | Maximum Span. 360 × Deflection | | Size of Angle, in. | Thickness, in. | 1 Foot Span | Maximum Span. 360 × Deflection | |
|---|---|---|---|---|---|---|---|---|---|
| | | Safe Load, Kips * | Safe Load, Kips * | Length, ft. | | | Safe Load, Kips * | Safe Load, Kips * | Length, ft. |
| 8 × 8 | 1 1/8 | 210.36 | 10.51 | 20.0 | 2 1/2 × 2 1/2 | 1/2 | 8.76 | 1.45 | 6.1 |
| | 13/16 | 157.32 | 7.71 | 20.4 | | 5/16 | 5.76 | 0.92 | 6.3 |
| | 1/2 | 100.44 | 4.83 | 20.8 | | 1/8 | 2.40 | 0.37 | 6.6 |
| 6 × 6 | 1 1/16 | 108.36 | 7.37 | 14.7 | 2 × 2 | 7/16 | 4.80 | 1.00 | 4.8 |
| | 3/4 | 79.92 | 5.29 | 15.1 | | 5/16 | 3.60 | 0.72 | 5.0 |
| | 3/8 | 42.36 | 2.71 | 15.6 | | 1/8 | 1.56 | .30 | 5.2 |
| 5 × 5 | 1 | 69.60 | 5.73 | 12.1 | 1 3/4 × 1 3/4 | 5/16 | 2.76 | .64 | 4.3 |
| | 11/16 | 50.40 | 4.02 | 12.5 | | 1/4 | 2.28 | .52 | 4.4 |
| | 3/8 | 29.04 | 2.25 | 12.9 | | 1/8 | 1.20 | .26 | 4.6 |
| 4 × 4 | 13/16 | 36.12 | 3.72 | 9.7 | 1 1/2 × 1 1/2 | 3/8 | 2.28 | .64 | 3.5 |
| | 9/16 | 26.28 | 2.63 | 10.0 | | 1/4 | 1.61 | .44 | 3.7 |
| | 1/4 | 12.60 | 1.21 | 10.4 | | 1/8 | 0.86 | .22 | 3.9 |
| 3 1/2 × 3 1/2 | 13/16 | 27.00 | 3.24 | 8.3 | 1 1/4 × 1 1/4 | 5/16 | 1.31 | .44 | 3.0 |
| | 9/16 | 19.80 | 2.29 | 8.7 | | 1/4 | 1.09 | .36 | 3.0 |
| | 1/4 | 9.48 | 1.05 | 9.1 | | 1/8 | 0.59 | .18 | 3.2 |
| 3 × 3 | 5/8 | 15.60 | 2.16 | 7.2 | 1 × 1 | 1/4 | 0.67 | .28 | 2.4 |
| | 7/16 | 11.40 | 1.52 | 7.5 | | 3/16 | 0.53 | .22 | 2.4 |
| | 1/4 | 6.96 | 0.90 | 7.7 | | 1/8 | 0.37 | .15 | 2.5 |

### Table 15.—Allowable Tension Values for Angles

Maximum, Intermediate and Minimum Weights
(Carnegie-Illinois Steel Corporation, Pittsburgh)
Net stresses given in Kips.* Unit stress, 18 Kips per sq. in.

| Size, in. | Thickness, in. | Wt. per ft., lb. | Area, sq. in. | Net Areas and Stresses—Two Holes Deducted | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | 7/8-in. Rivets | | 3/4-in. Rivets | | 5/8-in. Rivets | |
| | | | | Area | Stress | Area | Stress | Area | Stress |
| 8 × 8 | 1 1/8 | 56.9 | 16.73 | 14.48 | 260.6 | ........ | ........ | ........ | ........ |
| | 13/16 | 42.0 | 12.34 | 10.71 | 192.8 | 10.92 | 196.6 | ........ | ........ |
| | 1/2 | 26.4 | 7.75 | 6.75 | 121.5 | 6.87 | 123.7 | 7.00 | 126.0 |
| 8 × 6 | 1 1/8 | 49.3 | 14.48 | 12.23 | 220.1 | ........ | ........ | ........ | ........ |
| | 3/4 | 33.8 | 9.94 | 8.44 | 151.9 | 8.63 | 155.3 | ........ | ........ |
| | 7/16 | 20.2 | 5.93 | 5.05 | 90.9 | 5.16 | 92.9 | 5.27 | 94.9 |
| 8 × 4 | 1 | 37.4 | 11.00 | 9.00 | 162.0 | 9.25 | 166.5 | ........ | ........ |
| | 3/4 | 28.7 | 8.44 | 6.94 | 124.9 | 7.13 | 128.3 | ........ | ........ |
| | 7/16 | 17.2 | 5.06 | 4.18 | 75.2 | 4.29 | 77.2 | 4.40 | 79.2 |
| 7 × 4 | 1 | 34.0 | 10.00 | 8.00 | 144.0 | 8.25 | 148.5 | ........ | ........ |
| | 11/16 | 24.2 | 7.09 | 5.71 | 102.8 | 5.89 | 106.0 | ........ | ........ |
| | 3/8 | 13.6 | 3.99 | 3.24 | 58.3 | 3.33 | 59.9 | 3.43 | 61.7 |
| 6 × 6 | 1 1/16 | 39.6 | 11.62 | 9.49 | 170.8 | ........ | ........ | ........ | ........ |
| | 11/16 | 26.5 | 7.78 | 6.40 | 115.2 | 6.58 | 118.4 | ........ | ........ |
| | 3/8 | 14.9 | 4.36 | 3.61 | 65.0 | 3.70 | 66.6 | 3.80 | 68.4 |
| 6 × 4 | 1 | 30.6 | 9.00 | 7.00 | 126.0 | 7.25 | 130.5 | ........ | ........ |
| | 11/16 | 21.8 | 6.40 | 5.02 | 90.4 | 5.20 | 93.6 | ........ | ........ |
| | 3/8 | 12.3 | 3.61 | 2.86 | 51.5 | 2.95 | 53.1 | 3.05 | 54.9 |
| 6 × 3 1/2 | 1 | 28.9 | 8.50 | 6.50 | 117.0 | 6.75 | 121.5 | ........ | ........ |
| | 5/8 | 18.9 | 5.55 | 4.30 | 77.4 | 4.46 | 80.3 | 4.61 | 83.0 |
| | 5/16 | 9.8 | 2.87 | 2.24 | 40.3 | 2.32 | 41.8 | 2.40 | 43.2 |
| 5 × 5 | 1 | 30.6 | 9.00 | 7.00 | 126.0 | 7.25 | 130.5 | ........ | ........ |
| | 11/16 | 21.8 | 6.40 | 5.02 | 90.4 | 5.20 | 93.6 | ........ | ........ |
| | 3/8 | 12.3 | 3.61 | 2.86 | 51.5 | 2.95 | 53.1 | 3.05 | 54.9 |
| 5 × 3 1/2 | 7/8 | 22.7 | 6.67 | 4.92 | 88.6 | 5.14 | 92.5 | ........ | ........ |
| | 5/8 | 16.8 | 4.92 | 3.67 | 66.1 | 3.83 | 68.9 | 3.98 | 71.6 |
| | 5/16 | 8.7 | 2.56 | 1.93 | 34.7 | 2.01 | 36.2 | 2.09 | 37.6 |
| 5 × 3 | 13/16 | 19.9 | 5.84 | 4.21 | 75.8 | 4.42 | 79.6 | ........ | ........ |
| | 9/16 | 14.3 | 4.18 | 3.05 | 54.9 | 3.20 | 57.6 | 3.34 | 60.1 |
| | 5/16 | 8.2 | 2.40 | 1.77 | 31.9 | 1.85 | 33.3 | 1.93 | 34.7 |

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* 1 Kip = 1000 lb.

Table 15.—Allowable Tension Values for Angles (*Continued*)

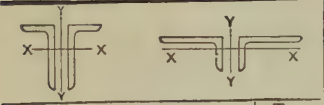
| Size, in. | Thickness in. | Wt. per ft., lb. | Area, sq. in. | Net Areas and Stresses—One Hole Deducted | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | 7/8-in. Rivet | | 3/4-in. Rivet | | 5/8-in. Rivet | |
| | | | | Area | Stress | Area | Stress | Area | Stress |
| 6 × 6 | 1 1/16 | 39.6 | 11.62 | 10.56 | 190.1 | ........ | ........ | ........ | ........ |
| | 11/16 | 26.5 | 7.78 | 7.09 | 127.6 | 7.18 | 129.2 | ........ | ........ |
| | 3/8 | 14.9 | 4.36 | 3.98 | 71.6 | 4.03 | 72.5 | 4.07 | 73.3 |
| 6 × 4 | 1 | 30.6 | 9.00 | 8.00 | 144.0 | 8.12 | 146.2 | ........ | ........ |
| | 11/16 | 21.8 | 6.40 | 5.71 | 102.8 | 5.80 | 104.4 | ........ | ........ |
| | 3/8 | 12.3 | 3.61 | 3.23 | 58.1 | 3.28 | 59.0 | 3.32 | 59.8 |
| 6 × 3 1/2 | 1 | 28.9 | 8.50 | 7.50 | 135.0 | 7.62 | 137.2 | ........ | ........ |
| | 5/8 | 18.9 | 5.55 | 4.92 | 88.6 | 5.00 | 90.0 | 5.08 | 91.4 |
| | 5/16 | 9.8 | 2.87 | 2.56 | 46.1 | 2.60 | 46.8 | 2.64 | 47.5 |
| 5 × 5 | 1 | 30.6 | 9.00 | 8.00 | 144.0 | 8.12 | 146.2 | ........ | ........ |
| | 11/16 | 21.8 | 6.40 | 5.71 | 102.8 | 5.80 | 104.4 | ........ | ........ |
| | 3/8 | 12.3 | 3.61 | 3.23 | 58.1 | 3.28 | 59.0 | 3.32 | 59.8 |
| 5 × 3 1/2 | 7/8 | 22.7 | 6.67 | 5.79 | 104.2 | 5.90 | 106.2 | ........ | ........ |
| | 5/8 | 16.8 | 4.92 | 4.29 | 77.2 | 4.37 | 78.7 | 4.45 | 80.1 |
| | 5/16 | 8.7 | 2.56 | 2.25 | 40.5 | 2.29 | 41.2 | 2.33 | 41.9 |
| 5 × 3 | 13/16 | 19.9 | 5.84 | 5.03 | 90.5 | 5.13 | 92.3 | ........ | ........ |
| | 9/16 | 14.3 | 4.18 | 3.62 | 65.2 | 3.69 | 66.4 | 3.76 | 67.7 |
| | 5/16 | 8.2 | 2.40 | 2.09 | 37.6 | 2.13 | 38.3 | 2.17 | 39.1 |
| 4 × 4 | 13/16 | 19.9 | 5.84 | 5.03 | 90.5 | 5.13 | 92.3 | ........ | ........ |
| | 1/2 | 12.8 | 3.75 | 3.25 | 58.5 | 3.31 | 59.6 | 3.37 | 60.7 |
| | 1/4 | 6.6 | 1.94 | 1.69 | 30.4 | 1.72 | 31.0 | 1.75 | 31.5 |
| 4 × 3 1/2 | 13/16 | 18.5 | 5.43 | 4.62 | 83.2 | 4.72 | 85.0 | ........ | ........ |
| | 9/16 | 13.3 | 3.90 | 3.34 | 60.1 | 3.41 | 61.4 | 3.48 | 62.6 |
| | 5/16 | 7.7 | 2.25 | 1.94 | 34.9 | 1.98 | 35.6 | 2.02 | 36.4 |
| 4 × 3 | 13/16 | 17.1 | 5.03 | 4.22 | 76.0 | 4.32 | 77.8 | ........ | ........ |
| | 1/2 | 11.1 | 3.25 | 2.75 | 49.5 | 2.81 | 50.6 | 2.87 | 51.7 |
| | 1/4 | 5.8 | 1.69 | 1.44 | 25.9 | 1.47 | 26.5 | 1.50 | 27.0 |
| 3 1/2 × 3 1/2 | 13/16 | 17.1 | 5.03 | 4.22 | 76.0 | 4.32 | 77.8 | ........ | ........ |
| | 1/2 | 11.1 | 3.25 | 2.75 | 49.5 | 2.81 | 50.6 | 2.87 | 51.7 |
| | 1/4 | 5.8 | 1.69 | 1.44 | 25.9 | 1.47 | 26.5 | 1.50 | 27.0 |
| 3 1/2 × 3 | 13/16 | 15.8 | 4.62 | 3.81 | 68.6 | 3.91 | 70.4 | ........ | ........ |
| | 1/2 | 10.2 | 3.00 | 2.50 | 45.0 | 2.56 | 46.1 | 2.62 | 47.2 |
| | 1/4 | 5.4 | 1.56 | 1.31 | 23.6 | 1.34 | 24.1 | 1.37 | 24.7 |
| 3 1/2 × 2 1/2 | 11/16 | 12.5 | 3.65 | 2.96 | 53.3 | 3.05 | 54.9 | ........ | ........ |
| | 7/16 | 8.3 | 2.43 | 1.99 | 35.8 | 2.04 | 36.7 | 2.10 | 37.8 |
| | 1/4 | 4.9 | 1.44 | 1.19 | 21.4 | 1.22 | 22.0 | 1.25 | 22.5 |
| 3 × 3 | 5/8 | 11.5 | 3.36 | 2.73 | 49.1 | 2.81 | 50.6 | 2.89 | 52.0 |
| | 7/16 | 8.3 | 2.43 | 1.99 | 35.8 | 2.04 | 36.7 | 2.10 | 37.8 |
| | 1/4 | 4.9 | 1.44 | 1.19 | 21.4 | 1.22 | 22.0 | 1.25 | 22.5 |
| 3 × 2 1/2 | 9/16 | 9.5 | 2.78 | 2.22 | 40.0 | 2.29 | 41.2 | 2.36 | 42.5 |
| | 3/8 | 6.6 | 1.92 | 1.54 | 27.7 | 1.59 | 28.6 | 1.63 | 29.3 |
| | 3/16 | 3.39 | 1.00 | 0.81 | 14.6 | 0.83 | 14.9 | 0.86 | 15.5 |
| 3 × 2 | 1/2 | 7.7 | 2.25 | ........ | ........ | 1.81 | 32.6 | 1.87 | 33.7 |
| | 3/8 | 5.9 | 1.73 | ........ | ........ | 1.40 | 25.2 | 1.44 | 25.9 |
| | 3/16 | 3.07 | 0.90 | ........ | ........ | 0.73 | 13.1 | 0.76 | 13.7 |
| 2 1/2 × 2 1/2 | 1/2 | 7.7 | 2.25 | ........ | ........ | 1.81 | 32.6 | 1.87 | 33.7 |
| | 5/16 | 5.0 | 1.47 | ........ | ........ | 1.20 | 21.6 | 1.24 | 22.3 |
| | 1/8 | 2.08 | 0.61 | ........ | ........ | 0.50 | 9.0 | 0.51 | 9.2 |
| 2 1/2 × 2 | 1/2 | 6.8 | 2.00 | ........ | ........ | 1.56 | 28.1 | 1.62 | 29.2 |
| | 5/16 | 4.5 | 1.31 | ........ | ........ | 1.04 | 18.7 | 1.08 | 19.4 |
| | 1/8 | 1.86 | 0.55 | ........ | ........ | 0.44 | 7.9 | 0.45 | 8.1 |
| 2 1/2 × 1 1/2 | 5/16 | 3.92 | 1.15 | ........ | ........ | ........ | ........ | 0.92 | 16.6 |
| | 1/4 | 3.19 | 0.94 | ........ | ........ | ........ | ........ | 0.75 | 13.5 |
| | 3/16 | 2.44 | 0.72 | ........ | ........ | ........ | ........ | 0.58 | 10.4 |
| 2 × 2 | 7/16 | 5.3 | 1.56 | ........ | ........ | ........ | ........ | 1.23 | 22.1 |
| | 1/4 | 3.19 | 0.94 | ........ | ........ | ........ | ........ | 0.75 | 13.5 |
| | 1/8 | 1.65 | 0.48 | ........ | ........ | ........ | ........ | 0.38 | 6.8 |
| 2 × 1 1/2 | 3/8 | 3.99 | 1.17 | ........ | ........ | ........ | ........ | 0.88 | 15.8 |
| | 1/4 | 2.77 | 0.81 | ........ | ........ | ........ | ........ | 0.62 | 11.2 |
| | 1/8 | 1.44 | 0.42 | ........ | ........ | ........ | ........ | 0.32 | 5.8 |

* 1 Kip = 1000 lb.

## Table 16.—Radii of Gyration for Two Angles

(Carnegie-Illinois Steel Corp., Pittsburgh)

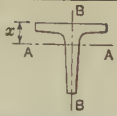
Values of radii of gyration for intermediate thicknesses may be obtained by interpolation



| One Angle | | | Two Angles | Radius of Gyration of Two Angles | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Equal Angles. Unequal Angles, Long Legs Vertical | | | | | | Unequal Angles, Short Legs Vertical | | | | | |
| | | | | | Axis YY | | | | | | Axis YY | | | | |
| Size, in. | Thick-ness, in. | Weight per Foot, lb. | Area, sq. in. | Axis XX | In Contact | 1/4 in. Apart | 3/8 in. Apart | 1/2 in. Apart | 3/4 in. Apart | Axis XX | In Contact | 1/4 in. Apart | 3/8 in. Apart | 1/2 in. Apart | 3/4 in. Apart |
| 8 × 8 | 1 1/8 | 56.9 | 33.46 | 2.42 | 3.42 | 3.51 | 3.55 | 3.60 | 3.69 | .... | .... | .... | .... | .... | .... |
| | 13/16 | 42.0 | 24.68 | 2.46 | 3.37 | 3.46 | 3.50 | 3.55 | 3.64 | .... | .... | .... | .... | .... | .... |
| | 1/2 | 26.4 | 15.50 | 2.50 | 3.33 | 3.41 | 3.45 | 3.50 | 3.59 | .... | .... | .... | .... | .... | .... |
| 8 × 6 | 1 | 44.2 | 26.00 | 2.49 | 2.39 | 2.48 | 2.52 | 2.57 | 2.66 | 1.73 | 3.64 | 3.73 | 3.78 | 3.83 | 3.92 |
| | 3/4 | 33.8 | 19.88 | 2.53 | 2.35 | 2.44 | 2.48 | 2.52 | 2.61 | 1.76 | 3.60 | 3.69 | 3.73 | 3.78 | 3.87 |
| | 7/16 | 20.2 | 11.86 | 2.57 | 2.31 | 2.39 | 2.43 | 2.48 | 2.56 | 1.80 | 3.55 | 3.64 | 3.68 | 3.73 | 3.82 |
| 8 × 4 | 1 | 37.4 | 22.00 | 2.52 | 1.47 | 1.56 | 1.61 | 1.66 | 1.76 | 1.03 | 3.95 | 4.05 | 4.10 | 4.15 | 4.25 |
| | 3/4 | 28.7 | 16.88 | 2.55 | 1.42 | 1.51 | 1.55 | 1.60 | 1.69 | 1.06 | 3.90 | 4.00 | 4.04 | 4.09 | 4.19 |
| | 7/16 | 17.2 | 10.12 | 2.60 | 1.37 | 1.45 | 1.49 | 1.53 | 1.62 | 1.09 | 3.84 | 3.93 | 3.98 | 4.03 | 4.12 |
| 7 × 4 | 1 | 34.0 | 20.00 | 2.18 | 1.53 | 1.62 | 1.67 | 1.72 | 1.82 | 1.06 | 3.40 | 3.49 | 3.54 | 3.59 | 3.69 |
| | 11/16 | 24.2 | 14.19 | 2.23 | 1.48 | 1.56 | 1.61 | 1.65 | 1.75 | 1.09 | 3.34 | 3.43 | 3.48 | 3.53 | 3.63 |
| | 3/8 | 13.6 | 7.98 | 2.27 | 1.43 | 1.51 | 1.55 | 1.59 | 1.68 | 1.13 | 3.28 | 3.37 | 3.42 | 3.47 | 3.56 |
| 6 × 6 | 1 | 37.4 | 22.00 | 1.80 | 2.59 | 2.68 | 2.72 | 2.77 | 2.87 | .... | .... | .... | .... | .... | .... |
| | 11/16 | 26.5 | 15.56 | 1.83 | 2.54 | 2.63 | 2.67 | 2.71 | 2.81 | .... | .... | .... | .... | .... | .... |
| | 3/8 | 14.9 | 8.72 | 1.88 | 2.49 | 2.58 | 2.62 | 2.66 | 2.75 | .... | .... | .... | .... | .... | .... |
| 6 × 4 | 1 | 30.6 | 18.00 | 1.85 | 1.60 | 1.69 | 1.74 | 1.79 | 1.89 | 1.09 | 2.85 | 2.95 | 2.99 | 3.04 | 3.14 |
| | 11/16 | 21.8 | 12.80 | 1.89 | 1.55 | 1.63 | 1.68 | 1.73 | 1.82 | 1.13 | 2.79 | 2.89 | 2.93 | 2.98 | 3.08 |
| | 3/8 | 12.3 | 7.22 | 1.93 | 1.50 | 1.58 | 1.62 | 1.67 | 1.76 | 1.17 | 2.74 | 2.83 | 2.87 | 2.92 | 3.02 |
| 6 × 3 1/2 | 1 | 28.9 | 17.00 | 1.85 | 1.37 | 1.47 | 1.51 | 1.56 | 1.66 | 0.92 | 2.92 | 3.02 | 3.07 | 3.12 | 3.22 |
| | 11/16 | 20.6 | 12.12 | 1.89 | 1.31 | 1.41 | 1.45 | 1.49 | 1.60 | 0.95 | 2.87 | 2.96 | 3.01 | 3.06 | 3.16 |
| | 5/16 | 9.8 | 5.74 | 1.95 | 1.25 | 1.33 | 1.37 | 1.42 | 1.50 | 1.00 | 2.81 | 2.90 | 2.95 | 3.00 | 3.09 |
| 5 × 5 | 1 | 30.6 | 18.00 | 1.48 | 2.19 | 2.28 | 2.33 | 2.38 | 2.47 | .... | .... | .... | .... | .... | .... |
| | 11/16 | 21.8 | 12.80 | 1.51 | 2.13 | 2.22 | 2.26 | 2.31 | 2.40 | .... | .... | .... | .... | .... | .... |
| | 3/8 | 12.3 | 7.22 | 1.56 | 2.09 | 2.17 | 2.21 | 2.26 | 2.35 | .... | .... | .... | .... | .... | .... |
| 5 × 3 1/2 | 7/8 | 22.7 | 13.34 | 1.53 | 1.42 | 1.51 | 1.56 | 1.61 | 1.71 | 0.96 | 2.36 | 2.45 | 2.50 | 2.55 | 2.65 |
| | 5/16 | 8.7 | 5.12 | 1.61 | 1.33 | 1.41 | 1.45 | 1.50 | 1.59 | 1.03 | 2.26 | 2.35 | 2.39 | 2.44 | 2.54 |
| 5 × 3 | 13/16 | 19.9 | 11.68 | 1.55 | 1.18 | 1.27 | 1.32 | 1.37 | 1.47 | 0.80 | 2.42 | 2.52 | 2.57 | 2.62 | 2.72 |
| | 5/16 | 8.2 | 4.80 | 1.61 | 1.09 | 1.17 | 1.22 | 1.26 | 1.35 | 0.85 | 2.33 | 2.42 | 2.47 | 2.52 | 2.61 |
| 4 × 4 | 13/16 | 19.9 | 11.68 | 1.18 | 1.75 | 1.85 | 1.89 | 1.94 | 2.04 | .... | .... | .... | .... | .... | .... |
| | 1/4 | 6.6 | 3.88 | 1.25 | 1.66 | 1.75 | 1.79 | 1.84 | 1.93 | .... | .... | .... | .... | .... | .... |
| 4 × 3 1/2 | 13/16 | 18.5 | 10.86 | 1.19 | 1.50 | 1.59 | 1.64 | 1.69 | 1.79 | 1.01 | 1.81 | 1.91 | 1.96 | 2.01 | 2.11 |
| | 5/16 | 7.7 | 4.50 | 1.26 | 1.42 | 1.51 | 1.55 | 1.60 | 1.69 | 1.07 | 1.73 | 1.81 | 1.86 | 1.91 | 2.00 |
| 4 × 3 | 13/16 | 17.1 | 10.06 | 1.21 | 1.25 | 1.35 | 1.40 | 1.45 | 1.55 | 0.83 | 1.88 | 1.98 | 2.03 | 2.08 | 2.18 |
| | 1/4 | 5.8 | 3.38 | 1.28 | 1.16 | 1.24 | 1.28 | 1.33 | 1.43 | 0.89 | 1.78 | 1.87 | 1.92 | 1.96 | 2.06 |
| 3 1/2 × 3 1/2 | 13/16 | 17.1 | 10.06 | 1.02 | 1.55 | 1.65 | 1.70 | 1.75 | 1.85 | .... | .... | .... | .... | .... | .... |
| | 1/4 | 5.8 | 3.38 | 1.09 | 1.46 | 1.55 | 1.59 | 1.64 | 1.73 | .... | .... | .... | .... | .... | .... |
| 3 1/2 × 3 | 13/16 | 15.8 | 9.24 | 1.04 | 1.30 | 1.40 | 1.45 | 1.50 | 1.60 | 0.85 | 1.61 | 1.71 | 1.76 | 1.81 | 1.91 |
| | 1/4 | 5.4 | 3.12 | 1.11 | 1.20 | 1.29 | 1.34 | 1.38 | 1.48 | 0.91 | 1.52 | 1.61 | 1.65 | 1.70 | 1.80 |
| 3 1/2 × 2 1/2 | 11/16 | 12.5 | 7.30 | 1.06 | 1.03 | 1.13 | 1.18 | 1.23 | 1.33 | 0.69 | 1.66 | 1.75 | 1.80 | 1.86 | 1.96 |
| | 1/4 | 4.9 | 2.88 | 1.12 | 0.95 | 1.04 | 1.09 | 1.13 | 1.23 | 0.74 | 1.58 | 1.67 | 1.71 | 1.76 | 1.86 |
| 3 × 3 | 5/8 | 11.5 | 6.72 | 0.88 | 1.32 | 1.41 | 1.46 | 1.51 | 1.61 | .... | .... | .... | .... | .... | .... |
| | 1/4 | 4.9 | 2.88 | 0.93 | 1.25 | 1.34 | 1.38 | 1.43 | 1.53 | .... | .... | .... | .... | .... | .... |
| 3 × 2 1/2 | 9/16 | 9.5 | 6.56 | 0.91 | 1.05 | 1.15 | 1.20 | 1.25 | 1.35 | 0.72 | 1.37 | 1.46 | 1.51 | 1.56 | 1.66 |
| | 1/4 | 4.5 | 2.64 | 0.95 | 1.00 | 1.09 | 1.13 | 1.18 | 1.28 | 0.75 | 1.31 | 1.40 | 1.45 | 1.50 | 1.59 |
| 3 × 2 | 1/2 | 7.7 | 4.50 | 0.92 | 0.80 | 0.89 | 0.94 | 1.00 | 1.10 | 0.55 | 1.42 | 1.52 | 1.57 | 1.62 | 1.72 |
| | 1/4 | 4.1 | 2.38 | 0.95 | 0.74 | 0.84 | 0.88 | 0.93 | 1.03 | 0.57 | 1.38 | 1.47 | 1.52 | 1.57 | 1.67 |
| 2 1/2 × 2 1/2 | 1/2 | 7.7 | 4.50 | 0.74 | 1.09 | 1.19 | 1.24 | 1.29 | 1.39 | .... | .... | .... | .... | .... | .... |
| | 1/4 | 4.1 | 2.38 | 0.77 | 1.05 | 1.14 | 1.19 | 1.24 | 1.34 | .... | .... | .... | .... | .... | .... |
| 2 1/2 × 2 | 1/2 | 6.8 | 4.00 | 0.75 | 0.84 | 0.94 | 0.99 | 1.04 | 1.15 | 0.56 | 1.15 | 1.25 | 1.30 | 1.35 | 1.46 |
| | 1/4 | 3.6 | 2.12 | 0.78 | 0.80 | 0.89 | 0.93 | 0.98 | 1.08 | 0.59 | 1.11 | 1.20 | 1.25 | 1.30 | 1.40 |
| 2 × 2 | 7/16 | 5.3 | 3.12 | 0.59 | 0.88 | 0.98 | 1.03 | 1.08 | 1.19 | .... | .... | .... | .... | .... | .... |
| | 1/4 | 3.2 | 1.88 | 0.61 | 0.85 | 0.94 | 0.99 | 1.04 | 1.14 | .... | .... | .... | .... | .... | .... |

### Table 17.—Properties of Tees, Elements of Sections

(Carnegie-Illinois Steel Corporation, Pittsburgh)



| Size, in. | | | | Weight per ft., lb. | Area of Section, sq. in. | Axis A–A | | | | Axis B–B | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Flange | Stem | Thickness Toe * | Thickness Root * | | | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | $x$ | Moment of Inertia, $I'$ in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r'$, in. |
| Equal Tees | | | | | | | | | | | | |
| 6 1/2 | 6 1/2 | 0.40, .45 | 0.55, .45 | 19.8 | 5.80 | 23.5 | 5.0 | 2.01 | 1.76 | 10.1 | 3.1 | 1.32 |
| 4 | 4 | 1/2 | 9/16 | 13.5 | 3.97 | 5.7 | 2.0 | 1.20 | 1.18 | 2.8 | 1.4 | 0.84 |
| 4 | 4 | 3/8 | 7/16 | 10.5 | 3.09 | 4.5 | 1.6 | 1.21 | 1.13 | 2.1 | 1.1 | 0.83 |
| 3 1/2 | 3 1/2 | 1/2 | 9/16 | 11.7 | 3.44 | 3.7 | 1.5 | 1.04 | 1.05 | 1.9 | 1.1 | 0.74 |
| 3 1/2 | 3 1/2 | 3/8 | 7/16 | 9.2 | 2.68 | 3.0 | 1.2 | 1.05 | 1.01 | 1.4 | 0.81 | 0.73 |
| 3 | 3 | 3/8 | 7/16 | 7.8 | 2.27 | 1.8 | 0.86 | 0.90 | 0.88 | 0.90 | 0.60 | 0.63 |
| 3 | 3 | 5/16 | 3/8 | 6.7 | 1.95 | 1.6 | 0.74 | 0.90 | 0.86 | 0.75 | 0.50 | 0.62 |
| 2 1/2 | 2 1/2 | 3/8 | 7/16 | 6.4 | 1.87 | 1.0 | 0.59 | 0.74 | 0.76 | 0.52 | 0.42 | 0.53 |
| 2 1/2 | 2 1/2 | 5/16 | 3/8 | 5.5 | 1.60 | 0.88 | 0.50 | 0.74 | 0.74 | 0.44 | 0.35 | 0.52 |
| 2 1/4 | 2 1/4 | 5/16 | 3/8 | 4.9 | 1.43 | 0.65 | 0.41 | 0.67 | 0.68 | 0.33 | 0.29 | 0.48 |
| 2 1/4 | 2 1/4 | 1/4 | 5/16 | 4.1 | 1.19 | 0.52 | 0.32 | 0.66 | 0.65 | 0.25 | 0.22 | 0.46 |
| 2 | 2 | 5/16 | 3/8 | 4.3 | 1.26 | 0.44 | 0.31 | 0.59 | 0.61 | 0.23 | 0.23 | 0.43 |
| 2 | 2 | 1/4 | 5/16 | 3.56 | 1.05 | 0.37 | 0.26 | 0.59 | 0.59 | 0.18 | 0.18 | 0.42 |
| Unequal Tees | | | | | | | | | | | | |
| 5 | 3 1/8 | 1/2, 13/32 | 9/16, 5/8 | 13.6 | 4.00 | 2.7 | 1.1 | 0.82 | 0.76 | 5.2 | 2.1 | 1.14 |
| 5 | 3 | 3/8, 13/32 | 7/16, 5/8 | 11.5 | 3.37 | 2.4 | 1.1 | 0.84 | 0.76 | 3.9 | 1.6 | 1.10 |
| 4 | 5 | 1/2 | 9/16 | 15.3 | 4.50 | 10.8 | 3.1 | 1.55 | 1.56 | 2.8 | 1.4 | 0.79 |
| 4 | 5 | 3/8 | 7/16 | 11.9 | 3.49 | 8.5 | 2.4 | 1.56 | 1.51 | 2.1 | 1.1 | 0.78 |
| 4 | 4 1/2 | 1/2 | 9/16 | 14.4 | 4.23 | 7.9 | 2.5 | 1.37 | 1.37 | 2.8 | 1.4 | 0.81 |
| 4 | 4 1/2 | 3/8 | 7/16 | 11.2 | 3.29 | 6.3 | 2.0 | 1.39 | 1.31 | 2.1 | 1.1 | 0.80 |
| 4 | 3 | 3/8 | 7/16 | 9.2 | 2.68 | 2.0 | 0.90 | 0.86 | 0.78 | 2.1 | 1.1 | 0.89 |
| 4 | 3 | 5/16 | 3/8 | 7.8 | 2.29 | 1.7 | 0.77 | 0.87 | 0.75 | 1.8 | 0.88 | 0.88 |
| 4 | 2 1/2 | 3/8 | 7/16 | 8.5 | 2.48 | 1.2 | 0.62 | 0.69 | 0.62 | 2.1 | 1.0 | 0.92 |
| 4 | 2 1/2 | 5/16 | 3/8 | 7.2 | 2.12 | 1.0 | 0.53 | 0.69 | 0.60 | 1.8 | 0.88 | 0.91 |
| 3 | 2 1/2 | 5/16 | 3/8 | 6.1 | 1.77 | 0.94 | 0.52 | 0.73 | 0.68 | 0.75 | 0.50 | 0.65 |
| 2 1/2 | 3 | 5/16 | 3/8 | 6.1 | 1.77 | 1.5 | 0.72 | 0.92 | 0.92 | 0.44 | 0.35 | 0.50 |

* Where two dimensions are shown, the first is for the flange, the second for the stem.

### Table 18.—Safe Uniform Loads on Tees

(Carnegie-Illinois Steel Corporation, Pittsburgh)

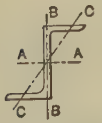
Neutral axis parallel to flanges. Values applicable only when sections are rigidly secured against lateral deflection. Maximum bending stress, 18,000 lb. per sq. in.

| Equal Tees | | | | | Unequal Tees | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Size, Flange × Stem, in. | Weight per Foot, lb. | 1 Foot Span | Maximum Span 360 × Deflection | | Size, Flange × Stem, in. | Weight per Foot, lb. | 1 Foot Span | Maximum Span 360 × Deflection | |
| | | Safe Load, Kips * | Safe Load, Kips * | Length, ft. | | | Safe Load, Kips * | Safe Load, Kips * | Length, ft. |
| 6 1/2 × 6 1/2 | 19.8 | 59.40 | 3.50 | 17.0 | 5 × 3 1/8 | 13.6 | 13.55 | 1.60 | 8.5 |
| 4 × 4 | 13.5 | 24.24 | 2.40 | 10.1 | 5 × 3 | 11.5 | 12.75 | 1.59 | 8.0 |
| 4 × 4 | 10.5 | 18.96 | 1.85 | 10.3 | 4 × 5 | 15.3 | 37.56 | 3.05 | 12.3 |
| 3 1/2 × 3 1/2 | 11.7 | 18.0 | 2.05 | 8.8 | 4 × 5 | 11.9 | 29.16 | 2.33 | 12.5 |
| 3 1/2 × 3 1/2 | 9.2 | 14.4 | 1.62 | 8.9 | 4 × 4 1/2 | 14.4 | 30.48 | 2.72 | 11.2 |
| 3 × 3 | 7.8 | 10.32 | 1.36 | 7.6 | 4 × 4 1/2 | 11.2 | 23.76 | 2.08 | 11.4 |
| 3 × 3 | 6.7 | 8.88 | 1.16 | 7.7 | 4 × 3 | 9.2 | 10.80 | 1.36 | 7.9 |
| 2 1/2 × 2 1/2 | 6.4 | 7.08 | 1.14 | 6.2 | 4 × 3 | 7.8 | 9.24 | 1.15 | 8.1 |
| 2 1/2 × 2 1/2 | 5.5 | 6.00 | 0.95 | 6.3 | 4 × 2 1/2 | 8.5 | 7.44 | 1.11 | 6.7 |
| 2 1/4 × 2 1/4 | 4.9 | 4.92 | 0.88 | 5.6 | 4 × 2 1/2 | 7.2 | 6.36 | 0.94 | 6.8 |
| 2 1/4 × 2 1/4 | 4.1 | 3.84 | 0.67 | 5.7 | 3 × 2 1/2 | 6.1 | 6.24 | 0.96 | 6.5 |
| 2 × 2 | 4.3 | 3.72 | 0.75 | 5.0 | 2 1/2 × 3 | 6.1 | 8.64 | 1.16 | 7.4 |
| 2 × 2 | 3.56 | 3.12 | 0.62 | 5.0 | ............ | ........ | ........ | ........ | ....... |

*1 Kip = 1000 lb.

## Table 19.—Properties of Zees, Elements of Sections

(Carnegie-Illinois Steel Corporation, Pittsburgh)



| Depth, in. | Flange, in. | Thickness, in. | Weight per ft., lb. | Area of Section, sq. in. | Axis A–A | | | Axis B–B | | | Axis C–C |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | Moment of Inertia, $I'$, in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r'$, in. | Least Radius of Gyration, $r''$, in. |
| 6 1/8 | 3 5/8 | 7/8 | 34.6 | 10.17 | 50.2 | 16.4 | 2.22 | 19.2 | 6.0 | 1.37 | 0.83 |
| 6 1/16 | 3 9/16 | 13/16 | 32.0 | 9.40 | 46.1 | 15.2 | 2.22 | 17.3 | 5.5 | 1.36 | 0.82 |
| 6 | 3 1/2 | 3/4 | 29.4 | 8.63 | 42.1 | 14.0 | 2.21 | 15.4 | 4.9 | 1.34 | 0.81 |
| 6 1/8 | 3 5/8 | 11/16 | 28.1 | 8.25 | 43.2 | 14.1 | 2.29 | 16.3 | 5.0 | 1.41 | 0.84 |
| 6 1/16 | 3 9/16 | 5/8 | 25.4 | 7.46 | 38.9 | 12.8 | 2.28 | 14.4 | 4.4 | 1.39 | 0.82 |
| 6 | 3 1/2 | 9/16 | 22.8 | 6.68 | 34.6 | 11.5 | 2.28 | 12.6 | 3.9 | 1.37 | 0.81 |
| 6 1/8 | 3 5/8 | 1/2 | 21.1 | 6.19 | 34.4 | 11.2 | 2.36 | 12.9 | 3.8 | 1.44 | 0.84 |
| 6 1/16 | 3 9/16 | 7/16 | 18.4 | 5.39 | 29.8 | 9.8 | 2.35 | 11.0 | 3.3 | 1.43 | 0.83 |
| 6 | 3 1/2 | 3/8 | 15.7 | 4.59 | 25.3 | 8.4 | 2.35 | 9.1 | 2.8 | 1.41 | 0.83 |
| 5 1/8 | 3 3/8 | 13/16 | 28.4 | 8.33 | 28.7 | 11.2 | 1.86 | 14.4 | 4.8 | 1.31 | 0.76 |
| 5 1/16 | 3 5/16 | 3/4 | 26.0 | 7.64 | 26.2 | 10.3 | 1.85 | 12.8 | 4.4 | 1.30 | 0.74 |
| 5 | 3 1/4 | 11/16 | 23.7 | 6.96 | 23.7 | 9.5 | 1.84 | 11.4 | 3.9 | 1.28 | 0.73 |
| 5 1/8 | 3 3/8 | 5/8 | 22.6 | 6.64 | 24.5 | 9.6 | 1.92 | 12.1 | 3.9 | 1.35 | 0.76 |
| 5 1/16 | 3 5/16 | 9/16 | 20.2 | 5.94 | 21.8 | 8.6 | 1.91 | 10.5 | 3.5 | 1.33 | 0.75 |
| 5 | 3 1/4 | 1/2 | 17.9 | 5.25 | 19.2 | 7.7 | 1.91 | 9.1 | 3.0 | 1.31 | 0.74 |
| 5 1/8 | 3 3/8 | 7/16 | 16.4 | 4.81 | 19.1 | 7.4 | 1.99 | 9.2 | 2.9 | 1.38 | 0.77 |
| 5 1/16 | 3 5/16 | 3/8 | 14.0 | 4.10 | 16.2 | 6.4 | 1.99 | 7.7 | 2.5 | 1.37 | 0.76 |
| 5 | 3 1/4 | 5/16 | 11.6 | 3.40 | 13.4 | 5.3 | 1.98 | 6.2 | 2.0 | 1.35 | 0.75 |
| 4 1/8 | 3 3/16 | 3/4 | 23.0 | 6.75 | 15.0 | 7.3 | 1.49 | 11.2 | 4.0 | 1.29 | 0.68 |
| 4 1/16 | 3 1/8 | 11/16 | 20.9 | 6.14 | 13.5 | 6.7 | 1.48 | 10.0 | 3.6 | 1.27 | 0.67 |
| 4 | 3 1/16 | 5/8 | 18.9 | 5.55 | 12.1 | 6.1 | 1.48 | 8.7 | 3.2 | 1.25 | 0.66 |
| 4 1/8 | 3 3/16 | 9/16 | 18.0 | 5.27 | 12.7 | 6.2 | 1.55 | 9.3 | 3.2 | 1.33 | 0.68 |
| 4 1/16 | 3 1/8 | 1/2 | 15.9 | 4.66 | 11.2 | 5.5 | 1.55 | 8.0 | 2.8 | 1.31 | 0.67 |
| 4 | 3 1/16 | 7/16 | 13.8 | 4.05 | 9.7 | 4.8 | 1.55 | 6.7 | 2.4 | 1.29 | 0.66 |
| 4 1/8 | 3 3/16 | 3/8 | 12.5 | 3.66 | 9.6 | 4.7 | 1.62 | 6.8 | 2.3 | 1.36 | 0.69 |
| 4 1/16 | 3 1/8 | 5/16 | 10.3 | 3.03 | 7.9 | 3.9 | 1.62 | 5.5 | 1.8 | 1.34 | 0.68 |
| 4 | 3 1/16 | 1/4 | 8.2 | 2.41 | 6.3 | 3.1 | 1.62 | 4.2 | 1.4 | 1.33 | 0.67 |
| 3 1/16 | 2 3/4 | 9/16 | 14.3 | 4.18 | 5.3 | 3.4 | 1.12 | 5.7 | 2.3 | 1.17 | 0.54 |
| 3 | 2 11/16 | 1/2 | 12.6 | 3.69 | 4.6 | 3.1 | 1.12 | 4.9 | 2.0 | 1.15 | 0.53 |
| 3 1/16 | 2 3/4 | 7/16 | 11.5 | 3.36 | 4.6 | 3.0 | 1.17 | 4.8 | 1.9 | 1.19 | 0.55 |
| 3 | 2 11/16 | 3/8 | 9.8 | 2.86 | 3.9 | 2.6 | 1.16 | 3.9 | 1.6 | 1.17 | 0.54 |
| 3 1/16 | 2 3/4 | 5/16 | 8.5 | 2.48 | 3.6 | 2.4 | 1.21 | 3.6 | 1.4 | 1.21 | 0.56 |
| 3 | 2 11/16 | 1/4 | 6.7 | 1.97 | 2.9 | 1.9 | 1.21 | 2.8 | 1.1 | 1.19 | 0.55 |

## Table 20.—Safe Uniform Loads on Zees*

| Size, Depth × Flange, in. | Thickness, in. | 1 Foot Span | Maximum Span 360 × Deflection | | Size, Depth × Flange, in. | Thickness, in. | 1 Foot Span | Maximum Span 360 × Deflection | |
|---|---|---|---|---|---|---|---|---|---|
| | | Safe Load, Kips † | Safe Load, Kips † | Length, ft. | | | Safe Load, Kips † | Safe Load, Kips † | Length, ft. |
| 6 1/8 × 3 5/8 | 7/8 | 196.80 | 17.95 | 11.0 | 5 × 3 1/4 | 5/16 | 64.08 | 7.16 | 9.0 |
| 6 1/16 × 3 9/16 | 13/16 | 182.64 | 16.83 | 10.9 | 4 1/8 × 3 3/16 | 3/4 | 87.13 | 11.80 | 7.4 |
| 6 × 3 1/2 | 3/4 | 168.48 | 15.68 | 10.7 | 4 1/16 × 3 1/8 | 11/16 | 79.80 | 10.97 | 7.3 |
| 6 1/8 × 3 5/8 | 11/16 | 169.20 | 15.43 | 11.0 | 4 × 3 1/16 | 5/8 | 72.60 | 10.14 | 7.2 |
| 6 1/16 × 3 9/16 | 5/8 | 153.84 | 14.18 | 10.9 | 4 1/8 × 3 3/16 | 9/16 | 74.16 | 10.04 | 7.4 |
| 6 × 3 1/2 | 9/16 | 138.60 | 12.91 | 10.7 | 4 1/16 × 3 1/8 | 1/2 | 66.00 | 9.08 | 7.3 |
| 6 1/8 × 3 5/8 | 1/2 | 134.64 | 12.28 | 11.0 | 4 × 3 1/16 | 7/16 | 57.96 | 8.10 | 7.2 |
| 6 1/16 × 3 9/16 | 7/16 | 117.96 | 10.87 | 10.9 | 4 1/8 × 3 3/16 | 3/8 | 56.04 | 7.59 | 7.4 |
| 6 × 3 1/2 | 3/8 | 101.28 | 9.43 | 10.7 | 4 1/16 × 3 1/8 | 5/16 | 46.92 | 6.45 | 7.3 |
| 5 1/8 × 3 3/8 | 13/16 | 134.40 | 14.65 | 9.2 | 4 × 3 1/16 | 1/4 | 37.68 | 5.26 | 7.2 |
| 5 1/16 × 3 5/16 | 3/4 | 124.08 | 13.69 | 9.1 | 3 1/16 × 2 3/4 | 9/16 | 41.16 | 7.51 | 5.5 |
| 5 × 3 1/4 | 11/16 | 113.64 | 12.70 | 9.0 | 3 × 2 11/16 | 1/2 | 36.72 | 6.84 | 5.4 |
| 5 1/8 × 3 3/8 | 5/8 | 114.84 | 12.52 | 9.2 | 3 1/16 × 2 3/4 | 7/16 | 35.76 | 6.52 | 5.5 |
| 5 1/16 × 3 5/16 | 9/16 | 103.44 | 11.42 | 9.1 | 3 × 2 11/16 | 3/8 | 30.84 | 5.74 | 5.4 |
| 5 × 3 1/4 | 1/2 | 92.16 | 10.30 | 9.0 | 3 1/16 × 2 3/4 | 5/16 | 28.56 | 5.21 | 5.5 |
| 5 1/8 × 3 3/8 | 7/16 | 89.28 | 9.73 | 9.2 | 3 × 2 11/16 | 1/4 | 23.04 | 4.29 | 5.4 |
| 5 1/16 × 3 5/16 | 3/8 | 76.68 | 8.46 | 9.1 | .............. | ........ | ........ | ........ | ........ |

* See Note at head of Table 18. † 1 Kip = 1000 lb.

## 4. COLUMNS

**SAFE LOADS.**—The tables of allowable loads on structural sections used as columns are based on the A.I.C.S. formula (revised 1928), which is

$$f = 18{,}000/[1 + \{(1/18{,}000) \times (l/r)^2\}] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

where $f$ = compressive unit stress, lb. per sq. in.; $l$ = unsupported length of column, in.; $r$ = least radius of gyration, in. The maximum unit stress at $l/r = 60$ is taken as 15,000 lb. per sq. in. The unsupported length of a column or main compression member should not exceed $120\,r$. Safe loads within this limit are between the heavy horizontal lines in the tables. Safe loads below the lower horizontal lines are for values of $l/r > 120$, and up to 200. They may be used for bracing and secondary members. The values given in the tables are for direct, central loads acting in the direction of the column axis.

**ECCENTRIC LOADING OF COLUMNS** produces bending stresses that must be taken into consideration. Columns must be so designed that the combined bending and compressive stresses do not exceed the values given by formulas [12] to [22].

Let $A$ = cross-sectional area of section, sq. in.; $d$ = distance of eccentric load from center of column, in.; $f_1$ = unit compressive stress for length $l$ of column, lb. per sq. in.; $f_2$ = unit compressive stress for length $l_2$ of column, lb. per sq. in.; $f_b$ = unit bending stress, lb. per sq. in.; $l$ = length of column, in. = $l_1 + l_2$; $l_1$ = distance, top of column to eccentric load, in.; $l_2$ = distance, bottom of column to eccentric load, in.; $M$ = maximum bending moment due to eccentric load, in.-lb.; $n$ = distance of extreme fiber from center of column, in.; $P_c$ = concentric load, lb.; $P_e$ = eccentric load, lb.; $r$ = radius of gyration of section in plane of bending, in. Then $P_c = Af_1$, and $P_e = Af_2$.

When eccentric load is applied above mid-point of column,

$$A = (P_c/f_1) + (P_e/f_2) + (P_e d\, l_2 n/l f_b r^2) \quad . \quad . \quad [12]; \qquad M = P_e d l_2/l \quad . \quad . \quad . \quad [13]$$

When eccentric load is applied at the mid-point of column,

$$A = (P_c/f_1) + (P_e/f_2) + (P_e dn/2 f_b r^2) \quad . \quad . \quad [14]; \qquad M = P_e d/2 \quad . \quad . \quad . \quad [15]$$

When eccentric load is applied below mid-point of column, the greater value of

$$A = (P_c/f_1) + (P_e d l_1 n/l f_b r^2) \ . \ [16], \text{ or } A = (P_c/f_1)+(P_e/f_2)+(P_e d l_2 n/l f_b r^2)[17]$$

$$M = P_e d l_1/l \quad . \quad . \quad . \quad [18]$$

When eccentric load is applied at top of column,

$$A = \{(P_c + P_e)/f_1\} + (P_e dn/f_b r^2) \quad . \quad . \quad . \quad . \quad [19]; \qquad M = P_e d \quad . \quad . \quad . \quad . \quad [20]$$

When eccentric load is applied at bottom of column,

$$A = (P_c/f_1) + (P_e dn/f_b r^2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [21]; \qquad M = P_e d \quad . \quad . \quad . \quad . \quad [22]$$

**BENDING FACTORS.**—The bending factor offers a convenient means for converting bending moments produced by eccentric loads or other means, such as knee braces, into equivalent direct central loads. If the bending factor $k$ be multiplied by the bending moment, in.-lb., the product is an equivalent central load on the column, producing the same compressive stress as the bending moment. This equivalent load is added to the direct central load to give the total load that the column must carry. The proper size of columns then can be selected from the values of safe loads in the tables, opposite the proper figure for length of column. In formulas [12] to [22], $k$ can be substituted for the factor $(n/r^2)$.

**BASES.**—Steel slabs are preferable to iron or steel castings for column bases. Slabs should be designed to support a given section, when carrying a maximum load stressing the column up to 15,000 lb. per sq. in., with values of $l/r < 60$, with a maximum fiber stress in the slab of 18,000 lb. per sq. in. The commonly allowed intensities of pressure of the slab on good masonry are $w$ = 500, 625 and 750 lb. per sq. in. The size of the slab is computed as follows: $MN = P/w$; $t = p\sqrt{w/6000}$, where $M$ and $N$ = dimensions of slab, in., corresponding to width of flange $B$, and depth of section $D$, respectively; $P$ = total load on column, lb.; $w$ = unit pressure on slab support, lb. per sq. in.; $t$ = thickness of slab, in.; $p$ = projection, in., of slab beyond hypothetical column = $\frac{1}{2}(N - 0.95\,D)$, or $\frac{1}{2}(M - 0.80\,B)$, whichever is greater. Slabs thicker than 2 in. should be planed flat on their upper face to give a square bearing for the column, and on both faces if the slab is to rest on steel. If it is to rest on masonry or concrete, full bearing can be had by grouting with strong cement.

**EXAMPLE IN COLUMN DESIGN.**—The following are examples in the use of the tables for the selection of columns of rolled sections.

**1. Simple Concentric Load.**—Select a column 20 ft. long to carry a concentric load of 400,000 lb. *Solution.*—Refer to Table 21, opposite an effective length of 20 ft.; three columns are found to have

(*Continued on p.* 17–76)

## Table 21.—Allowable Concentric Loads on Wide-flange Sections Used as Columns

(Carnegie-Illinois Steel Corporation, Pittsburgh)

Maximum and minimum weights only of each section. Loads given in kips (1 kip = 1000 lb.). Unit stress based on formula of Am. Inst. of Steel Construction. See p. 17–73.

| Effective Length, ft. | CB 183 18″ × 11 3/4″ | | CB 182 18″ × 8 3/4″ | | CB 181 18″ × 7 1/2″ | | CB 163 16″ × 11 1/2″ | | CB 162 16″ × 8 1/2″ | | CB 161 16″ × 7″ | | CB 146 | CB 145 14″ × 14 1/2″ | | CB 144 14″ × 12″ | | CB 143 14″ × 10″ | | CB 142 14″ × 8″ | | CB 141 14″ × 6 3/4″ | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 124 lb. | 96 lb. | 85 lb. | 64 lb. | 55 lb. | 47 lb. | 114 lb. | 88 lb. | 78 lb. | 58 lb. | 50 lb. | 36 lb. | 14″ × 16″ | 136 lb. | 87 lb. | 84 lb. | 78 lb | 74 lb. | 61 lb. | 58 lb. | 43 lb. | 42 lb. | 30 lb. |
| 7 | 547 | 423 | 375 | 282 | 243 | 207 | 503 | 388 | 344 | 256 | 221 | 159 | See Table 22, p. 17–76 | ..... | ..... | ..... | ..... | 326 | 269 | 256 | 190 | 185 | 132 |
| 8 | 547 | 423 | 375 | 282 | 243 | 205 | 503 | 388 | 344 | 256 | 218 | 153 | | ..... | ..... | ..... | ..... | 326 | 269 | 256 | 190 | 181 | 126 |
| 9 | 547 | 423 | 375 | 282 | 233 | 196 | 503 | 388 | 344 | 256 | 208 | 146 | | ..... | ..... | ..... | ..... | 326 | 269 | 256 | 190 | 173 | 120 |
| 10 | 547 | 423 | 375 | 279 | 223 | 187 | 503 | 388 | 341 | 250 | 198 | 138 | | ..... | ..... | ..... | ..... | 326 | 269 | 253 | 186 | 164 | 113 |
| 11 | 547 | 423 | 362 | 269 | 212 | 178 | 503 | 388 | 329 | 241 | 188 | 131 | | ..... | ..... | ..... | ..... | 326 | 269 | 244 | 179 | 156 | 107 |
| 12 | 547 | 423 | 349 | 258 | 202 | 169 | 503 | 388 | 317 | 231 | 178 | 123 | | ..... | ..... | ..... | ..... | 326 | 269 | 235 | 172 | 148 | 100 |
| 13 | 547 | 423 | 336 | 248 | 192 | 160 | 503 | 388 | 304 | 222 | 169 | 116 | | ..... | ..... | ..... | ..... | 321 | 264 | 225 | 165 | 139 | 94 |
| 14 | 546 | 419 | 323 | 238 | 182 | 151 | 500 | 382 | 292 | 212 | 159 | 109 | | ..... | ..... | ..... | ..... | 312 | 256 | 216 | 158 | 132 | 89 |
| 15 | 532 | 408 | 310 | 228 | 172 | 143 | 488 | 372 | 280 | 203 | 150 | 103 | | 600 | 383 | 371 | 344 | 303 | 248 | 207 | 151 | 124 | 83 |
| 16 | 519 | 397 | 297 | 218 | 163 | 135 | 475 | 362 | 268 | 194 | 142 | 97 | | 600 | 383 | 363 | 336 | 294 | 241 | 198 | 145 | 117 | 78 |
| 17 | 505 | 386 | 285 | 209 | 154 | 127 | 463 | 352 | 257 | 185 | 134 | 91 | | 600 | 383 | 355 | 329 | 285 | 233 | 189 | 138 | 110 | 73 |
| 18 | 491 | 375 | 273 | 200 | 146 | 120 | 450 | 342 | 245 | 177 | 126 | 85 | | 600 | 383 | 346 | 321 | 276 | 225 | 181 | 132 | 104 | 69 |
| 19 | 478 | 365 | 261 | 191 | 138 | 114 | 437 | 331 | 235 | 169 | 119 | 80 | | 598 | 380 | 338 | 313 | 267 | 218 | 173 | 126 | 98 | 65 |
| 20 | 464 | 354 | 250 | 182 | 130 | 107 | 425 | 321 | 224 | 161 | 113 | 76 | | 587 | 373 | 329 | 305 | 258 | 211 | 165 | 120 | 92 | 61 |
| 21 | 450 | 343 | 239 | 174 | 123 | 102 | 412 | 311 | 214 | 154 | 106 | 71 | | 577 | 366 | 321 | 297 | 249 | 203 | 158 | 115 | 87 | 57 |
| 22 | 437 | 333 | 228 | 166 | 117 | 96 | 400 | 302 | 204 | 146 | 101 | 67 | | 566 | 359 | 312 | 289 | 240 | 196 | 151 | 109 | 82 | 54 |
| 23 | 424 | 322 | 218 | 158 | 111 | 91 | 388 | 292 | 195 | 140 | 95 | 63 | | 555 | 351 | 304 | 281 | 232 | 189 | 144 | 104 | 78 | 51 |
| 24 | 411 | 312 | 209 | 151 | 105 | 86 | 376 | 283 | 187 | 133 | 90 | 60 | | 543 | 344 | 295 | 273 | 224 | 183 | 137 | 99 | 74 | ..... |
| 25 | 398 | 302 | 200 | 144 | 100 | 81 | 364 | 274 | 178 | 127 | 85 | ..... | | 532 | 337 | 287 | 265 | 216 | 176 | 131 | 95 | 70 | ..... |
| 26 | 386 | 293 | 191 | 138 | 94 | 77 | 353 | 265 | 170 | 121 | ..... | ..... | | 521 | 330 | 279 | 258 | 208 | 170 | 125 | 91 | ..... | ..... |
| 27 | 374 | 283 | 183 | 132 | ..... | ..... | 342 | 256 | 163 | 116 | ..... | ..... | | 510 | 323 | 271 | 251 | 201 | 164 | 120 | 87 | ..... | ..... |
| 28 | 362 | 274 | 175 | 126 | ..... | ..... | 331 | 248 | 156 | 111 | ..... | ..... | | 499 | 316 | 263 | 243 | 194 | 158 | 114 | 83 | ..... | ..... |
| 29 | 351 | 265 | 168 | 121 | ..... | ..... | 320 | 240 | 149 | 106 | ..... | ..... | | 488 | 309 | 256 | 236 | 187 | 152 | 109 | 79 | ..... | ..... |
| 30 | 340 | 257 | 161 | 115 | ..... | ..... | 310 | 232 | 143 | 101 | ..... | ..... | | 478 | 302 | 249 | 229 | 180 | 147 | 105 | 75 | ..... | ..... |
| 31 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 467 | 295 | 241 | 223 | ..... | ..... | ..... | ..... | ..... | ..... |
| 32 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 456 | 288 | 234 | 216 | ..... | ..... | ..... | ..... | ..... | ..... |
| 33 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 446 | 281 | 228 | 210 | ..... | ..... | ..... | ..... | ..... | ..... |
| 34 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 436 | 275 | 221 | 204 | ..... | ..... | ..... | ..... | ..... | ..... |
| 35 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 426 | 268 | 214 | 198 | ..... | ..... | ..... | ..... | ..... | ..... |
| 36 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 416 | 262 | 208 | 192 | ..... | ..... | ..... | ..... | ..... | ..... |
| 37 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 406 | 256 | 202 | 186 | ..... | ..... | ..... | ..... | ..... | ..... |
| 38 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | | 397 | 250 | 196 | 181 | ..... | ..... | ..... | ..... | ..... | ..... |

Safe loads above the upper zig-zag line are for ratios of $l/r$ not over 60; those between the zig-zag lines are for ratios of $l/r$ up to 120; those below the lower zig-zag line are for ratios of $l/r$ not over 200.

**Table 21.—Allowable Concentric Loads on Wide-flange Sections Used as Columns—*Continued***

| Effective Length, ft. | CB 124 12″ × 12″ | | CB 123 12″ × 10″ | | CB 122 12″ × 8″ | | CB 121 12″ × 6 1/2″ | | CB 103 10″ × 10″ | | CB 102 10″ × 8″ | | CB 101 10″ × 5 3/4″ | | CB 83 8″ × 8″ | | CB 82 8″ × 6 1/2″ | | CB 81 8″ × 5 1/4″ | | CB 61 5 3/4″ × 9 1/2″ | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 190 lb. | 65 lb. | 64 lb. | 53 lb. | 50 lb. | 40 lb. | 36 lb. | 25 lb. | 136 lb. | 49 lb. | 45 lb. | 33 lb. | 29 lb. | 21 lb. | 67 lb. | 31 lb. | 27 lb. | 24 lb. | 21 lb. | 17 lb. | 88 lb. | 40 lb. |
| 5 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 93 | 75 | ...... | ...... |
| 6 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 199 | 146 | 128 | 93 | ...... | ...... | ...... | ...... | 93 | 74 | ...... | ...... |
| 7 | ...... | ...... | ...... | ...... | 221 | 177 | 159 | 111 | ...... | ...... | 199 | 146 | 126 | 89 | ...... | ...... | ...... | ...... | 88 | 70 | ...... | ...... |
| 8 | ...... | ...... | ...... | ...... | 221 | 177 | 155 | 105 | ...... | ...... | 199 | 146 | 120 | 84 | 296 | 137 | 119 | 106 | 83 | 65 | ...... | ...... |
| 9 | ...... | ...... | ...... | ...... | 221 | 177 | 148 | 100 | ...... | ...... | 199 | 146 | 113 | 79 | 296 | 137 | 114 | 102 | 78 | 61 | ...... | ...... |
| 10 | ...... | ...... | ...... | ...... | 219 | 175 | 141 | 94 | ...... | ...... | 199 | 144 | 106 | 74 | 296 | 137 | 109 | 97 | 72 | 56 | ...... | ...... |
| 11 | ...... | ...... | ...... | ...... | 212 | 169 | 133 | 89 | ...... | ...... | 192 | 139 | 100 | 69 | 292 | 132 | 104 | 93 | 67 | 52 | 388 | 176 |
| 12 | ...... | 287 | 282 | 234 | 204 | 162 | 126 | 84 | 600 | 216 | 185 | 134 | 94 | 64 | 282 | 128 | 99 | 88 | 63 | 49 | 388 | 176 |
| 13 | ...... | 287 | 279 | 230 | 196 | 156 | 119 | 79 | 600 | 214 | 178 | 129 | 88 | 60 | 273 | 123 | 94 | 84 | 58 | 45 | 388 | 173 |
| 14 | ...... | 287 | 271 | 224 | 188 | 150 | 112 | 74 | 594 | 209 | 171 | 123 | 82 | 56 | 263 | 118 | 89 | 79 | 54 | 42 | 378 | 168 |
| 15 | 838 | 287 | 264 | 217 | 180 | 143 | 106 | 69 | 579 | 203 | 164 | 118 | 77 | 52 | 253 | 114 | 85 | 75 | 50 | 38 | 368 | 162 |
| 16 | 838 | 281 | 256 | 211 | 173 | 137 | 100 | 65 | 564 | 197 | 158 | 113 | 72 | 48 | 244 | 109 | 80 | 71 | 47 | 36 | 357 | 157 |
| 17 | 825 | 274 | 248 | 204 | 165 | 131 | 94 | 61 | 549 | 191 | 151 | 108 | 67 | 45 | 234 | 104 | 76 | 67 | 44 | 33 | 347 | 152 |
| 18 | 807 | 268 | 240 | 197 | 158 | 126 | 89 | 57 | 534 | 185 | 145 | 104 | 63 | 42 | 225 | 100 | 72 | 64 | 41 | 31 | 337 | 147 |
| 19 | 790 | 261 | 232 | 191 | 151 | 120 | 83 | 54 | 518 | 179 | 138 | 99 | 59 | 39 | 216 | 96 | 68 | 60 | 38 | 29 | 326 | 143 |
| 20 | 772 | 255 | 225 | 185 | 145 | 115 | 79 | 51 | 503 | 173 | 132 | 94 | 55 | 37 | 207 | 92 | 64 | 57 | 35 | ...... | 316 | 138 |
| 21 | 754 | 248 | 217 | 178 | 138 | 109 | 74 | 48 | 488 | 168 | 127 | 90 | 52 | ...... | 199 | 88 | 61 | 54 | ...... | ...... | 306 | 133 |
| 22 | 736 | 242 | 210 | 172 | 132 | 104 | 70 | 45 | 473 | 162 | 121 | 86 | 49 | ...... | 191 | 84 | 58 | 51 | ...... | ...... | 296 | 128 |
| 23 | 718 | 235 | 203 | 166 | 126 | 100 | 66 | 42 | 458 | 156 | 116 | 82 | ...... | ...... | 183 | 80 | 55 | 48 | ...... | ...... | 286 | 124 |
| 24 | 700 | 228 | 196 | 160 | 120 | 95 | 63 | ...... | 444 | 151 | 111 | 79 | ...... | ...... | 175 | 77 | 52 | 46 | ...... | ...... | 277 | 119 |
| 25 | 682 | 222 | 189 | 155 | 115 | 91 | 59 | ...... | 430 | 146 | 106 | 75 | ...... | ...... | 168 | 73 | 49 | 43 | ...... | ...... | 268 | 115 |
| 26 | 665 | 216 | 182 | 149 | 110 | 87 | ...... | ...... | 416 | 141 | 101 | 72 | ...... | ...... | 161 | 70 | 47 | 41 | ...... | ...... | 259 | 111 |
| 27 | 648 | 210 | 176 | 144 | 105 | 83 | ...... | ...... | 403 | 136 | 97 | 69 | ...... | ...... | 154 | 67 | 44 | ...... | ...... | ...... | 250 | 107 |
| 28 | 631 | 204 | 170 | 139 | 101 | 79 | ...... | ...... | 390 | 131 | 93 | 66 | ...... | ...... | 148 | 64 | ...... | ...... | ...... | ...... | 242 | 103 |
| 29 | 614 | 198 | 164 | 134 | 96 | 76 | ...... | ...... | 377 | 127 | 89 | 63 | ...... | ...... | 142 | 62 | ...... | ...... | ...... | ...... | 233 | 99 |
| 30 | 598 | 192 | 158 | 129 | 92 | 73 | ...... | ...... | 365 | 123 | ...... | ...... | ...... | ...... | 136 | 59 | ...... | ...... | ...... | ...... | 225 | 96 |
| 31 | 582 | 187 | 153 | 125 | ...... | ...... | ...... | ...... | 353 | 118 | ...... | ...... | ...... | ...... | 131 | 57 | ...... | ...... | ...... | ...... | 218 | 92 |
| 32 | 567 | 181 | 147 | 120 | ...... | ...... | ...... | ...... | 342 | 114 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 211 | 89 |
| 33 | 551 | 176 | 142 | 116 | ...... | ...... | ...... | ...... | 331 | 110 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 203 | 86 |
| 34 | 536 | 171 | 137 | 112 | ...... | ...... | ...... | ...... | 320 | 107 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 197 | 83 |
| 35 | 522 | 166 | 133 | 108 | ...... | ...... | ...... | ...... | 310 | 103 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 190 | 80 |

Safe loads above the upper zig-zag line are for ratios of $l/r$ not over 60; those between the zig-zag lines are for ratios of $l/r$ up to 120; those below the lower zig-zag line are for ratios of $l/r$ not over 200.

the requisite carrying capacity, *viz.*, CB 124, 12 × 12 in., 65-lb., 255,000 lb.; CB 143, 14 × 10 in., 74-lb., 258,000 lb.; CB 182, 18 × 8 3/4 in., 85-lb., 250,000 lb. Since CB 124 will carry the load at the least cost, it is selected.

**2. Concentric Load and Eccentric Load.**—Select a column 20 ft. long to carry a concentric load of 170,000 lb., together with an eccentric load of 27,500 lb. applied 4 ft. below the top of the column, and acting in the plane of greatest resistance, *i.e.* along axis AA, at a distance of 18 in. from the center of the column. *Solution.*—A preliminary selection is made of a 14 × 10-in., 74-lb. section, CB 143. From Table 3, the following data are derived: Area of section, $A = 21.76$ sq. in.; width of flange, $b = 10.072$ in.; radius of gyration $r$ about axis AA = 6.05; about axis BB = 2.48; distance to extreme fiber $n = 7.09$. From the conditions of the problem, $l = 20 \times 12$; $l_2 = 16 \times 12$; $d = 18$. (See notation for formulas [11] and [12]).

The unit compressive stresses are determined by formula [11] for both the concentric and eccentric loads.

$$f_1 = 18{,}000/[1 + \{(1/18{,}000) \times (20 \times 12/2.48)^2\}] = 11{,}842 \text{ lb. per sq. in.}$$

$$f_2 = 18{,}000/[1 + \{(1/18{,}000) \times (16 \times 12/2.48)^2\}] = 13{,}503 \text{ lb. per sq. in.}$$

Since the load is eccentric and applied above the mid-point of the column, formula [12] is used. Since the ratio $l/b = (20 \times 12)/10.125 = 23.70 > 15$, the unit fiber stress in bending, $f_b = 18{,}000$, must be reduced in accordance with the values of Table 2. Percentage of maximum unit stress corresponding to $l/b = 23.70$ is 86.78. Hence $f_b = 18{,}000 \times 0.8678 = 15{,}620$ lb. per sq. in.

Hence, applying formula [12],

$$A = \frac{170{,}000}{11{,}842} + \frac{27{,}500}{13{,}503} + \frac{27{,}500 \times 18 \times (16 \times 12) \times 7.09}{(20 \times 12) \times 15{,}620 \times (6.05)^2} = 21.32 \text{ sq. in.}$$

Since the area of the column selected is 21.76 sq. in., and an area of only 21.32 sq. in. is required, the selection is satisfactory.

**GRILLAGE.**—When grillage beams are used to support columns in order to distribute heavy loads properly upon soil bearing, the proper size of beams to be used in each tier is determined as follows: Maximum bending moment $M$, in ft.-lb., occurs at the center of the beam. $M = W(L - N)/8$, where $W$ = load supported by each beam in one tier, lb.; $L$ = length of beam, ft.; $N$ = length on which load is applied, ft. If $(L - N)$ be taken as the span, the size of, or safe load on, grillage beams may be obtained from the tables

(*Continued on page* 17–79)

**Table 22.—Allowable Concentric Loads on 14 × 16 in. Wide-flange Columns**

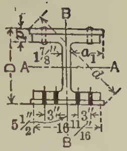
**(Carnegie-Illinois Steel Corporation, Pittsburgh)**

For concentric loads on other sizes, see Table 21; for loads on columns with cover plates, see Table 23

| Effective Length, ft. | 426 lb. | 398 lb. | 370 lb. | 342 lb. | 314 lb. | 287 lb. | 264 lb. | 246 lb. | 228 lb. | 211 lb. | 193 lb. | 176 lb. | 158 lb. | 142 lb. | 320* lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Allowable Load, Kips | | | | | | | | | | | | | | |
| 19 | ...... | ...... | ...... | ...... | ...... | ..... | ..... | ..... | .... | .... | 851 | 776 | 697 | 628 | 1412 |
| 20 | 1879 | 1755 | 1632 | 1509 | 1385 | 1266 | 1164 | 1085 | 1006 | 931 | 851 | 776 | 697 | 626 | 1412 |
| 21 | 1879 | 1755 | 1632 | 1509 | 1385 | 1263 | 1159 | 1078 | 997 | 921 | 840 | 764 | 685 | 615 | 1409 |
| 22 | 1870 | 1743 | 1615 | 1489 | 1362 | 1242 | 1140 | 1060 | 981 | 905 | 826 | 751 | 673 | 605 | 1386 |
| 23 | 1841 | 1715 | 1589 | 1466 | 1340 | 1221 | 1121 | 1042 | 964 | 890 | 812 | 738 | 662 | 594 | 1362 |
| 24 | 1811 | 1687 | 1563 | 1441 | 1317 | 1200 | 1101 | 1024 | 948 | 874 | 797 | 725 | 649 | 583 | 1339 |
| 25 | 1782 | 1659 | 1536 | 1416 | 1295 | 1179 | 1082 | 1006 | 930 | 858 | 782 | 711 | 637 | 572 | 1315 |
| 26 | 1751 | 1631 | 1510 | 1392 | 1271 | 1158 | 1062 | 988 | 913 | 843 | 768 | 698 | 625 | 561 | 1292 |
| 27 | 1721 | 1602 | 1483 | 1367 | 1249 | 1137 | 1042 | 969 | 896 | 826 | 753 | 684 | 613 | 550 | 1269 |
| 28 | 1692 | 1574 | 1457 | 1343 | 1226 | 1116 | 1023 | 950 | 879 | 811 | 739 | 671 | 601 | 539 | 1245 |
| 29 | 1661 | 1546 | 1430 | 1317 | 1202 | 1095 | 1003 | 932 | 862 | 795 | 724 | 657 | 589 | 528 | 1221 |
| 30 | 1631 | 1518 | 1404 | 1293 | 1180 | 1074 | 984 | 914 | 845 | 779 | 710 | 644 | 577 | 517 | 1198 |
| 31 | 1601 | 1489 | 1377 | 1269 | 1157 | 1053 | 964 | 896 | 828 | 763 | 695 | 631 | 565 | 506 | 1175 |
| 32 | 1571 | 1461 | 1351 | 1243 | 1135 | 1032 | 945 | 878 | 811 | 748 | 681 | 618 | 553 | 496 | 1152 |
| 33 | 1542 | 1433 | 1325 | 1220 | 1112 | 1012 | 926 | 861 | 795 | 732 | 667 | 605 | 542 | 485 | 1128 |
| 34 | 1512 | 1405 | 1300 | 1196 | 1090 | 991 | 907 | 843 | 779 | 717 | 653 | 592 | 530 | 475 | 1105 |
| 35 | 1483 | 1379 | 1273 | 1171 | 1068 | 971 | 889 | 825 | 763 | 702 | 639 | 580 | 519 | 464 | 1084 |
| 36 | 1455 | 1352 | 1248 | 1148 | 1046 | 951 | 870 | 808 | 746 | 687 | 626 | 567 | 508 | 454 | 1061 |
| 37 | 1426 | 1325 | 1223 | 1125 | 1025 | 932 | 852 | 791 | 731 | 673 | 612 | 555 | 497 | 444 | 1039 |
| 38 | 1397 | 1298 | 1199 | 1103 | 1004 | 912 | 834 | 775 | 716 | 658 | 599 | 543 | 486 | 435 | 1018 |
| 39 | 1370 | 1272 | 1174 | 1080 | 983 | 894 | 817 | 758 | 700 | 644 | 586 | 531 | 475 | 425 | 997 |
| 40 | 1342 | 1246 | 1151 | 1058 | 963 | 875 | 800 | 742 | 685 | 630 | 574 | 520 | 465 | 416 | 976 |
| 41 | 1315 | 1222 | 1127 | 1036 | 943 | 856 | 783 | 726 | 671 | 617 | 561 | 508 | 454 | 407 | 955 |
| 42 | 1289 | 1197 | 1104 | 1014 | 923 | 838 | 766 | 711 | 656 | 603 | 549 | 497 | 444 | 397 | 935 |
| 43 | 1263 | 1172 | 1081 | 993 | 903 | 820 | 750 | 695 | 642 | 590 | .... | .... | .... | .... | ..... |

* Column core section.

Safe loads above upper zig-zag line are for ratios of $l/r$ not over 60; those between zig-zag lines are for ratios of $l/r$ up to 120; those below lower zig-zag line are for ratios of $l/r$ not over 200.



## Table 23.—Properties and Dimensions of Columns with Cover Plates

(Carnegie-Illinois Steel Corporation, Pittsburgh)

Column made up of wide-flange section CB 146–320 lb. with cover plates riveted to flanges. For allowable concentric loads, see Table 21

| Cover Plate | | Elements | | | | | | | | | | | Column Dimensions | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Axis A–A | | | | Axis B–B | | | | | | | |
| Width, in. | Thickness, in. | Total Depth, in. | Total Weight per ft., lb. | Total Area, $A$ sq. in. | Moment of Inertia, $I$, in.$^4$ | Section Modulus, $S$, in.$^3$ | Radius of Gyration, $r$, in. | Bending Factor, $A/S$ | Moment of Inertia, $I'$, in.$^4$ | Section Modulus, $S'$, in.$^3$ | Radius of Gyration, $r'$, in. | Bending Factor, $A/S'$ | Depth, $D$, in. | Grip, $p$, in. | Diagonal, $d$, in. | Web to Edge, $a_1$ |
| 24 | 3 1/2 | 23.81 | 891 | 262.1 | 21,638 | 1818 | 9.09 | 0.144 | 9699 | 808 | 6.08 | 0.324 | 23 13/16 | 5 9/16 | 33 13/16 | 11 1/16 |
| 24 | 3 3/8 | 23.56 | 871 | 256.1 | 20,797 | 1765 | 9.01 | .145 | 9411 | 784 | 6.06 | .327 | 23 9/16 | 5 7/16 | 33 5/8 | 11 1/16 |
| 24 | 3 1/4 | 23.31 | 850 | 250.1 | 19,973 | 1714 | 8.94 | .146 | 9123 | 760 | 6.04 | .329 | 23 5/16 | 5 5/16 | 33 7/16 | 11 1/16 |
| 24 | 3 1/8 | 23.06 | 830 | 244.1 | 19,166 | 1662 | 8.86 | .147 | 8835 | 736 | 6.02 | .332 | 23 1/16 | 5 3/16 | 33 5/16 | 11 1/16 |
| 24 | 3 | 22.81 | 810 | 238.1 | 18,377 | 1611 | 8.79 | .148 | 8547 | 712 | 5.99 | .334 | 22 13/16 | 5 1/16 | 33 1/8 | 11 1/16 |
| 23 | 3 | 22.81 | 789 | 232.1 | 17,784 | 1559 | 8.75 | .149 | 7719 | 671 | 5.77 | .346 | 22 13/16 | 5 1/16 | 32 3/8 | 10 9/16 |
| 23 | 2 7/8 | 22.56 | 770 | 226.4 | 17,044 | 1511 | 8.68 | .150 | 7465 | 649 | 5.74 | .349 | 22 9/16 | 4 15/16 | 32 3/16 | 10 9/16 |
| 23 | 2 3/4 | 22.31 | 750 | 220.6 | 16,321 | 1463 | 8.60 | .151 | 7212 | 627 | 5.72 | .352 | 22 5/16 | 4 13/16 | 32 1/16 | 10 9/16 |
| 22 | 2 3/4 | 22.31 | 731 | 215.1 | 15,791 | 1416 | 8.57 | .152 | 6515 | 592 | 5.50 | .363 | 22 5/16 | 4 13/16 | 31 5/16 | 10 1/16 |
| 22 | 2 5/8 | 22.06 | 713 | 209.6 | 15,115 | 1370 | 8.49 | .153 | 6294 | 572 | 5.48 | .366 | 22 1/16 | 4 11/16 | 31 1/8 | 10 1/16 |
| 22 | 2 1/2 | 21.81 | 694 | 204.1 | 14,453 | 1325 | 8.41 | .154 | 6072 | 552 | 5.45 | .370 | 21 13/16 | 4 9/16 | 31 | 10 1/16 |
| 22 | 2 3/8 | 21.56 | 675 | 198.6 | 13,806 | 1281 | 8.34 | .155 | 5850 | 532 | 5.43 | .373 | 21 9/16 | 4 7/16 | 30 13/16 | 10 1/16 |
| 22 | 2 1/4 | 21.31 | 657 | 193.1 | 13,175 | 1236 | 8.26 | .156 | 5628 | 512 | 5.40 | .377 | 21 5/16 | 4 5/16 | 30 5/8 | 10 1/16 |
| 22 | 2 1/8 | 21.06 | 638 | 187.6 | 12,558 | 1193 | 8.18 | .157 | 5406 | 491 | 5.37 | .382 | 21 1/16 | 4 3/16 | 30 7/16 | 10 1/16 |
| 22 | 2 | 20.81 | 619 | 182.1 | 11,955 | 1149 | 8.10 | .159 | 5184 | 471 | 5.34 | .387 | 20 13/16 | 4 1/16 | 30 5/16 | 10 1/16 |
| 22 | 1 7/8 | 20.56 | 601 | 176.6 | 11,367 | 1106 | 8.02 | .160 | 4963 | 451 | 5.30 | .391 | 20 9/16 | 3 15/16 | 30 1/8 | 10 1/16 |
| 22 | 1 3/4 | 20.31 | 582 | 171.1 | 10,792 | 1063 | 7.94 | .161 | 4741 | 431 | 5.26 | .397 | 20 5/16 | 3 13/16 | 29 15/16 | 10 1/16 |
| 22 | 1 5/8 | 20.06 | 563 | 165.6 | 10,232 | 1020 | 7.86 | .162 | 4519 | 411 | 5.22 | .403 | 20 1/16 | 3 11/16 | 29 3/4 | 10 1/16 |
| 22 | 1 1/2 | 19.81 | 544 | 160.1 | 9,686 | 978 | 7.78 | .164 | 4297 | 391 | 5.18 | .410 | 19 13/16 | 3 9/16 | 29 5/8 | 10 1/16 |
| 20 | 1 1/2 | 19.81 | 524 | 154.1 | 9,182 | 927 | 7.72 | .166 | 3635 | 364 | 4.86 | .424 | 19 13/16 | 3 9/16 | 28 1/8 | 9 1/16 |
| 20 | 1 3/8 | 19.56 | 507 | 149.1 | 8,697 | 889 | 7.64 | .168 | 3468 | 347 | 4.82 | .430 | 19 9/16 | 3 7/16 | 28 | 9 1/16 |
| 20 | 1 1/4 | 19.31 | 490 | 144.1 | 8,225 | 852 | 7.55 | .169 | 3302 | 330 | 4.79 | .436 | 19 5/16 | 3 5/16 | 27 13/16 | 9 1/16 |
| 18 | 1 1/4 | 19.31 | 473 | 139.1 | 7,817 | 810 | 7.50 | .172 | 2850 | 317 | 4.53 | .439 | 19 5/16 | 3 5/16 | 26 3/8 | 8 1/16 |
| 18 | 1 1/8 | 19.06 | 458 | 134.6 | 7,403 | 777 | 7.42 | .173 | 2729 | 303 | 4.50 | .444 | 19 1/16 | 3 3/16 | 26 3/16 | 8 1/16 |
| 18 | 1 | 18.81 | 442 | 130.1 | 6,999 | 744 | 7.33 | .175 | 2607 | 290 | 4.48 | .449 | 18 13/16 | 3 1/16 | 26 1/16 | 8 1/16 |

## Table 24.—Allowable Concentric Loads on Columns with Cover Plates

(Carnegie-Illinois Steel Corporation, Pittsburgh)

Columns made up of wide-flange section CB 146–320 lb. (see Table 3) with cover plates riveted to flanges. For dimensions and properties of column, see Table 23

Unit stresses based on formula of Am. Inst. of Steel Construction; see p. 17–73

| Effective Length, ft. | Cover Plate, 24″ Wide | | | | | Cover Plate, 23″ Wide | | | Cover Plate, 22″ Wide | | | | | | | | | | | Cover Plate, 20″ Wide | | | Cover Plate, 18″ Wide | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Thickness, in. | | | | | Thickness, in. | | | Thickness, in. | | | | | | | | | | | Thickness, in. | | | Thickness, in. | | |
| | 3 1/2 | 3 3/8 | 3 1/4 | 3 1/8 | 3 | 3 | 2 7/8 | 2 3/4 | 2 3/4 | 2 5/8 | 2 1/2 | 2 3/8 | 2 1/4 | 2 1/8 | 2 | 1 7/8 | 1 3/4 | 1 5/8 | 1 1/2 | 1 1/2 | 1 3/8 | 1 1/4 | 1 1/4 | 1 1/8 | 1 |
| | Load, Kips* | | | | | Load, Kips* | | | Load, Kips* | | | | | | | | | | | Load, Kips* | | | Load, Kips* | | |
| 22 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2087 | 2019 | 1952 |
| 23 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2162 | 2076 | 2004 | 1934 |
| 24 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2312 | 2237 | 2160 | 2045 | 1974 | 1905 |
| 25 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2402 | 2289 | 2209 | 2130 | 2014 | 1943 | 1875 |
| 26 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2814 | 2732 | 2649 | 2567 | 2484 | 2399 | 2257 | 2177 | 2099 | 1982 | 1912 | 1845 |
| 27 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 3227 | 3144 | 3062 | 2979 | 2897 | 2809 | 2722 | 2633 | 2544 | 2456 | 2368 | 2225 | 2146 | 2068 | 1950 | 1881 | 1815 |
| 28 | ..... | ..... | ..... | ..... | ..... | 3482 | 3396 | 3309 | 3207 | 3121 | 3034 | 2948 | 2861 | 2774 | 2687 | 2599 | 2511 | 2423 | 2336 | 2192 | 2114 | 2037 | 1918 | 1850 | 1785 |
| 29 | ..... | ..... | ..... | ..... | 3572 | 3476 | 3384 | 3294 | 3168 | 3083 | 2996 | 2911 | 2824 | 2738 | 2652 | 2565 | 2478 | 2391 | 2304 | 2159 | 2081 | 2006 | 1886 | 1819 | 1754 |
| 30 | 3932 | 3842 | 3752 | 3662 | 3570 | 3435 | 3344 | 3255 | 3128 | 3043 | 2957 | 2873 | 2788 | 2702 | 2617 | 2531 | 2444 | 2358 | 2272 | 2126 | 2049 | 1975 | 1854 | 1788 | 1724 |
| 31 | 3906 | 3812 | 3719 | 3625 | 3530 | 3394 | 3304 | 3216 | 3087 | 3004 | 2919 | 2836 | 2751 | 2666 | 2582 | 2496 | 2410 | 2325 | 2240 | 2093 | 2017 | 1943 | 1822 | 1756 | 1693 |
| 32 | 3862 | 3769 | 3677 | 3584 | 3489 | 3353 | 3263 | 3176 | 3047 | 2964 | 2880 | 2798 | 2714 | 2630 | 2547 | 2461 | 2377 | 2292 | 2208 | 2060 | 1984 | 1912 | 1790 | 1725 | 1663 |
| 33 | 3818 | 3726 | 3634 | 3543 | 3449 | 3312 | 3223 | 3136 | 3006 | 2925 | 2841 | 2760 | 2677 | 2594 | 2511 | 2427 | 2343 | 2259 | 2176 | 2027 | 1952 | 1880 | 1758 | 1694 | 1633 |
| 34 | 3774 | 3683 | 3592 | 3501 | 3408 | 3270 | 3182 | 3096 | 2966 | 2885 | 2802 | 2722 | 2639 | 2557 | 2475 | 2392 | 2309 | 2226 | 2143 | 1994 | 1920 | 1849 | 1726 | 1663 | 1603 |
| 35 | 3729 | 3639 | 3549 | 3459 | 3367 | 3228 | 3141 | 3056 | 2925 | 2845 | 2763 | 2683 | 2602 | 2521 | 2440 | 2357 | 2275 | 2193 | 2111 | 1961 | 1888 | 1818 | 1695 | 1633 | 1574 |
| 36 | 3685 | 3595 | 3506 | 3417 | 3325 | 3186 | 3099 | 3016 | 2884 | 2805 | 2723 | 2645 | 2564 | 2484 | 2404 | 2322 | 2241 | 2159 | 2079 | 1928 | 1856 | 1787 | 1664 | 1603 | 1544 |
| 37 | 3640 | 3551 | 3463 | 3374 | 3284 | 3144 | 3058 | 2975 | 2843 | 2765 | 2684 | 2607 | 2527 | 2448 | 2368 | 2287 | 2207 | 2126 | 2047 | 1895 | 1824 | 1756 | 1633 | 1573 | 1515 |
| 38 | 3595 | 3507 | 3419 | 3332 | 3242 | 3102 | 3017 | 2935 | 2802 | 2725 | 2645 | 2569 | 2490 | 2411 | 2333 | 2253 | 2173 | 2094 | 2015 | 1863 | 1793 | 1725 | 1602 | 1543 | 1487 |
| 39 | 3550 | 3463 | 3376 | 3290 | 3201 | 3060 | 2976 | 2895 | 2761 | 2685 | 2606 | 2531 | 2453 | 2375 | 2298 | 2218 | 2139 | 2061 | 1983 | 1831 | 1762 | 1695 | 1572 | 1514 | 1458 |
| 40 | 3505 | 3419 | 3333 | 3247 | 3159 | 3018 | 2935 | 2854 | 2721 | 2646 | 2568 | 2493 | 2416 | 2339 | 2263 | 2184 | 2106 | 2028 | 1951 | 1799 | 1731 | 1665 | 1542 | 1485 | 1430 |
| 42 | 3415 | 3330 | 3246 | 3163 | 3076 | 2934 | 2853 | 2774 | 2640 | 2567 | 2491 | 2418 | 2343 | 2268 | 2193 | 2116 | 2040 | 1964 | 1889 | 1737 | 1670 | 1606 | 1484 | 1428 | 1375 |
| 44 | 3325 | 3243 | 3160 | 3079 | 2994 | 2852 | 2772 | 2695 | 2561 | 2489 | 2415 | 2344 | 2270 | 2197 | 2124 | 2049 | 1975 | 1901 | 1827 | 1675 | 1611 | 1549 | 1427 | 1373 | 1322 |
| 46 | 3236 | 3156 | 3075 | 2995 | 2912 | 2770 | 2692 | 2617 | 2483 | 2413 | 2340 | 2271 | 2199 | 2128 | 2057 | 1984 | 1911 | 1839 | 1767 | 1616 | 1553 | 1493 | 1372 | 1320 | 1271 |
| 48 | 3148 | 3070 | 2991 | 2913 | 2832 | 2689 | 2613 | 2540 | 2406 | 2338 | 2267 | 2200 | 2130 | 2060 | 1991 | 1920 | 1849 | 1778 | 1709 | 1558 | 1497 | 1439 | ..... | ..... | ..... |
| 50 | 3062 | 2985 | 2908 | 2832 | 2752 | 2610 | 2536 | 2465 | 2331 | 2265 | 2196 | 2130 | 2062 | 1994 | 1927 | 1857 | 1788 | 1719 | 1651 | 1502 | 1442 | ..... | ..... | ..... | ..... |
| 52 | 2976 | 2901 | 2826 | 2752 | 2674 | 2533 | 2460 | 2391 | 2258 | 2193 | 2126 | 2062 | 1996 | 1930 | 1864 | 1796 | 1729 | 1662 | 1596 | ..... | ..... | ..... | ..... | ..... | ..... |
| 54 | 2893 | 2819 | 2746 | 2673 | 2597 | 2457 | 2386 | 2318 | 2186 | 2124 | 2058 | 1996 | 1931 | 1867 | 1803 | 1737 | 1671 | 1606 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 56 | 2811 | 2739 | 2668 | 2597 | 2522 | 2383 | 2313 | 2248 | 2117 | 2056 | 1992 | 1932 | 1869 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 58 | 2730 | 2660 | 2591 | 2522 | 2449 | 2310 | 2243 | 2179 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 60 | 2652 | 2584 | 2516 | 2448 | 2378 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 62 | 2576 | 2509 | 2443 | 2377 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |

Safe loads above the upper zig-zag line are for ratios of $l/r$ not over 60; those between the zig-zag lines are for ratios of $l/r$ up to 120; those below the lower zig-zag line are for ratios of $l/r$ not over 200.

of safe loads on beams. If $(L - N)$ is less than spans given in tables, web shear or buckling governs, and the coefficient of strength or section modulus should be calculated.

$$C = W(L - N) \quad . \quad . \quad . \quad . \quad . \quad [23]; \quad S = 3W(L - N)/2f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [24]$$

$$V_{\text{max.}} = W(L - N)/2L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

where $C$ = coefficient of strength; $S$ = section modulus; $f$ = allowable unit stress in bending, lb. per sq. in.; $V_{\text{max.}}$ = maximum shear due to load $W$, lb. $V_{\text{max}}$ must not exceed allowable shear $V$. For computation of $V$, see formula [2]. The proper beam should satisfy formula [23] or [24], and formula [25].

## 5. RIVETING

Rivets usually are proportioned to transmit the entire stress, friction between the riveted members being disregarded. The number and size of rivets should be such as to give ample resistance to shear, and to give sufficient bearing area to avoid distortion of metal at the rivet holes. This latter condition sometimes requires increasing the thickness of webs of beams or girders. Table 25 gives shearing and bearing values of structural rivets. Machine-driven rivets may be calculated for higher unit stresses than hand-driven rivets, as shown in the tables.

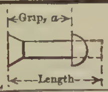
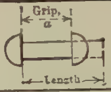
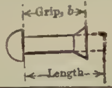
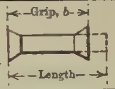
### Table 25.—Shearing and Bearing Values of Rivets

| Rivet diam., in. | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Rivet area, sq. in. | 0.1963 | .3068 | .4418 | .6013 | .7854 | .9940 | 1.2272 | .1963 | .3068 | .4418 | .6013 | .7854 | .9940 | 1.2272 |
| **Hand-driven Rivets** | | | | | | | | | | | | | | |
| Single shear, lb. | 1960 | 3070 | 4420 | 6010 | 7850 | 9940 | 12270 | 1960 | 3070 | 4420 | 6010 | 7850 | 9940 | 12270 |
| Double shear, lb. | 3930 | 6140 | 8840 | 12030 | 15710 | 19880 | 25540 | 3930 | 6140 | 8840 | 12030 | 15710 | 19880 | 25540 |
| Plate Thickness, in. | Bearing Value, lb., at 16,000 lb. per sq. in. Unit Stress | | | | | | | Bearing Value, lb., at 20,000 lb. per sq. in. Unit Stress | | | | | | |
| 1/8 | 1000 | 1250 | 1500 | 1750 | 2000 | 2250 | 2500 | 1250 | 1560 | 1880 | 2190 | 2500 | 2810 | 3130 |
| 3/16 | 1500 | 1880 | 2250 | 2630 | 3000 | 3380 | 3750 | 1880 | 2340 | 2810 | 3280 | 3750 | 4220 | 4690 |
| 1/4 | ...... | 2500 | 3000 | 3500 | 4000 | 4500 | 5000 | 2500 | 3130 | 3750 | 4380 | 5000 | 5630 | 6250 |
| 5/16 | ...... | ...... | 3750 | 4380 | 5000 | 5630 | 6250 | 3130 | 3910 | 4690 | 5470 | 6250 | 7030 | 7810 |
| 3/8 | ...... | ...... | ...... | 5250 | 6000 | 6750 | 7500 | 3750 | 4690 | 5630 | 6560 | 7500 | 8440 | 9370 |
| 7/16 | ...... | ...... | ...... | ...... | 7000 | 7880 | 8750 | ...... | 5470 | 6560 | 7660 | 8750 | 9840 | 10940 |
| 1/2 | ...... | ...... | ...... | ...... | ...... | 9000 | 10000 | ...... | ...... | 7500 | 8750 | 10000 | 11250 | 12500 |
| 9/16 | ...... | ...... | ...... | ...... | ...... | ...... | 11250 | ...... | ...... | 8440 | 9850 | 11250 | 12660 | 14060 |
| 5/8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 10940 | 12500 | 14060 | 15630 |
| 11/16 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 12030 | 13750 | 15470 | 17190 |
| 3/4 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 15000 | 16880 | 18750 |
| 13/16 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 18280 | 20310 |
| 7/8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 19690 | 21870 |
| 15/16 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 23440 |
| 1 | 8000 | 10000 | 12000 | 14000 | 16000 | 18000 | 20000 | 10000 | 12500 | 15000 | 17500 | 20000 | 22500 | 25000 |
| **Power-driven Rivets** | | | | | | | | | | | | | | |
| Single shear, lb. | 2650 | 4140 | 5960 | 8120 | 10600 | 13420 | 16570 | 2650 | 4140 | 5960 | 8120 | 10600 | 13420 | 16570 |
| Double shear, lb. | 5300 | 8280 | 11930 | 16240 | 21200 | 26840 | 33130 | 5300 | 8280 | 11930 | 16240 | 21200 | 26840 | 33130 |
| Plate Thickness, in. | Bearing Value, lb., at 24,000 lb. per sq. in. Unit Stress | | | | | | | Bearing Value, lb., at 30,000 lb. per sq. in. Unit Stress | | | | | | |
| 1/8 | 1500 | 1880 | 2250 | 2630 | 3000 | 3380 | 3750 | 1880 | 2340 | 2810 | 3280 | 3750 | 4220 | 4690 |
| 3/16 | 2250 | 2810 | 3380 | 3940 | 4500 | 5060 | 5630 | 2810 | 3520 | 4220 | 4920 | 5630 | 6330 | 7030 |
| 1/4 | ...... | 3750 | 4500 | 5250 | 6000 | 6750 | 7500 | 3750 | 4690 | 5630 | 6560 | 7500 | 8440 | 9380 |
| 5/16 | ...... | ...... | 5630 | 6560 | 7500 | 8440 | 9380 | 4690 | 5860 | 7030 | 8200 | 9380 | 10550 | 11720 |
| 3/8 | ...... | ...... | ...... | 7880 | 9000 | 10130 | 11250 | ...... | 7030 | 8440 | 9840 | 11250 | 12660 | 14050 |
| 7/16 | ...... | ...... | ...... | ...... | 10500 | 11810 | 13130 | ...... | 8200 | 9840 | 11480 | 13130 | 14770 | 16410 |
| 1/2 | ...... | ...... | ...... | ...... | ...... | ...... | 15000 | ...... | ...... | 11250 | 13130 | 15000 | 16880 | 18750 |
| 9/16 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 14770 | 16880 | 18980 | 21090 |
| 5/8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 18750 | 21090 | 23440 |
| 11/16 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 20630 | 23200 | 25780 |
| 3/4 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 25310 | 28130 |
| 13/16 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 30470 |
| 7/8 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 32810 |
| 1 | 12000 | 15000 | 18000 | 21000 | 24000 | 27000 | 30000 | 15000 | 18750 | 22500 | 26250 | 30000 | 33750 | ...... |

**CONVENTIONAL SYMBOLS FOR RIVETING** are shown in Fig. 4, which represents American Bridge Co. standards. The diagonal cross represents a countersink, the blackened circle represents a field rivet, and the diagonal stroke a flattened head. The position of the cross, with respect to the circle (inside and outside or both sides), indicates

### Table 26.—Length of Rivet Required for Various Grips

(Am. Inst. of Steel Construction Standard)

| Length of Grip, in. |  Grip *a*  | | | | | | |  Grip *b*  | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Diameter of Rivet, in. | | | | | | | Diameter of Rivet, in. | | | | | | |
| | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 |
| | Length of undriven rivet, in. | | | | | | | | | | | | | |
| 1/2 | 1 5/8 | 1 7/8 | 1 7/8 | 2 | 2 1/8 | ..... | ..... | 1 | 1 | 1 1/8 | 1 1/4 | 1 1/4 | ..... | ..... |
| 5/8 | 1 3/4 | 2 | 2 | 2 1/8 | 2 1/4 | ..... | ..... | 1 1/8 | 1 1/4 | 1 1/4 | 1 3/8 | 1 3/8 | ..... | ..... |
| 3/4 | 1 7/8 | 2 1/8 | 2 1/8 | 2 1/4 | 2 3/8 | ..... | ..... | 1 3/8 | 1 3/8 | 1 3/8 | 1 1/2 | 1 1/2 | ..... | ..... |
| 7/8 | 2 | 2 1/4 | 2 1/4 | 2 3/8 | 2 1/2 | ..... | ..... | 1 1/2 | 1 1/2 | 1 1/2 | 1 5/8 | 1 5/8 | ..... | ..... |
| 1 | 2 1/4 | 2 3/8 | 2 3/8 | 2 1/2 | 2 5/8 | 2 3/4 | 2 7/8 | 1 5/8 | 1 5/8 | 1 5/8 | 1 3/4 | 1 3/4 | 1 7/8 | 1 7/8 |
| 1 1/8 | 2 3/8 | 2 1/2 | 2 1/2 | 2 5/8 | 2 3/4 | 2 7/8 | 3 | 1 3/4 | 1 3/4 | 1 7/8 | 1 7/8 | 1 7/8 | 2 | 2 |
| 1 1/4 | 2 1/2 | 2 5/8 | 2 5/8 | 2 3/4 | 2 7/8 | 3 | 3 1/8 | 2 | 2 | 2 | 2 | 2 | 2 1/8 | 2 1/8 |
| 1 3/8 | 2 5/8 | 2 3/4 | 2 3/4 | 2 7/8 | 3 | 3 1/8 | 3 1/4 | 2 1/8 | 2 1/8 | 2 1/8 | 2 1/4 | 2 1/4 | 2 3/8 | 2 3/8 |
| 1 1/2 | 2 7/8 | 3 | 3 | 3 1/8 | 3 1/4 | 3 3/8 | 3 1/2 | 2 1/4 | 2 1/4 | 2 1/4 | 2 3/8 | 2 3/8 | 2 1/2 | 2 1/2 |
| 1 5/8 | 3 | 3 1/8 | 3 1/8 | 3 1/4 | 3 3/8 | 3 1/2 | 3 5/8 | 2 3/8 | 2 3/8 | 2 3/8 | 2 1/2 | 2 1/2 | 2 5/8 | 2 5/8 |
| 1 3/4 | 3 1/8 | 3 1/4 | 3 1/4 | 3 1/2 | 3 5/8 | 3 3/4 | 3 7/8 | 2 5/8 | 2 5/8 | 2 5/8 | 2 5/8 | 2 5/8 | 2 3/4 | 2 3/4 |
| 1 7/8 | 3 1/4 | 3 3/8 | 3 3/8 | 3 5/8 | 3 3/4 | 3 7/8 | 4 | 2 3/4 | 2 3/4 | 2 3/4 | 2 3/4 | 2 3/4 | 2 7/8 | 2 7/8 |
| 2 | 3 1/2 | 3 1/2 | 3 5/8 | 3 3/4 | 3 7/8 | 4 | 4 1/8 | 2 7/8 | 2 7/8 | 2 7/8 | 2 7/8 | 2 7/8 | 3 | 3 |
| 2 1/8 | 3 5/8 | 3 5/8 | 3 3/4 | 3 7/8 | 4 | 4 1/8 | 4 1/4 | 3 1/8 | 3 | 3 | 3 | 3 | 3 1/8 | 3 1/8 |
| 2 1/4 | 3 3/4 | 3 7/8 | 3 7/8 | 4 | 4 1/8 | 4 1/4 | 4 3/8 | 3 1/4 | 3 1/8 | 3 1/8 | 3 1/8 | 3 1/4 | 3 1/4 | 3 1/4 |
| 2 3/8 | 4 | 4 | 4 | 4 1/8 | 4 1/4 | 4 3/8 | 4 1/2 | 3 3/8 | 3 3/8 | 3 3/8 | 3 3/8 | 3 3/8 | 3 3/8 | 3 3/8 |
| 2 1/2 | 4 1/8 | 4 1/8 | 4 1/8 | 4 1/4 | 4 3/8 | 4 1/2 | 4 5/8 | 3 1/2 | 3 1/2 | 3 1/2 | 3 1/2 | 3 1/2 | 3 5/8 | 3 5/8 |
| 2 5/8 | 4 1/4 | 4 1/4 | 4 1/4 | 4 3/8 | 4 1/2 | 4 5/8 | 4 3/4 | 3 3/4 | 3 5/8 | 3 5/8 | 3 5/8 | 3 5/8 | 3 3/4 | 3 3/4 |
| 2 3/4 | 4 3/8 | 4 3/8 | 4 3/8 | 4 1/2 | 4 5/8 | 4 3/4 | 4 7/8 | 3 7/8 | 3 3/4 | 3 3/4 | 3 3/4 | 3 3/4 | 3 7/8 | 3 7/8 |
| 2 7/8 | 4 5/8 | 4 5/8 | 4 5/8 | 4 5/8 | 4 3/4 | 4 7/8 | 5 | 4 | 3 7/8 | 3 7/8 | 3 7/8 | 3 7/8 | 4 | 4 |
| 3 | ..... | 4 3/4 | 4 3/4 | 4 7/8 | 5 | 5 1/8 | 5 1/4 | ..... | 4 1/8 | 4 1/8 | 4 1/8 | 4 1/8 | 4 1/8 | 4 1/8 |
| 3 1/8 | ..... | 4 7/8 | 4 7/8 | 5 | 5 1/8 | 5 1/4 | 5 3/8 | ..... | 4 1/4 | 4 1/4 | 4 1/4 | 4 1/4 | 4 1/4 | 4 1/4 |
| 3 1/4 | ..... | 5 | 5 | 5 1/8 | 5 1/4 | 5 3/8 | 5 1/2 | ..... | 4 3/8 | 4 3/8 | 4 3/8 | 4 3/8 | 4 3/8 | 4 3/8 |
| 3 3/8 | ..... | 5 1/8 | 5 1/8 | 5 1/4 | 5 3/8 | 5 1/2 | 5 5/8 | ..... | 4 1/2 | 4 1/2 | 4 1/2 | 4 1/2 | 4 1/2 | 4 1/2 |
| 3 1/2 | ..... | 5 3/8 | 5 3/8 | 5 3/8 | 5 1/2 | 5 5/8 | 5 3/4 | ..... | 4 5/8 | 4 5/8 | 4 5/8 | 4 5/8 | 4 5/8 | 4 5/8 |
| 3 5/8 | ..... | 5 1/2 | 5 1/2 | 5 1/2 | 5 5/8 | 5 3/4 | 5 7/8 | ..... | 4 3/4 | 4 3/4 | 4 3/4 | 4 3/4 | 4 7/8 | 4 7/8 |
| 3 3/4 | ..... | 5 5/8 | 5 5/8 | 5 5/8 | 5 3/4 | 5 7/8 | 6 | ..... | 5 | 5 | 5 | 5 | 5 | 5 |
| 3 7/8 | ..... | 5 3/4 | 5 3/4 | 5 3/4 | 5 7/8 | 6 | 6 1/8 | ..... | 5 1/8 | 5 1/8 | 5 1/8 | 5 1/8 | 5 1/8 | 5 1/8 |
| 4 | ..... | ..... | 5 7/8 | 6 | 6 | 6 1/8 | 6 1/4 | ..... | ..... | 5 1/4 | 5 1/4 | 5 1/4 | 5 1/4 | 5 1/4 |
| 4 1/8 | ..... | ..... | 6 | 6 1/8 | 6 1/4 | 6 3/8 | 6 1/2 | ..... | ..... | 5 3/8 | 5 3/8 | 5 3/8 | 5 3/8 | 5 3/8 |
| 4 1/4 | ..... | ..... | 6 1/8 | 6 1/4 | 6 3/8 | 6 1/2 | 6 5/8 | ..... | ..... | 5 1/2 | 5 1/2 | 5 1/2 | 5 1/2 | 5 1/2 |
| 4 3/8 | ..... | ..... | 6 3/8 | 6 1/2 | 6 1/2 | 6 5/8 | 6 3/4 | ..... | ..... | 5 5/8 | 5 5/8 | 5 5/8 | 5 5/8 | 5 5/8 |
| 4 1/2 | ..... | ..... | 6 1/2 | 6 5/8 | 6 5/8 | 6 3/4 | 6 7/8 | ..... | ..... | 5 3/4 | 5 3/4 | 5 3/4 | 5 3/4 | 5 3/4 |
| 4 5/8 | ..... | ..... | 6 5/8 | 6 3/4 | 6 3/4 | 6 7/8 | 7 | ..... | ..... | 6 | 6 | 6 | 6 | 6 |
| 4 3/4 | ..... | ..... | 6 3/4 | 6 7/8 | 6 7/8 | 7 | 7 1/8 | ..... | ..... | 6 1/8 | 6 1/8 | 6 1/8 | 6 1/8 | 6 1/8 |
| 4 7/8 | ..... | ..... | 6 7/8 | 7 | 7 | 7 1/8 | 7 1/4 | ..... | ..... | 6 1/4 | 6 1/4 | 6 1/4 | 6 1/4 | 6 1/4 |
| 5 | ..... | ..... | ..... | 7 1/8 | 7 1/8 | 7 1/4 | 7 3/8 | ..... | ..... | ..... | 6 3/8 | 6 3/8 | 6 3/8 | 6 3/8 |
| 5 1/8 | ..... | ..... | ..... | 7 1/4 | 7 1/4 | 7 3/8 | 7 1/2 | ..... | ..... | ..... | 6 1/2 | 6 1/2 | 6 1/2 | 6 1/2 |
| 5 1/4 | ..... | ..... | ..... | 7 3/8 | 7 3/8 | 7 1/2 | 7 5/8 | ..... | ..... | ..... | 6 5/8 | 6 5/8 | 6 5/8 | 6 5/8 |
| 5 3/8 | ..... | ..... | ..... | 7 5/8 | 7 5/8 | 7 3/4 | 7 3/4 | ..... | ..... | ..... | 6 3/4 | 6 3/4 | 6 3/4 | 6 3/4 |
| 5 1/2 | ..... | ..... | ..... | 7 3/4 | 7 3/4 | 7 7/8 | 7 7/8 | ..... | ..... | ..... | 6 7/8 | 6 7/8 | 6 7/8 | 6 7/8 |
| 5 5/8 | ..... | ..... | ..... | 7 7/8 | 7 7/8 | 8 | 8 | ..... | ..... | ..... | 7 | 7 | 7 | 7 |
| 5 3/4 | ..... | ..... | ..... | 8 | 8 | 8 1/8 | 8 1/8 | ..... | ..... | ..... | 7 1/4 | 7 1/4 | 7 1/4 | 7 1/4 |
| 5 7/8 | ..... | ..... | ..... | 8 1/8 | 8 1/8 | 8 1/4 | 8 1/4 | ..... | ..... | ..... | 7 3/8 | 7 3/8 | 7 3/8 | 7 3/8 |
| 6 | ..... | ..... | ..... | 8 3/8 | 8 3/8 | 8 3/8 | 8 3/8 | ..... | ..... | ..... | 7 1/2 | 7 1/2 | 7 1/2 | 7 1/2 |
| 6 1/8 | ..... | ..... | ..... | 8 1/2 | 8 1/2 | 8 1/2 | 8 1/2 | ..... | ..... | ..... | 7 5/8 | 7 5/8 | 7 5/8 | 7 5/8 |
| 6 1/4 | ..... | ..... | ..... | 8 5/8 | 8 5/8 | 8 5/8 | 8 5/8 | ..... | ..... | ..... | 7 3/4 | 7 3/4 | 7 3/4 | 7 3/4 |
| 6 3/8 | ..... | ..... | ..... | 8 3/4 | 8 3/4 | 8 3/4 | 8 3/4 | ..... | ..... | ..... | 7 7/8 | 7 7/8 | 7 7/8 | 7 7/8 |
| 6 1/2 | ..... | ..... | ..... | 8 7/8 | 8 7/8 | 8 7/8 | 8 7/8 | ..... | ..... | ..... | 8 | 8 | 8 | 8 |
| 6 5/8 | ..... | ..... | ..... | 9 | 9 | 9 1/8 | 9 1/8 | ..... | ..... | ..... | 8 1/4 | 8 1/4 | 8 1/4 | 8 1/4 |
| 6 3/4 | ..... | ..... | ..... | 9 1/8 | 9 1/8 | 9 1/4 | 9 1/4 | ..... | ..... | ..... | 8 3/8 | 8 3/8 | 8 3/8 | 8 3/8 |
| 6 7/8 | ..... | ..... | ..... | 9 1/4 | 9 1/4 | 9 3/8 | 9 3/8 | ..... | ..... | ..... | 8 1/2 | 8 1/2 | 8 1/2 | 8 1/2 |

the location of the countersink, and similarly, the number and position of the diagonal strokes indicate the height and position of the flattened heads.

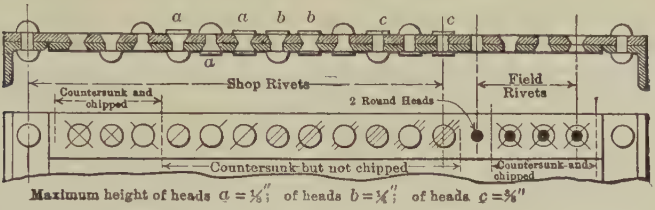

a
a
b
b
c
c
a
Shop Rivets
Countersunk and chipped
2 Round Heads
Field Rivets
Countersunk but not chipped
Countersunk and chipped
Maximum height of heads a = 1/8"; of heads b = 1/4"; of heads c = 3/8"

**Fig. 4.** Conventional Symbols for Riveting

## 6. ROOFING MATERIALS

**TERNE ROOF CONSTRUCTION.**—The following data were supplied by N. G. Taylor Co., Philadelphia: Roofs with less than $^1/_3$ pitch are made with flat seams, and preferably should be covered with 14 × 20-in. sheets, rather than with 20 × 28-in. sheets, as the larger number of seams tend to stiffen the surface and prevent buckles. For a flat seam roof the edges of the sheets are turned $^1/_2$ in., locked together and soldered. The sheets are fastened to the sheathing boards by cleats 8 in. apart and locked in the seams. Two 1-in. barbed and tinned nails are driven in each cleat. Steep tin roofs should be made with standing seams and from 28 × 20-in. sheets. The sheets are first single or double seamed and soldered together in a long strip reaching from eave to ridge. The sloping seams are composed of two "up stands" interlocked at the upper edge and held to the sheathing boards by cleats. No solder is used in standing seams as a rule. In soldering tin roofs, only a good rosin flux should be used. The use of acid must be carefully avoided.

**TIN PLATES** are made of soft sheet steel coated with tin, and are called in the trade "coke" or charcoal" plates according to the weight of coating. These terms have survived from the time when the highest quality of plate was made from charcoal-iron, while the lower grades were made from coke-iron. Plates to-day with the lighter coatings now are known as coke-plates, and are used for tin cans, etc. The various grades of charcoal-plates are designated by the letters A to AAAAA, the latter having the heaviest coating and the highest luster. One other brand, made with a heavier coating than 5A, is especially adapted for nickel-plating. The unit of value and measurement of tin plates is the "base-box," which is 112 sheets of 14 × 20 in. plate, or 31,360 sq. in. of any size. Plates lighter

**Table 27.—Weights of Roofing Materials**

| Material | Average Weight, lb. per sq. ft. |
|---|---|
| Corrugated Galvanized Sheet Steel | |
| No. 18 Galv. Sheet Gage | 2 3/4 |
| No. 20 Galv. Sheet Gage | 2 1/4 |
| No. 22 Galv. Sheet Gage | 1 3/4 |
| Copper, No. 22 B. & S. Gage | 1 1/4 |
| Felt, 2 layers | 1/2 |
| Felt and asphalt or coal tar | 2 |
| Glass, 1/8 in. thick | 1 3/4 |
| Hemlock sheathing, 1 in. thick | 2 1/2 |
| Lath and plaster ceiling (ordinary) | 6 to 8 |
| Lead, 1/8 in. thick | 7 1/2 |
| Shingles, 6 × 18—1/3 to weather | 2 |
| Skylight of glass, 3/16 to 1/2 in. including frame | 4 to 10 |
| Slag roof, 4-ply, with cement and sand | 4 |
| Slate, 1/8 in. thick, 3 in. double lap | 4 1/2 |
| Slate, 3/16 in. thick, 3 in. double lap | 6 3/4 |
| Spruce sheathing, 1 in. thick | 2 1/4 |
| Terne plate, IC | 5/8 |
| Terne plate, IX | 3/4 |
| Tiles (plain) 10 1/2 × 6 1/4 × 5/8 in.—5 1/4 in. to weather | 18 |
| Tiles (Spanish) 14 1/2 × 10 1/2 in.—7 1/4 in. to weather | 8 1/2 |
| White pine sheathing, 1 in. thick | 2 1/4 |
| Yellow pine sheathing, 1 in. thick | 3 1/2 |
| Zinc, No. 20 B. & S. Gage | 1 1/4 |

than 65 lb. per base-box (No. 36 gage) are known as taggers tin. No. 38 (55 lb. per base box) is the lightest plate regularly made.

**TERNE PLATES**, or Roofing Tin, are coated with an alloy of approximately 25% tin, 75% lead. The coating ranges from 40 lb. per box of 112 5A sheets to 20 lb. per box of 1A sheets and from 15 lb. to 8 lb. per box in other grades.

Terne plates are made in two thicknesses: IC, in which the iron body weighs about 50 lb. per 100 sq. ft., and IX, in which it weighs 62 1/2 lb. per 100 sq. ft. The IC grade is preferred for roofing, while the IX grade is used for spouts, valleys, gutters, and flashings. The standard weight of 14 × 20-in. IC plates is 107 lb. per base box, and of 14 × 20-in. IX plate 135 lb., with 8 lb. coating per double base box.

Long terne sheets are made in gages, Nos. 16 to 32, up to 48 in. wide and up to 144 in. long. They are made in three grades with coatings of "Regular," "Heavy," and 40-lb.

A box of 112 sheets 14 × 20 in. will cover approximately 180 sq. ft. of roof, flat seam, or 623 sheets 1000 sq. ft. For standing seam roofing a sheet 20 × 28 in. will cover 457 1/8 sq. in. or 316 sheets 1000 sq. ft. A box of 112 sheets 20 × 28 in. will cover approximately 356 sq. ft.

### Table 28.—Weights of Standard Galvanized Sheets

(American Sheet & Tin Plate Co., Pittsburgh)

| Gage | Oz. per sq. ft. | Lb. per sq. ft. | Gage | Oz. per sq. ft. | Lb. per sq. ft. | Gage | Oz. per sq. ft. | Lb. per sq. ft. | Gage | Oz. per sq. ft. | Lb. per sq. ft. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 92.5 | 5.781 | 15 | 47.5 | 2.969 | 20 | 26.5 | 1.656 | 25 | 16.5 | 1.031 |
| 11 | 82.5 | 5.156 | 16 | 42.5 | 2.656 | 21 | 24.5 | 1.531 | 26 | 14.5 | 0.906 |
| 12 | 72.5 | 4.531 | 17 | 38.5 | 2.406 | 22 | 22.5 | 1.406 | 27 | 13.5 | .844 |
| 13 | 62.5 | 3.906 | 18 | 34.5 | 2.156 | 23 | 20.5 | 1.281 | 28 | 12.5 | .781 |
| 14 | 52.5 | 3.281 | 19 | 30.5 | 1.906 | 24 | 18.5 | 1.156 | 29 | 11.5 | .719 |
| | | | | | | | | | 30 | 10.5 | .656 |

### Table 29.—Standard Weights and Gages of Tin Plate

(American Sheet & Tin Plate Co., Pittsburgh)

| Trade Term | Nearest U. S. Std. Gage | Wt. per sq. ft. lb. | Wt. of Box, 14 × 20 in., lb. | Trade Term | Nearest U. S. Std. Gage | Wt. per sq. ft. lb. | Wt. of Box, 14 × 20 in., lb. | Trade Term | Nearest U. S. Std. Gage | Wt. per sq. ft. lb. | Wt. of Box, 14 × 20 in., lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 55-lb. | 38 | 0.253 | 55 | 118-lb. | 29 | 0.542 | 118 | 5XL | 24 | 0.955 | 208 |
| 60 " | 37 | .276 | 60 | IXL | 29 | .588 | 128 | D2X | 24 | .964 | 210 |
| 65 " | 36 | .298 | 65 | IX | 28 | .620 | 135 | 5X | 24 | .987 | 215 |
| 70 " | 35 | .321 | 70 | DC | 28 | .638 | 139 | 6XL | 24 | 1.047 | 228 |
| 75 " | 34 | .344 | 75 | 2XL | 27 | .680 | 148 | 6X | 23 | 1.079 | 235 |
| 80 " | 33 | .367 | 80 | 2X | 27 | .712 | 155 | D3X | 23 | 1.102 | 240 |
| 85 " | 32 | .390 | 85 | 3XL | 26 | .771 | 168 | 7XL | 23 | 1.139 | 248 |
| 90 " | 31 | .413 | 90 | 3X | 26 | .804 | 175 | 7X | 23 | 1.171 | 255 |
| 95 " | 31 | .436 | 95 | DX | 25 | .827 | 180 | 8XL | 22 | 1.231 | 268 |
| ICL | 30 1/2 | .459 | 100 | 4XL | 25 | .863 | 188 | D4X | 22 | 1.240 | 270 |
| IC | 30 | .491 | 107 | 4X | 25 | .895 | 195 | 8X | 22 | 1.263 | 275 |

### Table 30.—Galvanized Sheet Gage

(American Sheet & Tin Plate Co., Pittsburgh)

| Gage No. | Oz. per sq. ft. | Lb. per sq. ft. | Gage No. | Oz. per sq. ft. | Lb. per sq. ft. | Gage No. | Oz. per sq. ft. | Lb. per sq. ft. | Gage No. | Oz. per sq. ft. | Lb. per sq. ft. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 8 | 112.5 | 7.031 | 15 | 47.5 | 2.969 | 22 | 22.5 | 1.406 | 29 | 11.5 | 0.719 |
| 9 | 102.5 | 6.406 | 16 | 42.5 | 2.656 | 23 | 20.5 | 1.281 | 30 | 10.5 | .656 |
| 10 | 92.5 | 5.781 | 17 | 38.5 | 2.406 | 24 | 18.5 | 1.156 | 31 | 9.5 | .594 |
| 11 | 82.5 | 5.156 | 18 | 34.5 | 2.156 | 25 | 16.5 | 1.031 | 32 | 9.0 | .563 |
| 12 | 72.5 | 4.531 | 19 | 30.5 | 1.906 | 26 | 14.5 | 0.906 | 33 | 8.5 | .531 |
| 13 | 62.5 | 3.906 | 20 | 26.5 | 1.656 | 27 | 13.5 | .844 | 34 | 8.0 | .500 |
| 14 | 52.5 | 3.281 | 21 | 24.5 | 1.531 | 28 | 12.5 | .781 | | | |

**TIN PLATE SPECIFICATIONS.**—(Condensed from U. S. Army Specification FXS No. 57—140A, December, 1929.) Applies to Grade A Light (Coke), Grade B Heavy (1A Charcoal) and Grade C Extra Heavy (2A Charcoal). Sheets to be of soft Bessemer or open hearth steel, or iron, free from laminations, blisters, slivers, open seams, scale pits, holes, and similar imperfections, and to be as uniform in thickness as practicable. Tin coating to be of commercially pure tin. Weights shall conform to Tin Plate Gage

(See Table 29) and the net weight of the finished product shall conform to the tolerances in Table 32.

**Coating.**—Grade A, no specific weight; Grade B, average of not less than 1.80 lb. per base box; Grade C, average of not less than 2.30 lb. per base box.

**Bending and Seaming.**—Sheets shall withstand bending cold through an angle of 180 deg., and flattening with a wooden mallet on a wooden block, without fracture on the outside of the bent portion; shall also stand double seaming without fracture across the direction of rolling.

**Squaring.**—Sheets shall not be out of square more than 1/8 in. per ft. Resquared sheets shall not be out of square more than 1/16 in., shall have no minus tolerance, and a plus tolerance of 1/16 in. in each dimension.

**LONG TERNE SHEET SPECIFICATIONS.**—(Condensed from U. S. Army Ordnance Department Specifications FXS No. 57—140B, November, 1929.) Base metal shall be either soft Bessemer or open hearth steel or iron, as ordered. Coated sheets of prime grade shall be free from laminations, blisters, slivers, open seams, mill scale, pit holes, and similar imperfections, and uncoated spots readily observable by the unaided eye. Seconds are sheets containing one or more of the above defects, and shall not exceed 20% of any one gage and size. A tolerance of 5% shall be allowed for sheets of one classification in a lot of another. All sheets shall be as nearly uniform in thickness as practicable; shall be commercially flat, and be as evenly coated on both sides as good mill practice will permit. Gage weights of coated sheets shall be those of the U. S. Standard Gage (see Table 12, p. 6–24).

**Tolerances in Dimensions.**—Sheets shall have no minus tolerance. Width of sheets shall not exceed that ordered by more than

1/4 in. for sheets ordered up to 36 in. wide and up to 120 in. long.
5/16 in. " " " " " 36 in. " " over 120 in. long.
3/8 in. " " " over 36 in. " " up to 120 in. long.
7/16 in. " " " " 36 in. " " over 120 in. long.

**Table 31.—Sheets Required for Tin Roofing**

(American Sheet & Tin Plate Co., Pittsburgh)

| No. of sq. ft. | Sheets Required | | No. of sq. ft. | Sheets Required | | No. of sq. ft. | Sheets Required | | No. of sq. ft. | Sheets Required | | No. of sq. ft. | Sheets Required | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Flat Seam, 14×20 Plate | Stand. Seam, 20×28 Plate | | Flat Seam, 14×20 Plate | Stand. Seam, 20×28 Plate | | Flat Seam, 14×20 Plate | Stand. Seam, 20×28 Plate | | Flat Seam, 14×20 Plate | Stand Seam, 20×28 Plate | | Flat Seam, 14×20 Plate | Stand. Seam, 20×28 Plate |
| 100 | 63 | 32 | 280 | 175 | 89 | 460 | 287 | 145 | 640 | 399 | 202 | 820 | 511 | 259 |
| 110 | 69 | 35 | 290 | 181 | 92 | 470 | 293 | 149 | 650 | 405 | 205 | 830 | 517 | 262 |
| 120 | 75 | 38 | 300 | 187 | 95 | 480 | 299 | 152 | 660 | 411 | 208 | 840 | 524 | 265 |
| 130 | 81 | 41 | 310 | 194 | 98 | 490 | 306 | 155 | 670 | 418 | 212 | 850 | 530 | 268 |
| 140 | 88 | 45 | 320 | 200 | 101 | 500 | 312 | 158 | 680 | 424 | 215 | 860 | 536 | 271 |
| 150 | 94 | 48 | 330 | 206 | 104 | 510 | 318 | 161 | 690 | 430 | 218 | 870 | 542 | 275 |
| 160 | 100 | 51 | 340 | 212 | 108 | 520 | 324 | 164 | 700 | 436 | 221 | 880 | 548 | 278 |
| 170 | 106 | 54 | 350 | 218 | 111 | 530 | 331 | 168 | 710 | 443 | 224 | 890 | 555 | 281 |
| 180 | 113 | 57 | 360 | 225 | 114 | 540 | 337 | 171 | 720 | 449 | 227 | 900 | 561 | 284 |
| 190 | 119 | 60 | 370 | 231 | 117 | 550 | 343 | 174 | 730 | 455 | 231 | 910 | 567 | 287 |
| 200 | 125 | 64 | 380 | 237 | 120 | 560 | 349 | 177 | 740 | 461 | 234 | 920 | 573 | 290 |
| 210 | 131 | 67 | 390 | 243 | 123 | 570 | 355 | 180 | 750 | 468 | 237 | 930 | 580 | 294 |
| 220 | 137 | 70 | 400 | 250 | 127 | 580 | 362 | 183 | 760 | 474 | 240 | 940 | 586 | 297 |
| 230 | 144 | 73 | 410 | 256 | 130 | 590 | 368 | 186 | 770 | 480 | 243 | 950 | 592 | 300 |
| 240 | 150 | 76 | 420 | 262 | 133 | 600 | 374 | 190 | 780 | 486 | 246 | 960 | 598 | 303 |
| 250 | 156 | 79 | 430 | 268 | 136 | 610 | 380 | 193 | 790 | 492 | 249 | 970 | 605 | 306 |
| 260 | 162 | 82 | 440 | 274 | 139 | 620 | 387 | 196 | 800 | 499 | 253 | 980 | 611 | 309 |
| 270 | 169 | 86 | 450 | 281 | 142 | 630 | 393 | 199 | 810 | 505 | 256 | 990 | 617 | 312 |

**Table 32.—Allowable Variation in Base Weight of Tin Plate**

| | Less than 272 lb. per Base Box * | 272 lb. to 544 lb. per Base Box † | Not Less than 544 lb. per Base Box † |
|---|---|---|---|
| All of one gage or base weight and size ‡ | ± 2.5% | ± 3.5% | ± 5.0% |
| If packed in single packages | ± 4.0% | ± 5.5% | ± 7.0% |
| Single sheets | ±10.0% | ±10.0% | ±10.0% |

* 20 oz. per sq. ft. † 40 oz. per sq. ft. ‡ Lots of not less than 6000 lb.

Length of sheets shall not vary more than 2% of the ordered length. Tolerances in weights of finished product will be as in Table 33.

**Table 33.—Allowable Variations in Base Weight of Terne Plate**

| Gage Range | | | | Permissible Variation in Weight of Sheets, Percentage of Theoretical Weight | | |
|---|---|---|---|---|---|---|
| Weight, oz. per sq. ft. | | Weight, lb. per sq. ft. | | | | |
| Less than | Not Less than | Less than | Not Less than | All of One Gage and Size in Shipment | Single Package | Single Sheet |
| .......... | 40 (No. 16) | ............ | 2.5 (No. 16) | ±5.0 | ±7.0 | ±10.0 |
| 40 (No. 16) | 20 (No. 22) | 2.5 (No. 16) | 1.25 (No. 22) | ±3.5 | ±5.5 | ±10.0 |
| 20 (No. 22) | .......... | 1.25 (No. 22) | ............ | ±2.5 | ±4.0 | ±10.0 |

**CORRUGATED SHEETS.**—Table 34 gives dimensions and weights of corrugated sheets, which are made in standard lengths of 5, 6, 7, 8, 9, and 10 ft.; maximum length, 12 ft. Covering widths of sheets, lapped one corrugation, are:

| | | | | | | |
|---|---|---|---|---|---|---|
| Covering Width, in.................... | 24 3/8 | 23 3/4 | 24 | 24 | 24 | 25 |
| Lap, in............................. | 5/8 | 1 1/4 | 2 | 2 1/2 | 3 | 5 |

Ordinary corrugated sheets should have a lap of 1 1/2 or 2 corrugations side lap and 6 in. end lap for roofing in order to make water-tight side seams; for siding, 1 corrugation side lap, and 4 in. end lap. Some manufacturers make a special high-edge corrugation on sides of sheets, and thereby are enabled to secure a waterproof side lap with one corrugation only, thus saving from 6% to 12% of material to cover a given area.

No. 28 gage corrugated iron is generally used for applying to wooden buildings; but for applying to iron framework No. 24 gage or heavier should be adopted.

Galvanized steel is about 2 1/2 oz. per square foot heavier than uncoated flat sheet steel of the same gage number.

**CORRUGATED ARCHES.**—For corrugated curved sheets for floor and ceiling construction in fireproof buildings, No. 16, 18 or 20 gage iron is commonly used, and sheets may be curved from 4 to 10 in. rise—the higher the rise the stronger the arch. By a series of tests it has been demonstrated that corrugated arches give the most satisfactory results with a base length not exceeding 6 ft., and 5 ft. or even less is preferable where great strength is required. These corrugated arches are made with 1 1/4 × 3/8, 2 1/2 × 1/2, 3 × 3/4, and 5 × 7/8 in. corrugations, and in the same width of sheet as above mentioned.

**SLATE.**—Slate in roofs is measured by the square, 1 square being equal to 100 superficial square feet. In measuring, the width of the eaves is allowed at the widest part.

**Table 34.—Weight of Corrugated Sheets**

**(American Sheet & Tin Plate Co., Pittsburgh)**

| Corrugations, | 5/8 in.* | | 1 1/4 in.* | | 2 in.* | | 2 1/2 in.* | | 2 1/2 in.† | | 3 in.* | | 5 in.* | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Formed Width | 25 in. Wide | | 25 in. Wide | | 26 in. Wide | | 26 in. Wide | | 27 1/2 in. Wide | | 26 in. Wide | | 28 in. Wide | |
| U. S. Std. Sheet Metal Gage | Painted | Galvanized | Painted | Galvanized | Painted | Galvanized | Painted | Galvanized | Painted | Galvanized | Painted | Galvanized | Painted | Galvanized |
| | Lb. per 100 sq. ft. | | | | | | | | | | | | | |
| 29 | ..... | 81 | ..... | 81 | ..... | 77 | ..... | 77 | ..... | 78 | ..... | 77 | ..... | ..... |
| 28 | 71 | 88 | 71 | 88 | 68 | 84 | 68 | 84 | 69 | 85 | 68 | 84 | ..... | ..... |
| 27 | 78 | 95 | 78 | 95 | 75 | 91 | 75 | 91 | 76 | 92 | 75 | 91 | ..... | ..... |
| 26 | 85 | 102 | 85 | 102 | 82 | 98 | 82 | 98 | 83 | 99 | 82 | 98 | ..... | ..... |
| 25 | 99 | 116 | 99 | 116 | 95 | 111 | 95 | 111 | 97 | 113 | 95 | 111 | ..... | ..... |
| 24 | 113 | 130 | 113 | 130 | 109 | 125 | 109 | 125 | 110 | 126 | 109 | 125 | ..... | ..... |
| 23 | ..... | ..... | 127 | 144 | 122 | 138 | 122 | 138 | 124 | 140 | 122 | 138 | ..... | ..... |
| 22 | ..... | ..... | 141 | 158 | 136 | 151 | 136 | 151 | 137 | 153 | 136 | 151 | 135 | 151 |
| 21 | ..... | ..... | 155 | 172 | 149 | 165 | 149 | 165 | 151 | 167 | 149 | 165 | 148 | 164 |
| 20 | ..... | ..... | 169 | 186 | 163 | 178 | 163 | 178 | 165 | 181 | 163 | 178 | 162 | 177 |
| 18 | ..... | ..... | ..... | ..... | 216 | 232 | 216 | 232 | 219 | 235 | 216 | 232 | 215 | 231 |
| 16 | ..... | ..... | ..... | ..... | 270 | 286 | 270 | 286 | 274 | 290 | 270 | 286 | 269 | 284 |
| 14 | ..... | ..... | ..... | ..... | ..... | ..... | 338 | 353 | 342 | 358 | 338 | 353 | 335 | 351 |
| 12 | ..... | ..... | ..... | ..... | ..... | ..... | 472 | 488 | 478 | 494 | 472 | 488 | 469 | 485 |
| 10 | ..... | ..... | ..... | ..... | ..... | ..... | 607 | 623 | 615 | 631 | ..... | ..... | ..... | ..... |

* Siding. † Roofing.

Hips, valleys, and cuttings are measured lineally and 6 in. extra is allowed. The thickness of slate for roofing ranges usually from $^1/_8$ to $^3/_{16}$ in. The weight ranges, when lapped, from 4 $^1/_2$ to 6 $^3/_4$ lb. per sq. ft. The laps range from 2 to 4 in., 3 in. being the standard. As slate usually is laid, the number of square feet of roof covered by one slate is $w\ (l - 3) \div 288$, $w$ and $l$ being the width and length respectively of the slate in inches.

**Table 35.—Number and Superficial Area of Slate for One Square of Roof**

| Size, in. | No. per Square | Area, sq. ft. | Size, in. | No. per Square | Area, sq. ft. | Size, in. | No. per Square | Area, sq. ft. | Size, in. | No. per Square | Area, sq. ft. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 6×12 | 533 | 267 | 10×14 | 261 | ...... | 10×20 | 169 | 235 | 12×24 | 114 | 228 |
| 7×12 | 457 | ...... | 8×16 | 277 | 246 | 11×20 | 154 | ...... | 14×24 | 98 | ...... |
| 8×12 | 400 | ...... | 9×16 | 246 | ...... | 12×20 | 141 | ...... | 16×24 | 86 | ...... |
| 9×12 | 355 | ...... | 10×16 | 221 | ...... | 14×20 | 121 | ...... | 14×26 | 89 | 225 |
| 7×14 | 374 | 254 | 9×18 | 213 | 240 | 16×20 | 137 | ...... | 16×26 | 78 | ...... |
| 8×14 | 327 | ...... | 10×18 | 192 | ...... | 12×22 | 126 | 231 | ........ | ...... | ...... |
| 9×14 | 291 | ...... | 12×18 | 160 | 240 | 14×22 | 108 | ...... | ........ | ...... | ...... |

**Table 36.—Weight of Slate, in Pounds, for One Square of Roof**

(1 cu. ft. slate = 175 lb.)

| Length of Slate, in. | Thickness of Slate, in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | $^1/_8$ | $^3/_{16}$ | $^1/_4$ | $^3/_8$ | $^1/_2$ | $^5/_8$ | $^3/_4$ | 1 |
| 12 | 483 | 724 | 967 | 1450 | 1936 | 2419 | 2902 | 3872 |
| 14 | 460 | 688 | 920 | 1379 | 1842 | 2301 | 2760 | 3683 |
| 16 | 445 | 667 | 890 | 1336 | 1784 | 2229 | 2670 | 3567 |
| 18 | 434 | 650 | 869 | 1303 | 1740 | 2174 | 2607 | 3480 |
| 20 | 425 | 637 | 851 | 1276 | 1704 | 2129 | 2553 | 3408 |
| 22 | 418 | 626 | 836 | 1254 | 1675 | 2093 | 2508 | 3350 |
| 24 | 412 | 617 | 825 | 1238 | 1653 | 2056 | 2478 | 3306 |
| 26 | 407 | 610 | 815 | 1222 | 1631 | 2039 | 2445 | 3263 |

**Terra Cotta.**—*Porous terra cotta* roofing 3 in. thick weighs 16 lb. per square foot and 2 in. thick, 12 lb. per square foot.

**Ceiling** made of the same material 2 in. thick, weighs 11 lb. per square foot.

**TILES.—Flat tiles** 6 $^1/_4$ × 10 $^1/_2$ × $^5/_8$ in. weigh from 1480 to 1850 lb. per square of roof (100 square feet), the lap being one-half the length of the tile.

**Tiles with Grooves and Fillets** weigh from 740 to 925 lb. per square of roof.

**Pan-tiles** 14 $^1/_2$ × 10 $^1/_2$ laid 10 in. to the weather weigh 850 lb. per square.

**SHINGLES.**—Pine shingles are 16 in. long, 2 $^1/_4$ to 14 in. wide, and $^5/_{16}$ in. thick at the butt by $^1/_{16}$ in. thick at the thin end. Cypress shingles are 16 in. long and $^2/_5$ in. thick at the butt. In laying, 4 in. of the shingle are exposed to the weather.

The figures below give the weight of pine shingles required to cover one square of a common gable roof. For hip roofs add 5%.

| | | | | | |
|---|---|---|---|---|---|
| Inches exposed to weather.................... | 4 | 4 $^1/_2$ | 5 | 5 $^1/_2$ | 6 |
| No. of shingles per square of roof.............. | 900 | 800 | 720 | 655 | 600 |
| Weight of shingles per square, lb.............. | 216 | 192 | 173 | 157 | 144 |

**Table 37.—Skylight Glass Required for One Square of Roof**

| | | | | |
|---|---|---|---|---|
| Dimensions, in...................... | 12 × 48 | 15 × 60 | 20 × 100 | 94 × 156 |
| Thickness, in......................... | $^3/_{16}$ | $^1/_4$ | $^3/_8$ | $^1/_2$ |
| Area, sq. ft.......................... | 3.997 | 6.246 | 13.880 | 101.768 |
| Weight per square, lb................. | 250 | 350 | 500 | 700 |

No allowance has been made in the above figures for lap. If ordinary window-glass is used, single thick glass (about $^1/_{16}$ in.) will weigh about 82 lb. per square, and double thick glass (about $^1/_8$ in.) will weigh about 164 lb. per square, *no allowance being made for lap.* A box of ordinary window-glass contains as nearly 50 square feet as the size of the panes will admit. Panes of any size are made to order by the manufacturers, but a great variety of sizes are usually kept in stock, ranging from 6 × 8 in. to 36 × 60 in.

ROOF PAINTS (American Sheet & Tin Plate Co., 1932).—All sheets should be clean, to give close contact of paint and metal. Uncoated sheets should be free from mill oxide and rust, and if necessary, wire brushed and wiped with turpentine or gasoline. Grease and oil may be washed off with naphtha or toluol. Terne coated (roofing tin) is best wiped with benzine, turpentine, or benzol to remove the oil film. Zinc coated (galvanized) is best prepared by simple exposure to the weather for a few months. New zinc coated sheets first should be brushed with a solution of 4 oz. copper sulphate, copper chloride, or copper acetate in 1 gal. water, and after drying brushed lightly. Better results may be obtained by cleaning with a solution of 200 parts water, 30 parts trisodium phosphate, and 4 parts sodium hydroxide, all by weight, at 140 to 180 deg. F., then washing with hot water, followed by etching at room temperature for about $^1/_2$ min., with a solution of 1000 parts water, 200 parts hydrochloric acid, and 20 parts ammonium nitrate, all by weight, and then washing and thoroughly drying.

The weather should be clear and dry and the paint preferably applied in the afternoon, as it is essential that the surface be moisture free. Coat the surfaces in laps with fairly thick priming paint as the sheets are placed in position. The priming coat should be thin enough to fill and wet all parts. The finish coat should be thicker. Not less than two coats should be applied, with time for drying between coats.

Paint applied to sheet metal should be rust inhibitive. The priming coat should have as a basic pigment, red lead, lead chromate, or zinc chromate. Zinc dust and zinc oxide, 3 or 4 to 1, make efficient paint both for priming and finishing, especially as primer for galvanized sheets. Iron oxide paint is used for priming, but much better protection is obtained if it contains 10% or more of basic or chromate pigment, as zinc oxide, red lead, or zinc chromate. The pigment for the finish may be any good moisture excluder of suitable color. Linseed oil, preferably boiled, is a good vehicle for pigment, and excessive dryer should not be used. Boiled linseed oil or durable mixing varnish as part of the last coat adds to gloss and water resistance. It is economy to use only the best grades of paint for metal work.

# Section 18

# INDUSTRIAL HEATING PROCESSES

## INDUSTRIAL FURNACES

By W. Trinks

## INDUSTRIAL ELECTRIC HEATING

By R. M. Cherry

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INDUSTRIAL FURNACES

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# INDUSTRIAL HEATING PROCESSES

## INDUSTRIAL FURNACES*

**By W. Trinks**

**References.**—Bullens, Steel and Its Treatment (John Wiley & Sons); Trinks, Industrial Furnaces (John Wiley & Sons), Vols. I and II; Mawhinney, Practical Industrial Furnace Design (John Wiley & Sons); monthly publication *Industrial Heating*, Pittsburgh, Pa.

Many shop processes require the application of heat to solid material without melting it. The process, known as industrial heating, is carried on in industrial furnaces. The charge or stock usually is metal, but occasionally is a ceramic material. For iron or steel, the temperature ranges given in Table 1 are used. For other metals, temperatures well below their melting points are used. For the latter values, see p. 1–05 or Fig. 1.

### 1. FURNACE CHARACTERISTICS

**CLASSIFICATION OF FURNACES.**—Furnaces are classified: 1. According to use, as heating furnaces, forge furnaces, annealing furnaces, bright-annealing furnaces, heat-treating furnaces, and normalizing furnaces.

2. According to material or shape which they heat, as: rivet furnaces, bolt heading furnaces, plate furnaces, and angle furnaces.

3. According to the method of movement of the charge in the furnace, as: batch-type, also called in-and-out type, continuous furnaces, automatic or continuous-conveyor furnaces, rocker-bar or intermittent-conveyor furnaces, rotating hearth, car-type, and revolving drum.

4. According to the fuel or other source of heat energy, as combustion type, gas-fired, oil-fired, coal-fired, electrical resistor type.

5. According to the method of fuel saving, as: regenerative, recuperative, preheating-chamber type.

In muffle furnaces, the charge is heated in a gas-tight enclosure, and is separated by it from the source of heat.

**ECONOMY OF HEATING** must be based on unit weight of perfect product. The latter term involves such features as freedom from internal cracks caused by too rapid heating, uniformly high quality of heat-treated products, uniform annealing, uniform enamel of first quality, proper distribution of hardness, freedom from scale and ashes, freedom from burning of high-carbon steels, etc. In certain operations, such as plate bending, railroad spike forming, bolt heading and tool hardening, uniformity of temperature throughout the heated stock is not necessary.

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* Nomenclature used in this section is as of 1937. Two separate committees are at present engaged in standardizing nomenclature.

**Table 1.—Furnace Temperature Range for Industrial Processes on Iron or Steel**

| Process | Temperature Range, deg. F. | Process | Temperature Range, deg. F. |
|---|---|---|---|
| Hammer or press welding | 2350 to 2550 | Hardening carbon steel (quenching) | 1450 to 1550 |
| Hardening high-speed steel * | 2200 to 2400 | Annealing cold-rolled steel | 1250 to 1400 |
| Drop forging | 2250 to 2370 | Drawing | 1150 to 1200 |
| Rolling (low carbon steel) | 2150 to 2275 | Nitriding | 950 |
| Bending and rough forming | 1900 to 2100 | Tempering | 500 to 800 |
| Normalizing, also for carburizing | 1650 to 1750 | Bluing † | 500 to 650 |
| Rough annealing | 1500 to 1650 | | |

* Special steels require various temperatures for hardening, usually ranging from 2400° F. for greatest hardness to 1850° F. for greatest toughness (for tungsten steel).

† Still lower temperatures are used for drying and japanning.

True heating economy is inversely proportional to total cost per unit of product. It can be expressed by

$$E = W \div (I + M + L + F),$$

where $E$ = true heating economy; $I$ = interest on investment; $M$ = cost of maintenance; $L$ = cost of labor, including overhead; $F$ = cost of fuel; $W$ = unit weight of perfect product.

The amount of capital which profitably can be invested in a furnace installation depends upon regularity and quantity of output of heated material. Cost of maintenance, $M$, depends on furnace temperature, furnace design, selection of refractories, care used in operation, and on cost per unit of labor and materials used in repairs. Cost of operating labor, $L$, depends on regularity of production with regard to continuous output, size and shape of stock, and on the provision of labor-saving equipment conjointly with unit cost of labor per hour. Cost of fuel or energy, $F$, depends on heating temperature, heating process, unit cost of fuel or energy, furnace design, and care used in operation. Frequently the importance of fuel cost is overemphasized at the expense of the other items (including quality of finished product), because it is more readily amenable to calculation.

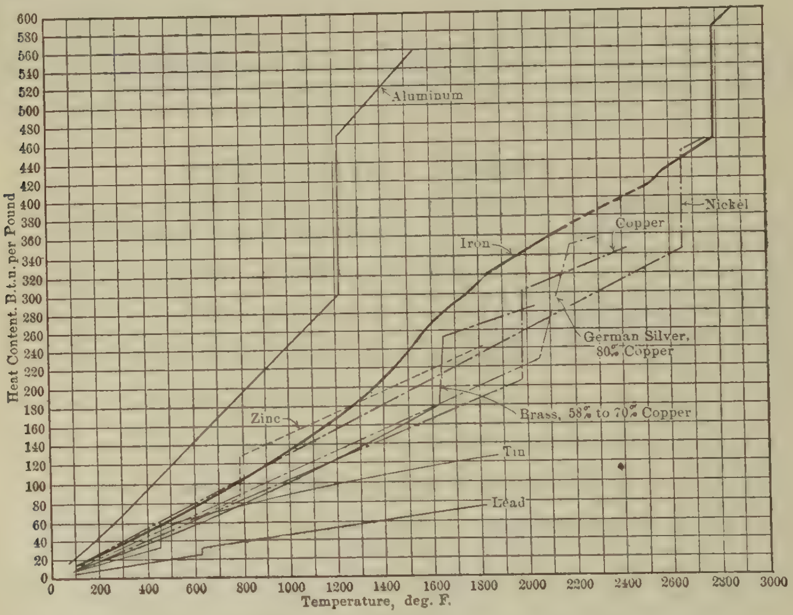

Fig. 1. Heat Content of Metals at Various Temperatures

**DISTRIBUTION OF HEAT.**—The potential heat energy of the fuel (or other source of heat) is distributed as follows: 1. Useful heat in charge. 2. Heat unavailably stored in the furnace walls. 3. Heat losses through solid walls. 4. Heat radiated through openings. 5. Heat to cooling water. 6. Unconsumed combustible in refuse. 7. Sensible heat in stack gases. 8. Combustible matter in spent gases. 9. Heat conducted through resistor terminals in electric furnaces.

Heating requirements and heating processes are so different that all of these items vary within wide limits. In electrically-heated furnaces, items 6, 7, and 8 are absent. The fuel consumption per unit weight of heated material can be computed by detail calculation of each of the eight items. While this method is followed for large furnaces, it seldom is used for the average type and size of industrial furnaces. Frequently, fuel consumption is divided into *holding consumption* and *useful consumption.* The former, also called *furnace constant*, is the rate at which fuel or energy must be supplied to keep the empty furnace hot.

The human element and variations in use of furnace cause excessive variations of fuel or energy consumption per unit of heated material. For that reason, the data contained in Table 2 are rough averages only. The unit fuel consumption may be as low as 60% of the values in the table, or may exceed them by 100%.

In furnaces which operate only during the day and are allowed to cool down over night, much heat is lost by storage in, and later dissipation from, the firebrick walls. In such cases, insulating refractories should be used. These weigh about one-third as much and have only one-third of the thermal conductivity of firebrick. When firebrick is replaced by the insulating refractories, the time required for heating up the furnace before starting work in the morning usually is reduced between one-third and one-fourth of the time required with firebrick walls. The fuel consumption for heating up is reduced in about the same proportion. In furnaces which are shut down at night the wall losses usually are about $1\,^1/_2$ times the heat imparted to the charge, depending, however, on the rate of heating in pounds of metal per square foot of hearth area per hour. The use of insulating refractories thus effects a saving of about 40% in the daily fuel consumption.

**CAPACITY OF FURNACES.**—The number of furnaces, and the size of each furnace, which are required to heat a given weight of metal to a given temperature in a given time, depend on several variables, as: 1. Thermal conductivity and specific heat of metal. 2. Requirements for uniformity of temperature in charge. 3. Rate at which heat may be applied without injury to stock. 4. Ratio of surface of charge to mass. 5. Ratio of brick surface to exposed surface of charge. 6. State of subdivision of charge. 7. Circulation of furnace gases. 8. Method of heat liberation, and size of heat liberating devices. 9. Cycle of operation of heating process. 10. Arrangement of charge in furnace. In determining number and size of furnaces, total hearth area required is determined from the following rules, and then is subdivided into standard size units to suit plant conditions.

In view of the great number of variables, the following practical rules for furnace capacity must be used with great care:

1. In iron or steel, heat penetrates at the rate of $^1/_8$ in. in five minutes. A plate $^1/_4$ in. thick, suspended in a furnace, practically assumes furnace temperature in the center of the plate after five minutes. This rule was derived from forging practice, and is sufficiently accurate for practical purposes so long as the temperature is not much higher or lower than forging temperatures. In brass, temperatures are equalized twice as fast, in aluminum six times as fast, and in copper nine times as fast as in steel, but in spite of this fact, the rate at which these metals can be heated, per square foot of furnace hearth area per hour, usually is less than for steel. This is because the absorption of radiant heat is reduced by the smooth surfaces of these metals. Black copper surfaces, however, absorb heat almost as rapidly as oxidized steel surfaces.

2. Forging ingots are heated at the rate of 16 to 20 min. per inch of diameter if of soft steel, and at the rate of 30 to 40 min. per inch of diameter if of high-carbon or alloy steel. Certain steels are very sensitive to temperature changes at "blue brittleness" (around 500° to 800° F.). After they show color all through, they cannot be injured by rapid heating.

3. In ordinary furnaces, steel can be treated at the rate of 25 to 35 lb. per sq. ft. of hearth area per hour for heat treating; at the rate of 60 lb. for annealing; 80 lb. for uniform temperature forging and rolling; and 110 to 130 lb. for work in which 40° to 50° F. temperature difference per inch of thickness of stock is allowable.

4. Steel for drop forging can be heated at the rate of 5 to 10 min. per in. of diam.

**Table 2.—Average Fuel Consumption of Industrial Furnaces**
(Material heated is steel, excepted where otherwise noted)

| Type of Furnace and Operation | Highest Temperature to which Stock is Heated, deg. F. | B.t.u. in Fuel per Pound of Material Heated | Thermal Efficiency, percent |
|---|---|---|---|
| Drop forging or bolt heading furnace, oil fuel | 2300 | 3200 | 11 |
| Bolt heading furnace, coal fired | 2100 | 6000 | 6 |
| Ingot forging furnace, in-and-out type, oil fired | 2200 | 2500 | 14 |
| Ingot forging furnace, in-and-out type, regenerative, gas fired | 2200 | 1900 | 18 |
| Mill-type continuous furnace | 2200 | 1100 | 32 |
| Mill-type continuous furnace, recuperative | 2200 | 900 | 39 |
| Car type annealing furnace for castings | 1600 | 1200 | 21 |
| Annealing furnace for brass | 750 | 750 | 10 |
| Case-hardening (carburizing) furnace | 1700 | 2500 | 11 |
| Heat-treating (quenching) furnace | 1400 | 1150 | 19 |
| Heat-treating (tempering) furnace | 1150 | 800 | 20 |
| Enameling furnace (muffle) | .... | .... | 15 |
| Electric heat-treating furnace | 1500 | 2500 | 80*; 10† |

* Based on electrical energy. † Based on fuel.

5. The time in minutes required for heating high-speed steel tools for hardening to a depth of $d$ inches is $8.5\ (d)^{0.78}$, according to R. C. Deale. This does not include the preheating time, but only the time to heat from 1600° to 2375° F. If the tool is to be hardened throughout, $d = 1/2$ thickness.

Fig. 2 shows the relation between heating time, temperature of bar and furnace temperature. Fig. 3 shows the difference between surface temperature and temperature in the center of the bar per inch of diameter.

The preceding rules are based on furnaces in which enough heat can be liberated to maintain furnace temperature. The data below assist in checking heat liberating capacity.

**Heat Liberation.**—In electrically heated furnaces (resistor ribbon type), working with temperature of 1850° F., the input energy usually is limited to not over 3 kw. per sq. ft. of side wall plus roof, with nickel-chromium resistor material. Somewhat higher rates would be permissible since 1936, on account of better high temperature alloys, but these seldom are used in furnaces for this temperature range.

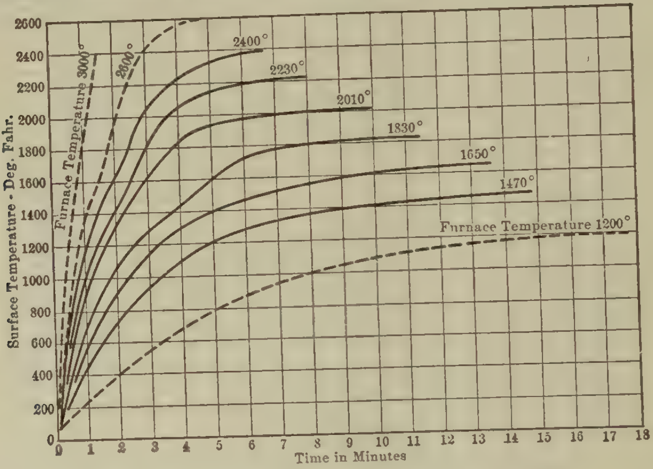

FIG. 2 Heating Time at Various Furnace Temperatures

In furnaces of the combustion type, from 1.5 to 33 B.t.u. can be liberated per cubic foot of combustion space per second. The low value applies to very poor mixing of fuel and air, very imperfect utilization of volume of combustion chamber by flame or gas jets, coarse fuel and cold air. The high value is attained by thorough mixing of fuel and air, perfect utilization of combustion space, very fine atomization or powdering of fuel, and air preheated to 1000° F.

**FLUES AND PORTS.**—If furnaces are properly proportioned, the following port sizes are approximately correct, based on steel as stock to be heated, and for 100 lb. per sq. ft. of hearth per hour. If a smaller quantity is to be heated, the outlet flues should be partly covered.

| | | | |
|---|---|---|---|
| Temperature of furnace, deg. F. | 1400 | 1800 | 2200 |
| Port area, square inches per foot of hearth | 2.0 | 3.5 | 7.3 |

These figures hold for "batch type" furnaces of medium size without recuperation or regeneration. For large and important furnaces, the ports and flues must be calculated on the basis of the flue gas quantity in unit time.

Ports should be located in or near the hearth, never in the roof. They should be so distributed as to cause flow of hot gases to all parts of the furnace. Some of the ports should be in the corners near the door or doors.

## 2. FUELS

**SELECTION OF FUEL OR ENERGY.**—The cost of fuel (or other sources of heat energy) is, in many instances, a very small fraction of the total cost of the manufactured product, and a minor fraction of the heating cost. In such cases, the effect of the form of heat energy on the finished product is most important, and the unit cost of energy is not a matter of consideration. In other cases, the fuel cost is a large item in the total cost, and cheapness of fuel is the deciding factor. The following tabulation, adapted from

Trinks, Industrial Furnaces, Vol. II, Chap. 6 (John Wiley & Sons), may be used in the selection.

The advantages and disadvantages of many of the fuels are denoted by numbers which have these meanings:

**Advantages.**—1. Heating value is high. 2. High flame temperature can be attained. 3. Pipe mains may be small because of 1. 4. Fuel can be burned efficiently. 5. Accurate temperature control is possible. 6. Accurate atmosphere control is possible. 7. Uniform quality of product is possible. 8. Damage to product is slight (few rejections of finished material). 9. Furnace installation is simple. 10. Cost of furnace installation is low. 11. Fuel cost based on heat units is low. 12. Fire risk is low. 13. Plant can be kept clean without difficulty. 14. Cost of upkeep of furnace may be kept low. 15. Labor cost is low (no handling of fuel and ashes). 16. Oxidation or scaling of heated material can be kept low. 17. Furnace is made ready for use by opening valve. 18. No fuel-preparation plant is needed.

**Disadvantages.**—19. Cost, based on heat units, is high. 20. Cost of installing gas-producing plant is high. 21. Installation is complicated. 22. Fuel supply is restricted. 23. Fuel requires handling. 24. Clinkers must be removed. 25. Ashes must be disposed of. 26. Cost of operating labor for gas making is high. 27. Heat value of fuel is low. 28. Large pipe mains are required because of 27.

**NATURAL GAS.**—Excellent fuel for all heating purposes. *Advantages:* 1 to 10 inclusive, and 12 to 18 inclusive. *Disadvantages:* Unreliability of supply, particularly in cold weather. Cost in most localities is rather high and the trend is upward.

**CITY GAS.**—Especially adapted to small-size and medium-size, high-grade, high-temperature work. *Advantages:* 1 to 10 inclusive, and 12 to 18 inclusive. *Disadvantages:* Fuel cost is very high. Gas is available only in cities with gas plants.

**WATER GAS.**—Valuable for high-grade, high-temperature work, in large-scale heating and welding operations, especially where high flame temperature must be produced without preheating the air. *Advantages:* 3 to 8 inclusive, 12, 16. Very high flame temperatures can be attained. Heat value is moderately high. Furnace room can be kept clean without difficulty. Items 5, 6, 7, 8, 16 depend on correct operation of the gas-making plant. *Disadvantages:* 19 (in small plants), 20, 21, 22 (coke or anthracite is required), 23, 24, 25, 26. Gas is odorless, but poisonous, and is a menace to life and health. Danger of backfiring, due to rapid flame propagation.

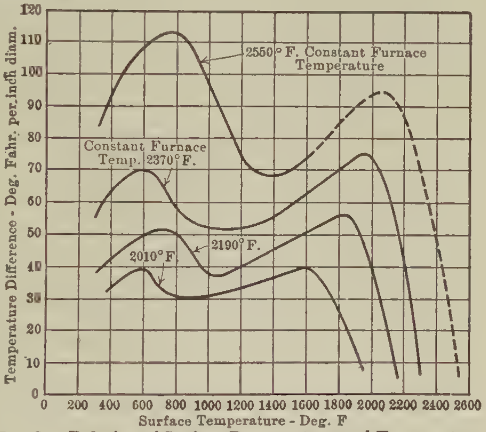

FIG. 3. Relation of Surface Temperature and Temperature at Center of Bar

**RAW PRODUCER GAS.**—The most commonly-used fuel for large furnaces in which close regulation of temperature and of furnace atmosphere is not required. *Advantages:* Cost on basis of heat units is low. Gas can be made wherever bituminous coal is available. Gas-making operation is comparatively simple. Gas burns with luminous flame. Installation cost is lowest of any gas-making equipment. *Disadvantages:* 22 (screened gas coal, for good results), 23, 24, 25, 27, 28 (brick-lined flues required). Furnace must be located close to producers, because not only do tarry vapors condense, but sensible heat of gas is lost in long pipe lines. Gas ducts become clogged with soot and tar, and must be burned out at regular intervals. Difficult to maintain constant quality and composition of gas. Difficult to keep intelligent workmen in dirty gas house. Steam plant required for furnishing steam to producers. Flow of gas can be measured only with difficulty, because of tar and soot deposits. Difficult manually, and practically impossible automatically, to control temperature and furnace atmosphere. Sulphur exists in the products of combustion.

**COLD CLEAN PRODUCER GAS.**—Efficient use of this gas is limited to about 2000° F., unless regenerative or recuperative furnaces are used. Useful for installations of large size only. *Advantages:* 4, 5, 6, 7, 8, 16. Temperature of combustion is low, which is desirable for furnaces with less than 1600° F. temperature. Long life of furnace, if properly designed and operated. Reliability of fuel supply. Stability of fuel cost. Gas burns without smoke, even if atmosphere is reducing. Fuel cost is comparatively low, in large installations. Gas can be distributed to many small furnaces. Advantages 5, 6, 7, 8, 16 depend on correct operation of the gas-making plant. *Disadvantages:* 20, 22 (screened gas coal) 23, 24, 25, 26, 27, 28. Installation is more complicated than with oil, city gas, or natural gas. Tar made in producers must be disposed of. Good furnace operation depends on zeal and skill on the part of the gas house operators.

**HEAVY FUEL OIL.**—Standard fuel for medium-size furnaces, except where convenient and cheap gaseous fuels are available. It is very viscous (7 to 12 Bé., 500 Saybolt viscosity at 100° F.) and usually contains sediment and dirt. *Advantages:* 1, 2, 10, 11. Convenience of handling and of storage. Simplicity of installation. Oil can be had anywhere. *Disadvantages:* Fluctuation of fuel cost. Furnace temperature and atmosphere difficult to control automatically. Possibility of damaging heated product, caused by temperature and atmosphere variations, or by impingement of flame. Fire hazards are high. Cost of labor is high because of constant attention required for burners and auxiliary equipment. Life of container boxes is likely to be short as compared with non-oxidizing gas fuel. Preheated air is required, if reducing atmosphere is to be obtained without excessive smoke. Recirculating system, heating and careful filtering of oil required. Life of furnace brickwork is shorter than for gaseous fuels.

**LIGHT FUEL OIL** usually is a distillate, and is quite fluid (24 to 40 Bé. gravity). It usually is free from sediment or dirt. *Advantages:* Same as for heavy fuel oil. Even more convenient to handle. Does not require heating, nor such great care in filtering. Installation is especially simple and inexpensive. Can be used in furnaces having many small burners; hence it is suitable for work requiring uniformity of product. Temperature can be controlled automatically. More suitable for fine work than heavy oil. *Disadvantages:* Fuel cost fluctuates, and usually is about 25% higher than for heavy oil, while the heat value per gallon is lower. Supply is limited. Atmosphere control is much more difficult than with gaseous fuels; hence there is greater danger of damage to product. Danger of burning product by impingement of flame (but less than with heavy oil). Fire hazard is high. Labor cost is higher than for gaseous fuels, because more attention is required for burners and auxiliary equipment. Life of container boxes and life of furnace brickwork is shorter than for gaseous fuels. Not so suitable as heavy fuel oil for furnaces requiring long, luminous flame, *e.g.*, open hearth.

Specifications for the six grades of fuel oil, standardized by the American Society for Testing Materials, are given in Table 3.

**COAL ON GRATES OR MECHANICAL STOKERS** can be used efficiently only for fairly continuous service and large furnaces. *Advantages:* Fuel cost is low. First cost of installation is low. Total operating cost is low if reliable heaters are employed, and suitable furnaces are used. Fuel supply is reliable. Fuel price is stable. *Disadvantages:* 23, 25. Firebox requires frequent repair. Floor space required is large. Competent firemen are needed, and difficult to keep. Scaling loss is high. Control of furnace temperature and atmosphere is difficult. A stack is required. Black smoke is produced if a reducing atmosphere is wanted. Sulphur exists in the products of combustion.

**COKE ON THE GRATE.**—Products of combustion are free from water vapor; hence they are very useful for heating drying-chambers. Gives high temperature and reducing atmosphere in open fire, without smoke; hence, is used for blacksmith forge fires. In all other respects, similar to coal on the grate.

**POWDERED COAL.**—At present (1937), useful in large furnaces only, because satisfactory small individual pulverizers have not yet been developed for powdering coal to sufficient fineness for combustion in industrial furnaces of average size.

*Advantages:* No labor is necessary to bring fuel to, or inject it into, the furnace. Control of temperature and control of furnace atmosphere are easy, as compared with coal on the grate. Fuel cost is low in large plants. *Disadvantages:* Flame temperature * is high. Combustion chamber difficult to maintain. Cost of pulverizing and coal equipment is high. Fine ash and slag must be disposed of. Recuperators or regenerators cannot be used. Explosion hazard in pulverizing plant. Ash deposits may spoil heated product, or floating in air of shop may damage machinery. Difficult to maintain an even temperature over a large hearth. Control of furnace atmosphere is difficult, as compared to gaseous fuels.

**ELECTRICAL ENERGY** is favored for many heating operations. *Advantages:* 5 to 9, 12 to 16, 18. Installation has great convenience and flexibility, including possibility of placing furnace anywhere. No products of combustion in furnace workroom. No piping or stacks required. Furnaces can be placed directly in the production line. Life of furnaces is long. Operation is simple. There is no auxiliary equipment of the kind that requires attention. Inconvenience to workmen in plant is a minimum. Furnace is made ready for use by closing a switch. Heat can be developed in the furnace hearth. *Disadvantages:* Cost, based on heat units, is very high. First cost of furnace and control is high. Furnace heats slowly from cold start. Cost of heat lost by storage in furnace walls is high in case of intermittent operation. Cost of repair is high. Delay results if resistors burn out. " Service " or " demand " charge of public utility is high in case of intermittent operation. Not suitable where high-temperature heating by convection is required.

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* Whether high flame temperature is an advantage or disadvantage depends on the class of work. For low-temperature work, such as annealing, it is difficult with the rich fuels to keep the temperature low enough. For high-temperature work, such as welding, it is difficult to obtain sufficiently high temperature with lean fuels unless air and fuel are preheated.

**TAR AND COKE-OVEN GAS,** as a rule, are available only in steel plants. The former is similar to heavy oil, but more difficult to handle. Coke-oven gas is similar to city gas, but much cheaper.

**GASEOUS VS. LIQUID FUELS.**—Gaseous fuels are most convenient, particularly if drawn from large sources of supply (city gas, natural gas). They allow regulation within wide limits, easy control of air supply, scattered small burners for the sake of temperature control, and good control of furnace atmosphere. Regulation is more difficult if the furnace unit requires a large share of the capacity of the gas-generating plant.

Liquid fuels are not quite as convenient with regard to regulation as gaseous fuels, but they can be stored more conveniently. Gasoline and kerosene are used for very small heating operations only, while fuel oil is (in 1937) the standard fuel for medium-scale operations. For small-scale heating, heavy fuel oil is not suitable, because it must be

**Table 3.—Detailed Requirements for Fuel Oils**[a]

(A.S.T.M. Specifications)

Fuel oils are to be hydrocarbon oils free from acid, grit, fibrous or other foreign matter likely to clog or injure burner or valves.

| Grade | Flash Point, deg. F. | | Pour Point, deg. F. | Water and Sediment, per-cent | Carbon Residue, per-cent | Ash, per-cent | Distillation Temperatures, deg. F. | | | | | Viscosity, seconds | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | 10% Point | 90% Point | | End Point | | Saybolt Universal, at 100° F. | | Saybolt Furol, at 122° F. | |
| | Min. | Max. | Max. | Max. | Max. | Max. | Max. | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. |
| No. 1. A distillate oil for use in burners requiring a volatile fuel. | 100 or legal | 150 | 15[b] | 0.05 | 0.02 | ..... | 420 | ..... | ..... | 600 | ..... | ..... | ..... | ..... | ..... |
| No. 2. A distillate oil for use in burners requiring a moderately volatile fuel. | 110 or legal | 190 | 15[b] | 0.05 | 0.05 | ..... | 440 | 620 | ..... | ..... | 600 | ..... | ..... | ..... | ..... |
| No. 3. A distillate oil for use in burners requiring a low-viscosity fuel. | 110 or legal | 200 | 15[b] | 0.10 | 0.15 | .... | ..... | ..... | 620[d] | ..... | ..... | 70 | ..... | ..... | ..... |
| No. 4. An oil for use in burners requiring low-viscosity fuel. | 150 | ..... | [c] | 1.0 | ..... | 0.1 | ..... | ..... | ..... | ..... | ..... | 500 | 70[e] | ..... | ..... |
| No. 5. An oil for use in burners equipped with preheaters permitting a medium-viscosity fuel. | 150 | ..... | ..... | 1.0 | ..... | 0.15 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 100 | 25 |
| No. 6. An oil for use in burners equipped with preheaters permitting a high-viscosity fuel. | 150 | ..... | ..... | 2.0[f] | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 300 | 100[g] |

[a] For heat-treating and other special uses, sulphur may be specified:

| Grade of fuel oil........ | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Sulphur, percent, max... | 0.5 | 0.5 | 0.75 | 1.25 | no limit | no limit |

[b] Lower or higher pour points may be specified, as required, but not below 0° F. under any conditions. [c] Lower or higher pour points may be specified, as required, but not lower than 15° F. under any conditions. [d] Shall be waived when carbon residue > 0.07% and < 0.15%. [e] Shall be waived when carbon residue > 1.0%. [f] A deduction in quantity shall be made for all water and sediment in excess of 1.0%. [g] Shall be waived when carbon residue is 4% or more.

heated and pumped, and because of the difficulty of maintaining uniformity of operation. With clean gas-oil, however, very uniform heating can be effected.

The relative cost of fuels is shown in Fig. 4. The broken line curve for city gas applies in cities having special rates for gas used for industrial purposes.

## 3. BURNERS

**OIL BURNERS.**—Oil burners atomize the oil, and should thoroughly mix it with air in definite proportions. They should maintain uniformly fine atomization with various rates of flow and varying viscosity of oil, and also should maintain the correct air-to-oil ratio under all conditions. These goals, practically, never are attained.

Mechanical atomizers which atomize by oil pressure only, as used for firing boilers, and rotary burners, as used in house-heating furnaces, seldom are applied to industrial furnaces, because they have insufficient turn-down range; also they give trouble from coke formation in the burner tip, when exposed to radiation from the hot furnace while the oil flow is shut off. The oil burners actually used for industrial furnaces, of which there are innumerable makes, fall into three general classes or types. The characteristics of these are discussed below.

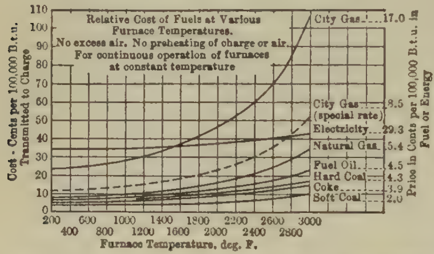

Fig. 4. Relative Cost of Fuels for Heating Furnaces

1. High-pressure burners, in which steam or compressed air at 30 to 150 lb. per sq. in. is used as the atomizing agent. An example is shown in Fig. 5.

2. Low-pressure burners, in which the atomization is effected by air supplied from a displacement- or turbo-blower at $^3/_4$ to 5 lb. per sq. in. Examples are shown in Fig. 6, *a* and *b*. In some burners of this kind, the oil pressure need be but slight, while in others, the pressure must be as high as in the following class of burners. The oil pressure helps atomization.

3. Fan-blast burners, in which atomization is effected chiefly by the pressure of the oil itself, from 30 to 150 lb. per sq. in. The energy of the air which is supplied by a fan at pressures of 4 oz. to 1 lb. per sq. in., is used merely for mixing the oil spray with the air, with some slight auxiliary effect in atomization. See Fig. 7.

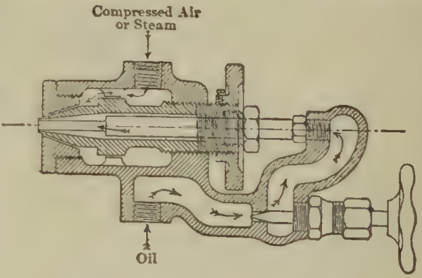

Fig. 5. High-pressure Oil Burner

Everything else being equal, higher air or steam pressures produce better atomization than low pressures, and require less of the atomizing agent under pressure. The heavier or more viscous the oil, the more difficult is atomization, and the higher is the pressure required. Light oil can be burned satisfactorily with fan-blast burners, but for the very heavy residues high-pressure burners are practically essential. Steam is preferable to compressed air for atomizing very heavy tar or oil, although it depresses the flame temperature slightly. The heavy oils must be heated to between 180° and 250° F. before atomization, and recirculated in order to maintain high temperature at the burner. If both oil and combustion air are sufficiently preheated, vaporization largely takes the place of atomization, and good, quick combustion can be obtained with low air pressure. However, cracking and coke deposition, or vaporization and sputtering

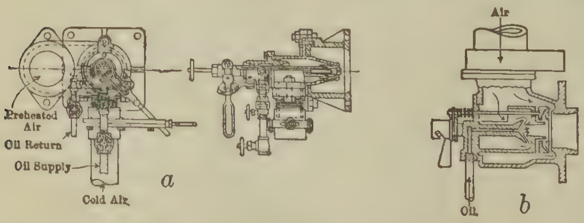

Fig. 6. Low-pressure Oil Burners

occur if the oil is heated too highly. High-pressure burners produce a long flame of comparatively small thickness; low-pressure burners produce a shorter, " bushy " flame. Flat instead of cone-shaped flames can be produced by suitable burners. Steam makes a longer flame than compressed air.

In fan-blast burners, atomization is effected by discharging high-pressure oil in a whirling spray through a small orifice. Fig. 8 shows an atomizing tip. In most of these burners, the range of turn-down is comparatively small, if satisfactory atomization is to be maintained, although in the best designs a turn-down of 5 to 1 is obtained. The very small openings required, except for very large units, clog easily unless gas oil is used, or efficient oil filters are provided in the case of heavy oil, and carbonization is entirely prevented.

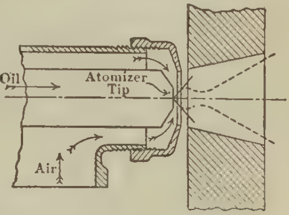

Fig. 7. Fan-blast Oil Burner

Steam atomizers use $^1/_3$ lb. to 1 lb. of steam per pound of oil; high-pressure burners using air require 10% to 15% of the total combustion air to be supplied under pressure; low-pressure burners require 30% to 50% to be supplied by the fan. In fan-blast burners, from 70% to 100% of the air is supplied by the fan.

For use with preheated air, the oil tube must be surrounded by a space through which cold air flows (as at 2 in Fig. 8); otherwise, carbonization and coke deposition will occur. Even with cold air, this takes place if oil is allowed to stand in the tip when the burner is shut down but exposed to radiation from the hot furnace interior. When shutting down a burner, all remaining oil should be blown out of it by air under pressure. The burner of Fig. 6*a* is designed also for preheated air.

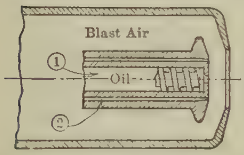

Fig. 8. Atomizing Tip

The proportions of the opening in the burner block are important for satisfactory operation. The opening should be conical or flared, diverging from the burner toward the furnace interior with a total included angle of 15 deg. to 35 deg. For cold air, the diameter, in inches, at the small end should be 1.0 to 1.3 × $\sqrt{\text{gallons per hour burned}}$. If the air is preheated, the diameter should be 40% to 50% larger. The length of the cone should be usually not less than 1 $^3/_4$ times the smallest diameter.

Where there is no opening around the burner, a hole should be provided in the furnace wall for observing the flame and for inserting oily waste when lighting. Blowing-out of the flame due to pulsations, or difficulty in lighting, may be obviated by providing a refractory tip extending into the flame path at the large end of the burner-block opening, or by setting a piece of firebrick in the opening so that the flame plays upon it.

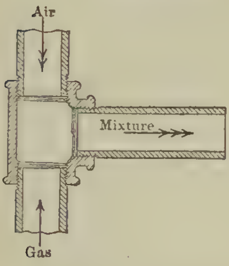

Fig. 9. Tee-mixing Burner for Gas

Up to about 1930, it was impossible to burn heavy oil satisfactorily in furnaces with many small burners and to produce consistently a product equal to that obtained with gas firing. This was because silt and other foreign matter in the oil clogged the small openings in the burners, and even with constant attention, produced constant fluctuations in furnace temperature and atmosphere. Improvements in oil filters have made it possible to obviate this difficulty, and with light oil as fuel and provision for preventing carbon deposits, results practically equal to heating with gas are obtained.

**GAS BURNERS.**—The functions which gas burners should perform are: Proportioning the gas and air; producing sufficiently rapid, but not too rapid, mixing; and giving direction to the flame. In only a few types is the proportioning effected satisfactorily over any considerable range of turn-down, unless special equipment is used, such as interconnected flow meters.

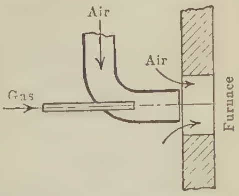

Fig. 10. Burner with Gas and Part of Air under Pressure

The simplest form of gas burner is the tee-mixer, Fig. 9. Proportioning requires the adjustment of two valves, one in the air pipe and one in the gas pipe. Sometimes the valves are mechanically interconnected. The completeness of mixing (which controls luminosity of flame and speed of combustion) cannot be varied. In the burner of Fig. 10,

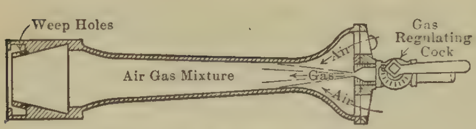

Fig. 11. Inspirator Gas Burner

however, this can be done by moving the gas pipe in or out with reference to the furnace wall, and to some extent by varying the opening between the tube and the wall. Both of these burners have a wide turndown range without danger of backfiring. Other advantages are simplicity, and wide range of adjustment of furnace atmosphere.

A widely-used type is the inspirator burner. One form is shown in Fig. 11. In this form, called the high-pressure type, the gas under pressure induces air from the atmosphere. In another form, the low-pressure type, air is supplied from a fan or blower under a small amount of pressure, and induces the gas which is maintained at atmospheric pressure by a reducing valve or regulator. The inspirator burner has the advantage of maintaining a practically constant gas-to-air ratio over a considerable range. If turned down to (usually) about 1/3 or 1/4 of its maximum capacity, backfiring occurs. The air-to-fuel ratio is quite sensitive to variations in furnace pressure. The inspirator burner produces almost perfect mixing and rapid combustion, with a very hot, nearly non-luminous "Bunsen" flame, which is an advantage in some cases, but a disadvantage in others. Several manufacturers now make good inspirator burners.

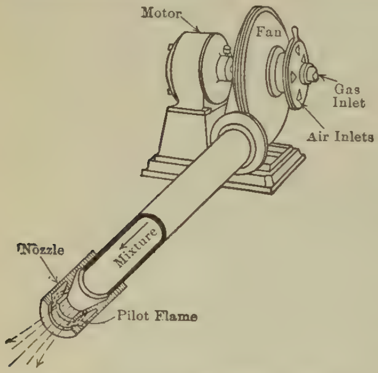

Fig. 12. Pre-mix Fan and Burner

The high-pressure type of burner has the advantage of requiring only one pipe and one fluid under pressure; it is, therefore, especially suited for installations having many small furnaces. No fan is needed, and adjustment is effected by a single valve burner. For city gas, or gases of lower calorific value, it can induce enough air without difficulty. For natural gas, the gas pressure must be quite high (in some cases up to 30 lb. per sq. in.) in order to induce the 10 or 11 volumes of air required per volume of gas. The high-pressure inspirator is more sensitive to change of furnace pressure than is the low-pressure type, is noisy in operation, and does not permit adjustment of the flame length. Inspirator burners have not been used with preheated air in the U. S., because of the danger of explosions.

In another system, the gas is mixed with about 1/4 of the total quantity of air required, put under pressure by a rotary blower, and passed to inspirator burners wherein it induces the remainder of the air required for combustion. The mixture in the blower and pipes is non-explosive. Pressure regulators are used to maintain the correct proportions of air to gas.

In the "pre-mix" burner, both air and gas are sucked into the inlet of a fan, mixed therein and supplied through a pipe to the burner nozzle. See Fig. 12. The nozzle has small holes for maintaining a pilot flame and preventing the flame from blowing away from the nozzle. The proportioning can be changed by moving the air shutter. The pre-mix burner also gives a Bunsen flame, although some luminosity can be produced by supplying only part of the air through the fan and inducing the rest around the nozzle. The danger of backfiring is even greater than with the inspirator burner, and explosion flaps or safety discs must be provided in the pipe line to prevent serious accidents.

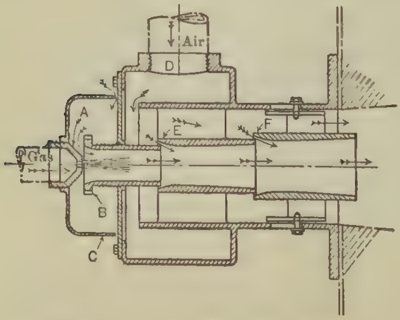

Fig. 13. Luminous Flame Burner

Luminous flames (such as are produced by the so-called diffusion burners) transmit heat more rapidly than do non-luminous gases, and they also reduce scaling of metallic objects being heated, by maintaining a reducing furnace atmosphere. An example of a luminous-flame burner is shown in Fig. 13. Gas under pressure issues from nozzle or "spud" *A*, while air from a fan enters at *D*.

A part of the air is induced into tube *E*, another part into *F*, and the rest flows outside of tube *F*. When cap *C* is screwed to the left, some air from the atmosphere is induced into tube *B*. The positions of tubes *B* and *F* are adjustable, while *E* is stationary. When *B* and *F* are moved closer to *E*, most of the air passes outside of *F*, and the flame becomes long and luminous. When *B* and *F* are moved away from *E*, the flame becomes shorter and less luminous. The turn-down range is 20 to 1.

Raw producer gas, because of the tar and soot which it contains, is difficult to handle in ordinary burners, and usually is sent into the furnace through openings in the firebrick wall. A special burner designed for producer gas is shown in Fig. 14. Air at about 1 lb. pressure induces the gas into the throat of the ignition tube, effecting excellent mixing. The relative proportions of the mixing and inducing actions can be varied by turning the handle at the back. The valve is designed to insure ease of opening even when stuck shut by tar deposits.

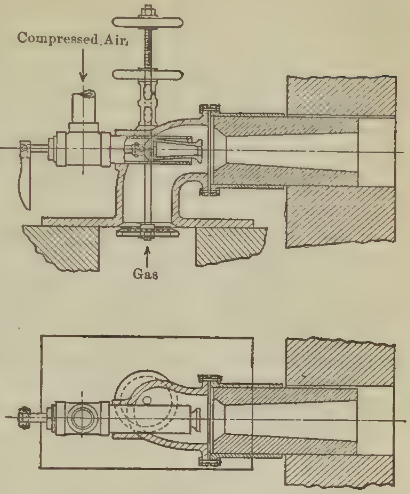

Fig. 14. Burner for Producer Gas

## 4. SELECTION OF FURNACES

**TYPES OF FURNACES.**—The type of furnace which is best adapted to carry out a given heating operation depends on many factors, as: Shape and weight of stock; number of times the process is to be repeated (job shop, or manufacturing establishment); cycle of heating (and cooling); highest temperature; uniformity requirements for heating; fuel which is available; capital which is available for investment. There are between 90 and 100 different types of furnace in use.

For general blacksmith work, the blacksmith's coal or coke fire is still found; more often gas- or oil-heated furnaces or fires as shown in Figs. 15, 16 and 17 are preferred. For more special work the types shown in Figs. 18 or 19 are common. Coal-fired heating furnaces for general forge work are built as in Fig. 20 with an end slot, or else with a side door. Heat-treating furnaces often are under-fired, for the sake of having a hot hearth, as in

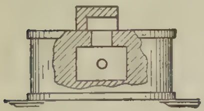

Fig. 15

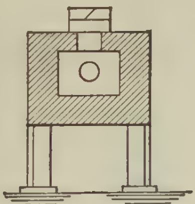

Fig. 16

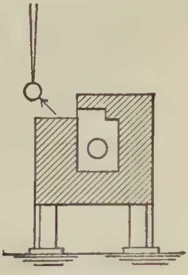

Fig. 17

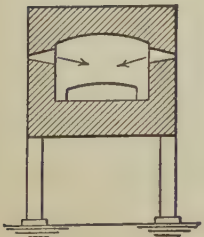

Fig. 18

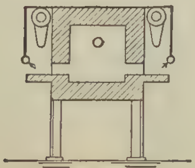

Fig. 19

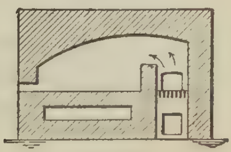

Fig. 20

Fig. 21. Pack hardening (carburizing) furnaces often are made under-fired (Fig. 22) with perforated hearth to secure uniformity of temperature. Tool-hardening furnaces usually have a preheating chamber above the main heating chamber, Fig. 23. Furnaces for heavy material are made of the car type, either side-fired as in Fig. 24, or over-fired as in Fig. 25. The side-fired type can carry a higher temperature than the over-fired type, which latter is commonly used for annealing or for heat treating. The car type has a cold hearth, hence the charge must be blocked up. Automatic conveyor furnaces, Fig. 26, are used for quantity heating of light material. The same is true of revolving-drum furnaces, Fig. 27, which also are used for gas carburizing. Bright annealing furnaces, Fig. 28, are used to anneal copper or similar material in a neutral atmosphere

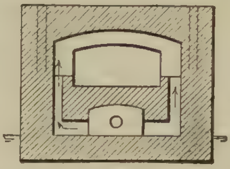
Fig. 21

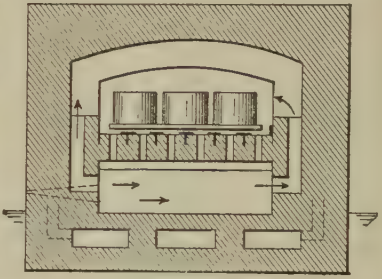
Fig. 22

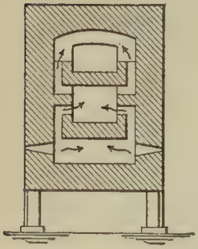
Fig. 23

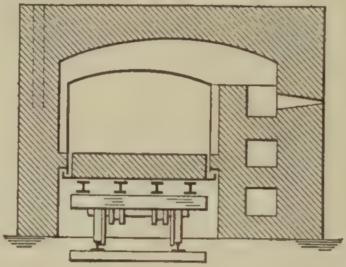
Fig. 24

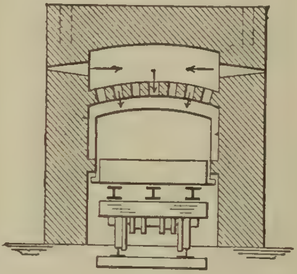
Fig. 25

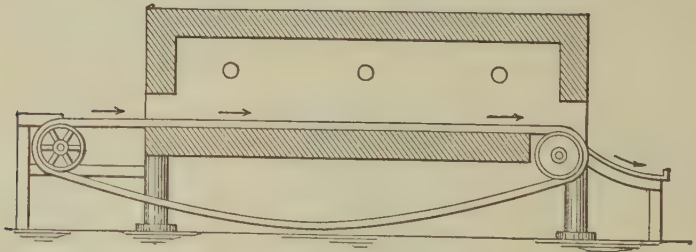
Fig. 26

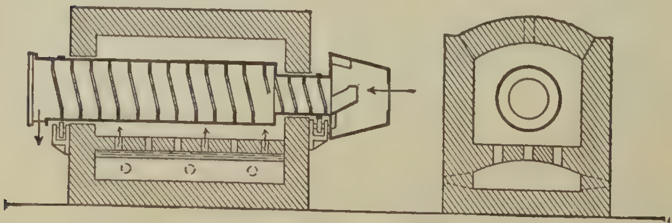
Fig. 27 Rotary Cylindrical Furnace

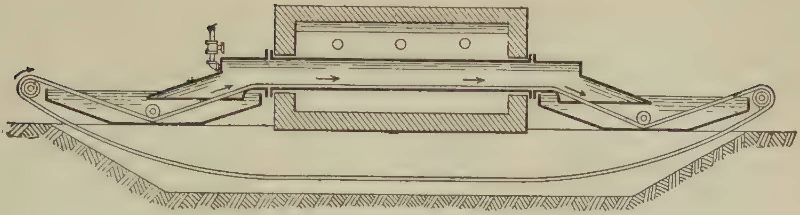

Fig. 28. Bright Annealing Furnace

Elevator-type furnaces, Fig. 29, are used for annealing; they avoid air leakage and oxidation. A portable type of furnace is the self-heated annealing cover, Fig. 30, in which combustion occurs within the tubes and the latter radiate heat to the charge. Tubes sometimes are placed under the top of the cover as well as at the sides. In operation, the material to be annealed is placed on the bottom part, which is separate from the cover; the latter is lifted by a crane and placed over the charge, fitting into a seal at the edges of the bottom part. In many cases, the air is drawn out and the interior filled with inert gases to prevent oxidation.

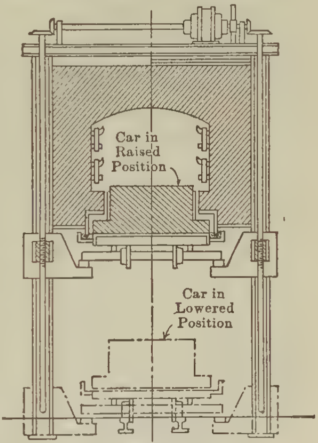

Fig. 29. Elevator Type Furnace

The bell-type electric furnace, Fig. 31, operates on the same principle, but is used for annealing coiled strip or wire. The "retort" or bell has a well in the center into which the central heating element, attached to the cover, fits when the cover is set in place. Inert gas is admitted into the retort through the pipe shown at the bottom. In larger furnaces of this type, a motor-driven fan is provided inside the retort at the bottom, for circulating the inert gas and effecting rapid cooling after the cover has been removed.

Pit furnaces, Fig. 32, are used for hardening large dies and long shafts. Rotating hearth furnaces, Fig. 33, are used to handle material which cannot be well moved through a furnace on chains or by pushers. Intermittent-conveyor, or rocker-bar, furnaces, Fig. 34, serve the same purpose. Their maintenance is difficult. Cyanide bath furnaces are very common for hardening. Pot furnaces, Fig. 35, heated either electrically or by flame, are used as lead baths or salt baths for heat treating to prevent oxidation; or with compounds containing sodium cyanide and other salts for liquid carburizing. Cyanide fumes are very poisonous, and hoods with stacks must be provided to carry them out of the furnace-room. In electrically-heated salt baths, the salt itself acts as the resistor in which the heat is

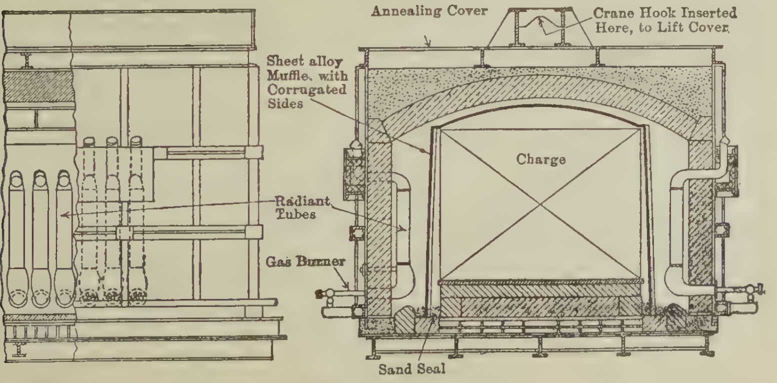
Annealing Cover
Crane Hook Inserted Here, to Lift Cover
Sheet alloy Muffle, with Corrugated Sides
Charge
Radiant Tubes
Gas Burner
Sand Seal

Fig. 30. Portable Annealing Furnace

developed; current flows either from one electrode to the pot, or between two electrodes which dip into the bath. In the latter design, the current produces a circulating motion of the molten salt, which helps to maintain uniformity of temperature.

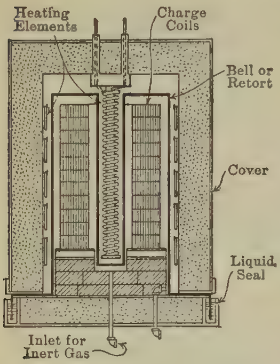

FIG. 31. Bell Type Furnace

Another type of furnace, much used for the heat-treating of many small pieces, is the convection-type furnace. One design is shown in Fig. 36. In this, heating is effected entirely by convection, by drawing the heated air through the charge. The convection furnace also may be fuel-fired, and combustion products or inert gases may be circulated instead of air. Close temperature control is obtained. High-temperature furnaces usually are unsuitable for low-temperature work, and *vice versa*, but a combination furnace has been designed in which high-temperature heating (up to 2400° F.) is effected, chiefly by radiation, while by re-circulating the gases, low-temperature heating (below 1200° F. with accurate control) is effected by convection.

**ELECTRIC FURNACES.**—Electric furnaces are so different from the usual combustion types that building them in the shop with the aid of an ordinary bricklayer seldom is attempted. With very few exceptions, they are purchased complete from manufacturers specializing in this work. For that reason, details of heating elements are not given here. Electric furnaces with resistors of nickel-chromium-iron or nickel-chromium alloy are excellent for heat-treating at 1200° to 1400° F., and for annealing at 1600° F. Above 1800° F. furnace temperature, such resistors burn out in short order. For forging (2200° F.) or for hardening high-speed steel (up to 2400° F.), bars of silicon-carbide or special alloys must be used as resistors. Two of these special alloys are: Kanthal, which contains up to 14% aluminum, up to 30% chromium, up to 6%

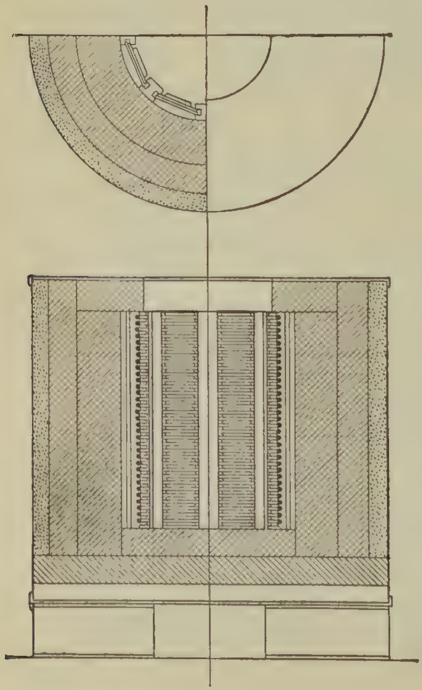

FIG. 32

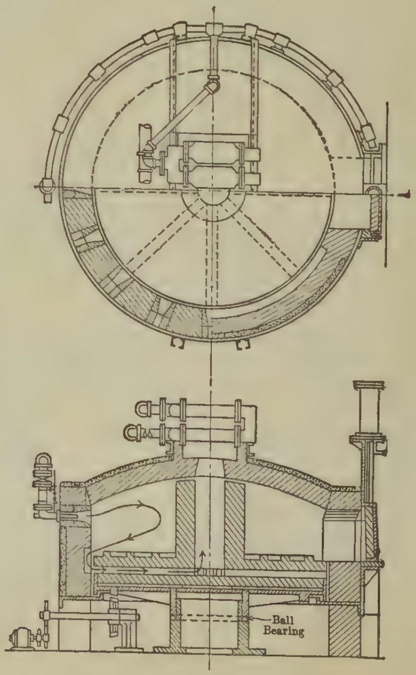

FIG. 33. Rotating Hearth Furnace

cobalt, and the remainder iron; Smith No. 10 alloy, which contains 37 $^{1}/_{2}$% chromium, 7 $^{1}/_{2}$% aluminum, and the remainder iron.

A resistor brought out in 1924, consisting of a metallic core imbedded in a ceramic material, did not prove successful, because at high temperatures the metal slowly vaporized and escaped through the porous ceramic material. However, a newly introduced composite resistor, called Stratite, is claimed to operate at 3000° F. over long periods. It consists of a core of molybdenum, surrounded by oxides which are not reduced by the action of the metallic vapor, the whole being cased in a material which becomes glass-like at high temperatures and prevents escape of the metallic vapor.

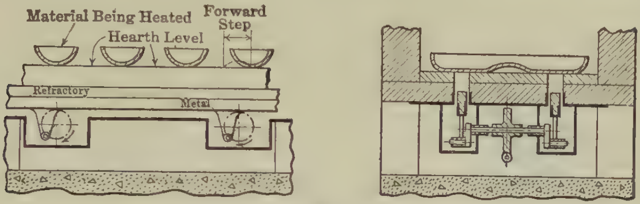

Fig. 34. Intermittent Conveyor Furnace

The silicon-carbide heating elements are almost unlimited as to temperature, but they gradually increase in resistance, to the extent of 30% to 80% after 400 hours operation. The expansion of the silica formed by internal oxidation produces a sudden increase of resistance when heating up from cold, which makes the oxidized bars practically unusable unless the furnace is maintained continuously at a temperature above 600° F. It is claimed that improvements in manufacture have eliminated this difficulty in 1936, and the bars last much longer. The difficulties in manufacture of the high-temperature alloy resistors have not been entirely overcome.

Because there are no products of combustion, the electric furnace eliminates the stack loss. On the other hand, heat is lost in the terminals, which must pass through the walls. If the terminals are made large in diameter, the conduction loss is great; if they are made small, heat is developed in and lost from the terminals themselves, with consequent danger of burning out. An advantage of the electric furnace is that heat can be developed in the hearth, insuring uniform heating throughout the charge.

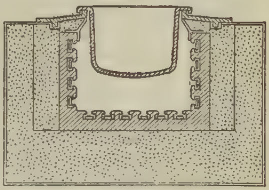

Fig. 35. Lead Pot Furnace

Electric heating can be effected without a furnace enclosure, as in the machines for heating rivets, or for heating bars for forging or upsetting. A heavy current at low voltage is passed through the bar between electrodes which grip the ends, or one end and an intermediate area, the bar acting as resistor. The process is to some extent self-regulating because of the radiation and the rapid increase of resistance of iron with temperature. Heating also can be effected by induction, as in a machine for hardening the bearing surface of automobile crankshafts, wherein a high-frequency, low-voltage current is passed through inductor blocks which closely surround but do not actually touch the bearing surfaces. The current thereby induced in the surface layer of the metal rapidly heats it to the required temperature; the current then is shut off and the heated surface is quenched by a spray from a water jacket within the inductor block. See pp. 18–22 to 18–28 for additional data on electric furnaces.

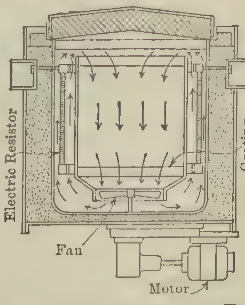

Fig. 36. Convection Type Electric Furnace

**CONVEYING OR MATERIAL HANDLING MEANS.**—Saving of labor is an important advantage of the continuous furnace in which the material to be heated is conveyed through the furnace by mechanical means. The suitability of the various conveying devices depends largely on the temperature. Up to 1200° or even 1400° F., the chain conveyor (Fig. 26) with solid links or woven wire mesh is satisfactory. Above this, and up to 1800° F., the rocker-bar (Fig. 34) or roller-hearth conveyors are suitable; the drum-type furnace also may be used for handling small pieces. For 2100° F. or over, the material either must be picked up and set down by tongs or charging machines intermittently inserted into the furnace, or must be pushed over the hearth. (See Trinks, Industrial Furnaces, Vol. II.)

**TEMPERATURE CONTROL.**—Automatic devices often are installed for controlling the furnace temperature; usually for keeping it constant, but occasionally for varying it with time according to a set program. The measuring device (see Governing, Regulation and Control, p. 10–03) usually is a thermocouple, which actuates the pointer of a depressor-bar millivoltmeter. When the pointer deviates from the desired temperature, contact is made to actuate a relay and contactors which, in electric furnaces, throw resistors into or out of circuit; or in fuel-fired furnaces operate a motor or solenoid to open or close the fuel and air valves as required. One form of temperature regulator uses air from a blower as the motive power, and thus avoids the use of solenoids.

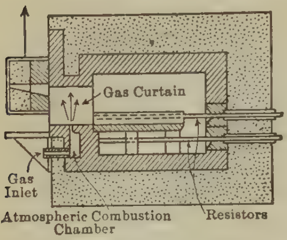

Fig. 37. Tool Hardening Furnace with Gas Curtain

The objections to present temperature-controllers for small furnaces are the first cost, the upkeep cost (especially in high-temperature furnaces), and the supervision which they require. If an inexpensive but rugged controlling device could be put on the market, many more furnaces would be automatically controlled than at present.

In low-temperature furnaces or ovens, gas or vapor thermometers or bi-metallic thermostats are used. With the thermometers, the ordinary contact type of instrument is used for simplicity, instead of the depressor-bar type. In automatic control of oil-burning furnaces operating above 800° F., the fuel and air never are shut off entirely, because the oil remaining in the burner tip would coke, and clog the opening. "All-on or all-off" control in this case means maximum-minimum control. Recently, proportional or "anticipating" controllers have been put on the market, in which the fuel and air supply is gradually adjusted over the range between the minimum and maximum.

**CONTROLLED ATMOSPHERE IN FURNACES.**—Oxidation or scaling of ferrous metals while being heated occurs if an atmosphere of air, carbon dioxide, or water vapor exists in the furnace, and to an even greater extent if actual combustion takes place in the gases in contact with the metal. The scaling grows rapidly with the temperature. It can be reduced or practically avoided by maintaining in the furnace a reducing atmosphere containing either a considerable amount of carbon monoxide (at least 8% is required for entire prevention of scaling), or a haze of carbon particles produced by cracking hydrocarbons. The former requires merely an excess of fuel. In direct-fired furnaces using natural gas, coke-oven gas, or coal gas, luminous-flame burners may be used to envelop the charge on the hearth in a blanket of fuel gas, which cracks and forms carbon particles; or secondary combustion devices, or "smoke burners" may be provided. In electric furnaces or muffle furnaces, a non-oxidizing atmosphere may be formed by supplying either hydrogen or the products of thermal decomposition of ammonia, or of natural gas, city gas, butane, or propane. These gases may be decomposed (the heat being applied by partial combustion) in a separate producer, or in the furnace itself. In small electric furnaces for tool hardening and similar work, where the door must remain partly open most of the time, a non-oxidizing atmosphere is maintained in the furnace by providing in front of the door a gas-flame curtain, as shown in Fig. 37. The oxygen in any air which succeeds in entering at the door is taken up by the gas, and an atmosphere containing CO and hydrogen, and sometimes free carbon and undecomposed hydrocarbons, is formed.

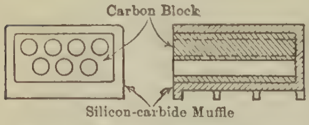

Fig. 38. Electric Furnace Muffle

For annealing copper or brass, an atmosphere of steam is used in furnaces such as those in Figs. 28 and 29. For heating high-carbon steel parts, such as tools, the atmosphere may be made not only non-scaling but non-decarburizing. An example is the electric-furnace muffle shown in Fig. 38, which consists of a hollow carbon block. Any oxygen entering at the open end reacts with the glowing carbon to form CO in high concentration. For brazing, an atmosphere of hydrogen is used; the parts are clamped together, coated with a paint containing finely divided copper or brass, treated with a flux, and placed in the brazing furnace in a hydrogen atmosphere. The copper is drawn between the contact surfaces by capillary attraction. Instead of the copper paint, a wire of special oxygen-free copper sometimes is used.

In addition to those mentioned, atmospheres of other gases are used for special purposes. For nitriding, the atmosphere consists of ammonia; for Chapmanizing, the ammonia is bubbled through a molten salt bath in which the pieces are immersed. For gas carburizing, a carbon-rich gas, *e.g.*, propane, is introduced into the furnace muffle; and in a recent development, carburizing and nitriding are combined by using a mixture of hydrocarbon gas and ammonia. A carburizing atmosphere also may be produced by allowing liquid hydrocarbons to drip slowly into the furnace from above.

Some of the devices used for maintaining a constant composition of furnace atmosphere by keeping constant the ratio of fuel to air, regardless of changes in the fuel rate, are: Interconnected fuel and air valves (only effective when both fuel pressure and air pressure are kept constant); positive blowers or fan blowers with proportioning mixers; inspirating burners (Fig. 11), in which the velocity of the gas is used to produce a suction which induces an approximately proportional amount of air, or *vice versa*; interconnected pressure-difference meters in the fuel and air lines, arranged to vary the air supply to maintain a constant fuel-to-air ratio. (Trinks, Industrial Furnaces, Vol. II, Chap. 4.)

**HEAT-SAVING DEVICES.**—In combustion furnaces of the batch type the products of combustion leave at high temperature and carry with them a large amount of heat which cannot be utilized directly in the furnace. A part of this heat may be recovered either by preheating the cold stock or by preheating the combustion air. Fig. 39 shows a furnace for preheating cold stock. The combustion air may be preheated either by a recuperator or a regenerator. In the recuperator the outgoing flue gases transfer a portion of their heat to the incoming air in a steady flow through a wall. In a regenerator the products of combustion impart heat to brickwork in a regenerator chamber, while the incoming air absorbs heat from the brickwork of another regenerator chamber which previously has been heated by the flue gases. The hot gases are caused to flow through first one and then the other regenerator chambers at regular intervals. The calculation of regenerative and recuperative surface is too long and tedious to be given here. For details of these calculations, see Trinks, Industrial Furnaces, Vol. I.

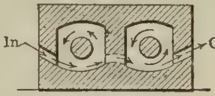

FIG. 39

For small, intermittently operated furnaces, it seldom pays to use heat-saving devices. Their proper field is in large, continuously-operated furnaces, especially in regions where fuel is expensive.

**EXAMPLE OF FURNACE CALCULATION.**—Furnace for heat-treating small steel parts, at the rate of 500 lb. per hour. Furnace is of the under-fired type, Fig. 21. Temperature of furnace = 1450° F. From Fig. 1, increase of heat content from 60° F. to 1450° F. = (222 − 6) = 216 B.t.u. per lb. For steel, the heat content is actually slightly higher than this value, which applies to pure iron. From Table 2, the thermal efficiency of the given type of furnace under average conditions = 19%. Heat required in fuel per pound of steel = 216/0.19 = 1140 B.t.u. If the fuel to be used is oil of 140,000 B.t.u. per gal., the fuel consumption is (500 × 1140)/140,000 = 4.1 gal. per hour.

From Rule 3, p. 18–05, the rate of heating will be about 30 lb. per hour per sq. ft. of hearth. Required area of hearth = 500/30 = 16.7 ft. The shape of the hearth and height of the heating chamber are determined by the size of the pieces and the method of charging, or in some cases by the nearest standard sizes of the furnace builders.

Taking rate of heat liberation by combustion as 12 B.t.u. per cu. ft. per second, the volume of combustion space required is (500 × 1140) ÷ (3600 × 12) = 13.2 cu. ft. The height of chamber required to give this volume of combustion space depends upon the shape and size of hearth supports, but would approximate 15 in. The rate of 12 B.t.u. per cu. ft. per sec. is somewhat high for the combustion of fuel oil with burners of the usual type, but this simply means that part of the combustion takes place in the upper or heating chamber. No harm is caused thereby.

From the table of flue sizes (see p. 18–06), area of flue openings required for furnace temperature of 1400° F. = 2.0 sq. in. per sq. ft. of hearth; for 1450° F. furnace temperature it would be somewhat more, or about 2.5 sq. in. Hence, 2.5 × 16.7 = 42 sq. in., total area of flues. The flue dimensions are made multiples of 2 1/2 in. and 4 1/2 in.

The size of doors is determined by the size and shape of heating stock and the method of charging.

# INDUSTRIAL ELECTRIC HEATING

By R. M. Cherry

## 1. FIELD AND CLASSIFICATION

Electric heat may replace advantageously fuel and steam in many industrial heating processes, because the heat generated by electricity usually can be applied at a higher efficiency than is realized with fuel. Its application for heating often results in increased production, a better quality of work, and in many cases lower overall costs.

Whether or not electric heating is used for a given application depends on the complete cost analysis, including such factors as cost of heat, quality of product, labor of attendants, subsequent operations, working conditions, complaints, rejects, etc.

Industrial electric heating gives perfect heat distribution, and permits great flexibility in temperature control. Automatic control is recommended for practically all applications, as it not only gives better control of temperature and heat distribution, but also eliminates the labor of attendants, danger of spoilage, and insures that parts subjected to heat receive exactly the same treatment, thereby producing uniform results.

In general, industrial electric heating is classified as:

1. Small heating units and devices, for such applications as small ovens, heating watchmen's houses, crane cabs, soft metal melting pots, liquid heating, glue pots, process machinery, etc.
2. Ovens for baking, drying, etc.
3. Resistance furnaces, for heat-treating metals, porcelain enameling, etc.
4. Melting furnaces for melting steel, copper and brass.
5. Inductive heating of metals.

## 2. SMALL HEATING UNITS AND DEVICES

Manufacturers' catalogs contains fairly complete information on small heating units and devices, method of application, and determination of type, size and rating of the units required for a particular service.

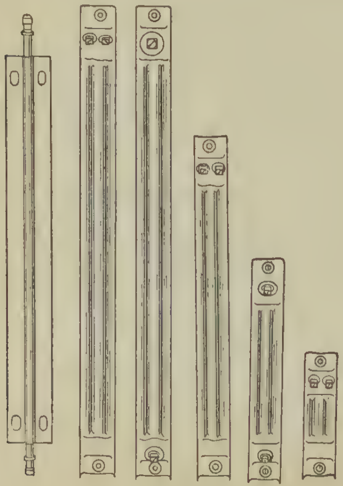

Fig. 1. Strip Heaters

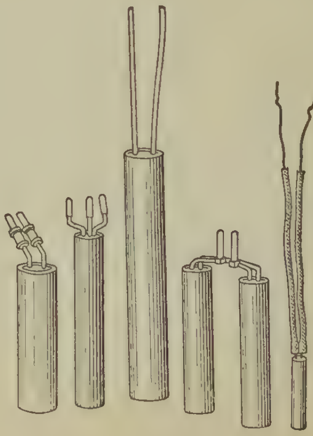

Fig. 2. Cartridge Units

Small heating units and devices are, in general, classified as: Air heating, liquid heating, and contact heating. For these applications, there are various types, sizes and ratings of standard heaters, generally known as strip heaters, tubular type heaters, and cartridge heaters.

**STRIP HEATERS** are used principally for air heating and contact heating. They consist of an outer metal sheath with the coiled resistance wire electrically insulated from the outer sheath and provided with suitable terminals. These heaters are usually approximately 1 1/2 in. wide, 1/2 in. thick or less, and available in standard lengths from approximately 7 in. up to 4 ft. They are used principally for heating watchmen's houses, crane cabs, valve houses, isolated rooms, small ovens, pump houses, ticket booths, process machinery, etc.

**TUBULAR-TYPE HEATERS** are used principally for liquid heating and contact heating. They comprise a tubular outer metal sheath in which the resistance wire, electrically insulated from the outer sheath, is coiled. The outer sheath material may be steel, nickel, copper, stainless steel, Monel metal, etc., depending on the application of the unit. This type of unit can be formed into almost any desired shape. Also, it may be cast into gray iron, alloy iron, aluminum, or other metals. Such cast-in units provide a large area in contact with the liquid to be heated, and form a mechanically strong unit.

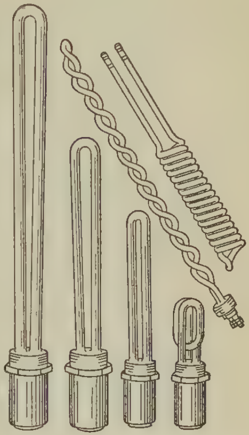
FIG. 3. Immersion Heating Units

Tubular units, either of the plain sheath type or cast-in type, are used principally for heating such liquids as water, oils, paraffin, alkaline baths, nickel-, copper- or chrome-plating bath solutions, mild sulphuric acid solutions up to 10% concentration, if no other acids are present; for melting soft metals, as tin, lead, solder, babbitt, type metal, and similar alloys; and for heating salt baths to temperatures up to approximately 1000° F. Locating the heating units directly in the liquid to be heated, gives a maximum efficiency of heating, in that all of the heat is generated in the liquid itself; also, this method of applying heat provides for the best possible temperature control.

The tubular type heating unit is made in sizes varying from approximately 1/4 to 3/4 in. diameter and from 2 to 20 ft. long. The size and rating depend on the application.

**CARTRIDGE HEATERS.**—Cartridge heating units are used principally for contact heating. The construction is similar to that of the tubular unit, in that it consists of an outer shell of metal, and coiled resistance wire suitably insulated from the outer shell. The two terminals of the heating unit are brought out at one end. Cartridge heaters are used where small quantities of heat are required at a definite location. They are applied usually by drilling a hole in the part to be heated and inserting the heating unit.

Cartridge heating units are available in sizes from 3/8 in. outside diameter by 1 1/2 in. long to approximately 1 1/4 in. outside diameter by 8 1/2 in. long. Ratings range from 50 watts in the smaller size, up to approximately 1200 watts in the larger size. These ratings are based on a sheath material of brass. Cartridge heating units also are available with an alloy steel sheath, suitable for a maximum sheath temperature of 1200° F. The electrical rating for these units is approximately 50% higher than those listed above.

Typical heaters are shown in Figs. 1, 2 and 3. Fig. 1 shows several forms of strip heaters, and Fig. 2 various arrangements of cartridge heaters. Fig. 3 illustrates a number of varieties of liquid heating units, technically known as immersion units. As shown, the units may terminate in a pipe fitting which may be screwed into the wall of the chamber containing the fluid to be heated. Typical methods of application of the strip, tubular and cartridge units are shown in Figs. 4, 5 and 6.

## 3. OVENS FOR BAKING AND DRYING

Electric ovens are widely used for many industrial applications. Examples are core baking, paint drying, japanning, bread and cake baking, and baking of such electrical parts as armatures and coils.

**Batch-type Ovens** are used principally for small production. A typical batch oven is the box type, with a door at one or both ends. The material to be baked is loaded on trays, trucks, or an overhead intermittently-operated conveyor.

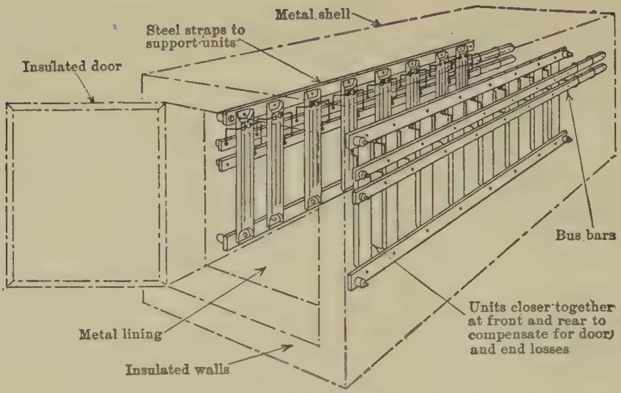
Metal shell
Steel straps to support units
Insulated door
Bus bars
Units closer together at front and rear to compensate for door and end losses
Metal lining
Insulated walls

FIG. 4. Strip Heaters Mounted in a Small Oven

**Continuous Ovens** are used for quantity production. Typical continuous ovens are the straight-through conveyor type, straight-through conveyor oven with ventilating air passing counterflow to the work, or continuous counterflow oven where outgoing hot material preheats incoming cold work.

The electrical equipment for such ovens consists of standard type oven heaters and suitable automatic control equipment. The heaters usually are open-type ribbon resistors, wound on suitable insulators. These, in turn, are supported in a metal frame, suitable for mounting on the walls or floor of the oven. A typical installation of strip heaters is shown in Fig. 4.

**VENTILATION** is an important factor for many oven applications, such as those used for paint drying and japanning, where considerable quantities of volatiles are given off during heating. The amount of ventilation required depends on the size of oven, the size and surface area of parts being baked, and the amount of volatiles to be driven off. Most of the volatiles are driven off during the first part of the heating cycle. Therefore, in a continuous oven they must be removed near the entrance to the oven. All dipped or sprayed parts should be allowed to drip thoroughly before being placed in the oven.

**ENERGY REQUIRED** for the baking of particular parts depends on the production, and size and type of oven. The approximate energy required for various applications is as follows:

| Application.................... | Japanning | | Core Baking | | Bread Baking |
|---|---|---|---|---|---|
| Type of oven................. | Batch | Continuous | Batch | Continuous | Continuous |
| Approx. lb. per Kw.-hr......... | 15–20 | 25–45 | 10–12 | 12–15 | 10–12 |

## 4. RESISTANCE FURNACES

The resistance furnace consists of a heating chamber, containing heating units, which may be mounted on the side walls, roof, floor, doors, depending on the application. Heat is transmitted to the work by radiation and convection. Resistance furnaces are used for such heating applications as annealing, normalizing, hardening, drawing, carburizing, porcelain enameling and brazing.

**RESISTORS** are principally the metallic resistor in rod, ribbon or cast form, or a non-metallic resistor, as silicon carbide, usually in round bar form. Nickel-chromium alloy (approximately 80% Ni, 20% Cr) is universally used for operating temperatures up to a maximum of approximately 2100° F., which covers the majority of industrial heating applications. A metallic resistor material (an Al-Cr-Fe alloy) recently developed (1937) may increase the maximum temperature range of the metallic resistor.

**TYPES OF FURNACE.**—(See also pp. 18-13 to 18-18). Many standard types of furnace are available, the selection of type and size of furnace depending on the application, production, and method of handling desired. The more common types of furnaces are:

**Box-type Furnace,** which is essentially a box, with a door at one or both ends. The parts to be treated are handled by hand, either individually or in containers.

**Pit Furnace,** which may be either circular or rectangular, with the opening at the top, and provided with a suitable cover.

**Bell Furnace,** which may be either circular or rectangular. The material to be treated is loaded on bases, and the furnace is lifted and moved from base to base; or the furnace may be lifted and the bases moved. Bell furnaces are used either with or without an inner retort, depending on the application.

**Elevator Furnace,** which may be either rectangular or circular, usually rectangular. The work is loaded on a car, which is pushed under the furnace and elevated into the furnace chamber.

**Car-bottom Furnace,** which is similar to the box-type furnace, except that the hearth is mounted on wheels to permit it to be removed for loading and unloading.

**Pot Furnace,** which consists of an inner metallic pot, with the heating units located outside of the pot. Pot furnaces are used principally for lead hardening, cyanide or salt hardening, and hot dip galvanizing.

**Pusher Furnace,** in which the work is pushed through the furnace, one piece against the other where the parts can be so handled; or the parts are loaded on stools or trays, which are pushed through the furnace one position at a time.

**Rotary-hearth Furnace,** in which a rotating table, provided with sand seals between the rotating table and stationary part of the furnace, carries the parts to be treated. Loading and removal of work is through the same or adjacent doors, thereby providing for loading and unloading from one central point.

**Roller-hearth Furnaces** have a hearth of driven rolls, on which the material being treated is carried through the furnace. The spacing between rolls depends on the parts or trays to be handled. The rolls extend through the walls of the furnace to bearings mounted on the outside.

**Conveyor Furnaces** are of many types, depending on the type of conveyor used, as a mesh-belt, cast link belt, monorail conveyor, etc. For such applications as hardening of steel, the conveyor should be arranged to remain in the furnace at all times to conserve heat. When the furnace is used for annealing, the parts being cooled in the furnace, the conveyor usually returns underneath it.

**ATMOSPHERE EQUIPMENT FOR RESISTOR FURNACES.**—The trend in resistor furnace applications is the use of protective gas atmosphere in the heating and cooling chambers to prevent oxidation during the heating cycle. Typical applications are for bright annealing copper and steel, scale-free hardening, scale-free annealing of castings, electric furnace brazing, etc. For most of these applications, a suitable protective atmosphere is provided by equipment which processes the available gas, such as city gas, natural gas, butane, propane, etc.

An equipment for re-forming the available gas consists of suitable visual flowmeters to indicate the amount of air and gas used, a motor-driven air-gas mixer, a combustion chamber with burner, and a surface cooler for removing excess moisture. In operation, air and gas are fed into the combustion chamber at a definite ratio, which depends on the gas used and the required analysis of the output gas. In the combustion chamber, the gas is burned with a deficiency of air, in the presence of a catalyst under controlled conditions. The output gas contains principally nitrogen, hydrogen, carbon monoxide and carbon dioxide. The relative proportions of the constituent gases depend upon the air-gas ratio used.

The output gas may be purified, depending on the requirements of the application.

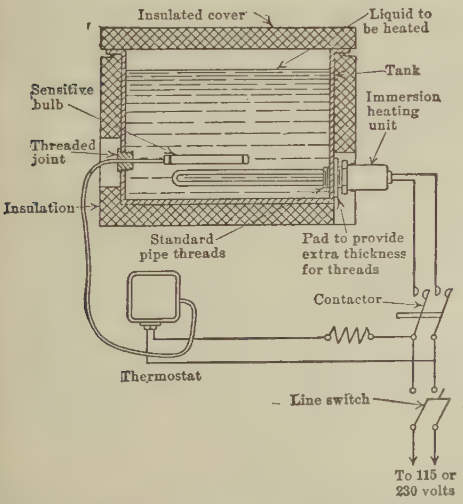

Fig. 5. Immersion Heater Applied to a Tank, Including Automatic Control Equipment

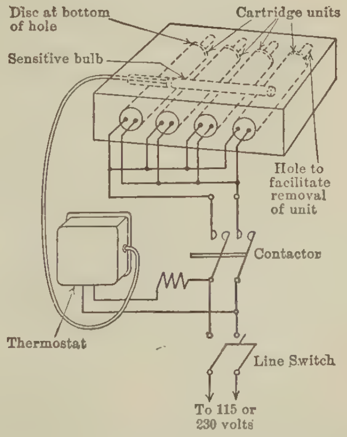

Fig. 6. Application of Cartridge Units with Automatic Temperature Control

For example, most city gases contain a small amount of sulphur, which must be removed if material such as copper is to be bright annealed. In other applications, it is essential to remove more of the moisture from the gas, either by refrigeration or chemical dryers, or both.

The cost of a protective atmosphere is relatively low. For instance, in bright annealing copper, and using available city gas of 530 B.t.u. content per cubic foot, the air-gas ratio used is approximately 4 : 1. Therefore, a gas output of 1000 cu. ft. would require approximately 250 cu. ft. of city gas. For scale-free hardening, the air-gas ratio may be approximately 2 1/2 : 1, and there would be required approximately 400 cu. ft. of city gas of 530 B.t.u., per 1000 cu. ft. of output gas.

When a higher reducing gas than that provided by processing available gases is desired, an ammonia dissociator may be used. The equipment is essentially a tube or coil, containing a catalyst, through which the ammonia gas passes, inserted in an electric furnace. The ammonia gas dissociates when heated to the required temperature, the resulting gas containing 75% hydrogen, 25% nitrogen. Many sizes of standard atmosphere equip-

**Table 1.—Typical Applications of Electric Resistance Furnaces**

| Process | Material | Type of Furnace | Typical Applications | Approx. Energy Required, lb. per kw.-hr. |
|---|---|---|---|---|
| Annealing | Copper and Brass | Box<br>Pusher Tray<br>Conveyor<br>Roller Hearth<br>Bell | Copper wire on reels<br>Copper tubing, straight lengths<br>Copper tubing, in coils<br>Brass wire or strip, in coils | 25–35<br>12–20<br>10–15<br>15–20 |
| | Malleable Iron | Elevator<br>Pusher Tray<br>Roller Hearth | Malleable castings | 5–7 |
| | Steel bar stock and tubes | Car-bottom<br>Pit<br>Elevator<br>Roller Hearth | | |
| | Steel sheets coiled strip | Bell<br>Elevator | Steel strip in coils | 11–14 |
| | Steel strip, (continuous) | Roller Hearth | | |
| | Steel castings and forgings | Car-bottom<br>Elevator<br>Pusher Tray<br>Roller Hearth | | 7–10 |
| | Steel wire in coils | Bell | | 10–13 |
| Copper Brazing | | Box<br>Conveyor<br>Roller Hearth | | |
| Hardening | Steel | Box<br>Pot<br>Pit<br>Conveyor<br>Pusher<br>Rotary Hearth<br>Roller Hearth | Hardening at approximately 1500° F. | 7–11 |
| Drawing | Steel | Pit<br>Box<br>Oil Bath<br>Salt Bath | Drawing<br>at approximately 400° F.<br>at approximately 800° F. | <br>40–60<br>20–25 |
| Galvanizing | Steel | Pot | Hot dip galvanizing | 12–20 |
| Porcelain Enameling | Steel<br>Cast iron | Box<br>Continuous conveyor | Sheet steel, single coat<br>in box furnace<br>in continuous furnace<br>Cast-iron, wet process single coat<br>in box furnace<br>in continuous furnace | <br>5–8<br>10–15<br><br>6–10<br>12–15 |

ments are available for a wide range of outputs and applications. Makers should be consulted in regard to equipment for special conditions.

The quantity of atmosphere required depends on the type and size of furnace, and the application. In general, it is a relatively small item in the total cost. Such protective atmosphere easily can be supplied to practically all types of furnaces, if they are properly designed for use with protective atmosphere.

The protective atmosphere eliminates oxidation of most materials. A considerable saving is effected thereby, by the elimination of the subsequent cleaning operation, and the quality and appearance of the product is improved. The cost of fabrication, such as brazing fabricated metal parts in the electric furnace, is lowered.

**SELECTION OF TYPE. ENERGY REQUIRED.**—Table 1 gives, in general, the types of furnaces used for various applications. The type and size of furnace to be selected depend on the size, shape, weight and material of parts to be treated, as well as the production and the desired heating and cooling cycle. The energy required, usually stated in net pounds per kilowatt-hour, based on continuous operation, depends upon the size of furnace, production, and the heating cycle. The data in Table 1 are approximate and are to be used only as a rough guide.

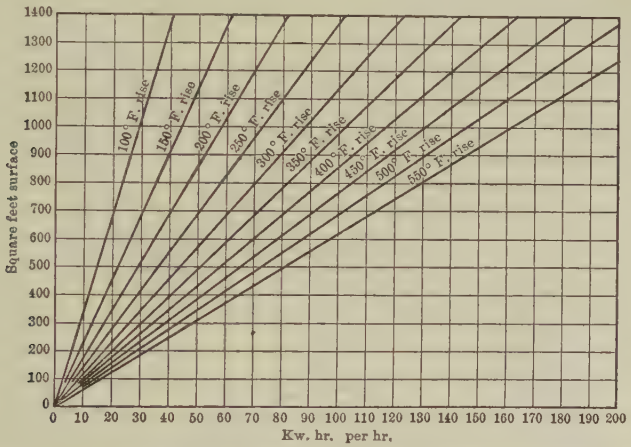

FIG. 7. Heat Loss Through 1-in. Thick Insulation of 85% Magnesia or Similar Insulator. Heat Loss Varies Inversely as Thickness

**ENERGY CONSUMPTION AND ELECTRICAL RATING.**—An estimate of the energy required for a given application should include the heat absorption of the material being heated, heat absorption of auxiliaries used for carrying the work, as trays, conveyor, etc., heat absorption of the ventilating air if used, heat absorption of the protective gas atmosphere if used, heat losses through the walls, doors, etc.

The heat absorption of the material to be heated, including the net work, conveyor, trays, etc., can be calculated by

$$\text{Kw.-hr.} = \{w \times h_s \times (t_2 - t_1)\}/3412,$$

where $w$ = weight of material being heated, $h_s$ = average specific heat, $t_1$ and $t_2$ = respectively, initial and final temperatures of the material, deg. F. Heat losses for many applications may be estimated by means of Figs. 7–11. The heat loss data in the above curves are principally for low temperature applications. For the higher temperature resistance-type furnaces, operating up to approximately 1850° F., a good rule-of-thumb method for estimating the heat loss in Kw.-hr. is

$$\text{Kw.-hr.} = (0.8\ A \times T)/10{,}000,$$

where $A$ = inside wall area of heating chamber, sq. ft. and $T$ = operating temperature, deg. F. The application of this rule will give the approximate heat losses through the insulated walls and closed doors, based on a well-insulated furnace.

**REFRACTORIES AND INSULATION FOR OVENS AND RESISTOR FURNACES.**—The usual construction of ovens, for temperatures up to approximately 600° F., comprises a metal inner liner and an outer shell, the space between being filled with insulating material, as 85% magnesia, rock wool, mineral wool, etc., care being used to have little or no through metal.

The thickness of insulation used depends on the operating temperature and, in general, should be at least as follows:

| | | | | |
|---|---|---|---|---|
| Oven temperature, deg. F. | 200 | 400 | 600 | 800 |
| Approx. thickness of insulation, in. | 2 | 4 | 5 | 6 |

For electric resistance furnaces, the wall construction consists of an inner refractory lining of brick, backed by a suitable thickness of insulating material. Several types of light-weight refractory are available in standard brick sizes. These bricks are of less than half the weight of regular firebrick, thereby providing for additional insulation, and absorbing less heat while the furnace is being heated to operating temperature. For the average size heating chamber, the inner refractory lining of 4 1/2-in. standard light-weight refractory brick is sufficient. For high walls and long spans for arches, this thickness should be increased. For data on refractory brick see pp. 5–03 to 5–20 of this book. For additional data on heat insulation see Kent's Mechanical Engineers' Handbook—Power, pp. 3–54 to 3–71, forming Vol. 2 of this series.

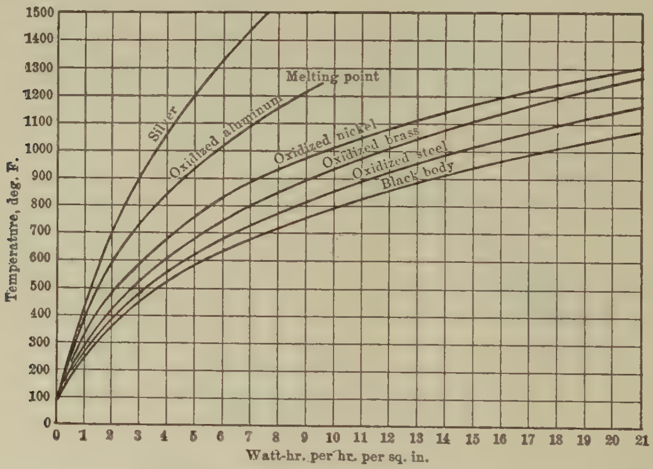

FIG. 8. Heat Losses from Various Metals

The thickness of insulation depends on the operating temperature and the complete time-temperature cycle. A furnace to be held at operating temperature for long periods should be well insulated. For a cycle requiring cooling of the charge in the furnace, the thickness of the insulation must be in accordance with required heating and cooling cycle.

In furnaces held at the operating temperature, the thickness of the insulation, in addition to the refractory lining, should be approximately as follows:

| | | | | |
|---|---|---|---|---|
| Operating temperature, deg. F. | 800–1100 | 1100–1400 | 1400–1800 | 1800–2100 |
| Approx. thickness of insulation, in. | 5 | 7 1/2 | 10 | 12 1/2 |

## 5. MELTING FURNACES

The three types of electric melting furnaces are, in general, the resistance furnace, arc furnace, and induction furnace.

**RESISTANCE FURNACES**, usually of the pot type, are used for melting soft metals, as tin, babbitt, lead, solder, type metal, etc. The heating units may be immersed directly in the metal to be melted, or placed outside of the melting crucible, depending on the application.

**ARC FURNACES**, in general, are of two types, the direct arc and the indirect arc.

**The Direct Arc Furnace** is used principally for melting and refining steel. The 3-phase arc furnace, generally used, has three electrodes. The electric current passes between the electrodes and the bath of metal. Each electrode is provided with a motor-driven mechanism which, by suitable automatic control, raises and lowers the electrodes to maintain a constant power input.

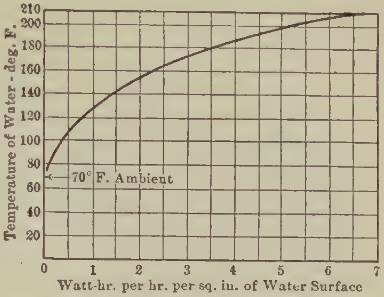

Fig. 9. Heat Losses from Open Water Surfaces

**The Indirect Arc Furnace** usually has two electrodes. The current passes between the electrodes and above the bath, which is heated by radiant heat from the electric arc, as well as by conduction and convection. This type of furnace is used frequently for melting of copper and brass.

**Electrical Equipment** of an electric arc furnace consists of a step-down transformer, primary and secondary control, motors for raising and lowering the electrodes, and a tilting motor if the furnace is to be of the tilting type. The primary control consists of an oil-circuit breaker on the high-voltage side of the step-down transformer, suitable means for changing taps on the transformer, and metering equipment. The secondary control consists of control equipment for controlling the movement of the electrodes to maintain constant power input. Voltages used on the arc vary, depending on the character of the application. The higher voltages are used normally for melting down, and the lower voltages for refining.

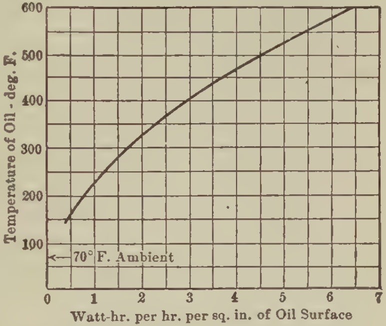

Fig. 10. Heat Losses from Surfaces of Oil Baths

**INDUCTION FURNACES** are of two types, the core type, used on standard frequencies, and the coreless or high-frequency furnace.

**Core-type Furnaces** consist essentially of an electrical transformer circuit with a primary coil, a laminated iron core, and the bath arranged for completion of the secondary circuit. Heating is accomplished by the induced current in the secondary. This type of induction furnace is used principally for melting brass and copper.

**Coreless or High-frequency** furnaces consist of a crucible to hold the material to be melted, around which is placed a water-cooled copper coil to which the high-frequency source is connected. Heat is generated by the electric current induced in the charge. The rate of heating is limited practically only by the capacity of the high-frequency source. This type of furnace provides for rapid heating, accurate control of heating, agitation of the bath, and is used principally for the melting of alloy steel.

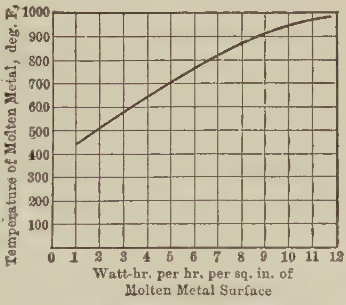

Fig. 11. Heat Losses from Surface of Molten Lead, Tin, Type Metal, Babbitt, etc.

**ENERGY REQUIRED FOR MELTING** in induction furnaces depends on the size of furnace, and the material to be melted. In general, the approximate electric energy consumption for melting for various applications is as follows:

| | |
|---|---|
| Brass, yellow (core induction furnace) | 220 to 350 Kw.-hr. per ton (2000 lb.) |
| Brass, red ( " " " ) | 250 to 400 " " " " |
| Copper (indirect arc furnace) | 250 to 400 " " " " |
| Steel (3-phase arc furnace melting only) | 450 to 600 " " " " |
| Steel (coreless induction furnace) | 500 to 800 " " " " |

The selection of the right type and size of furnace for a given application requires experience and, in some cases, tests. When there is any question as to size and type of furnace, manufacturers of such equipment should be consulted. Complete data should be given, *i.e.*, voltage, phase and frequency of the power circuit; floor space and head-room available; the size, shape and material of parts to be treated; the complete heating and cooling cycle desired; the efficiency of loading, *i.e.*, pounds of material to be loaded per square foot or cubic foot; production per given time, hours to be operated per day or week.

## 6. INDUCTIVE HEATING OF METALS

The use of high frequency (1000 to 3000 cycles) for heating (not melting) metals, for such application as skin hardening is of recent (1937) development. This method of heating permits heating only that portion of the metal that requires hardening. In addition it permits of heating only the surface of that portion. A typical example is the heating of the bearing surfaces of crank-shafts for hardening. A high-frequency current of low voltage and high amperage is passed through inductor blocks which surround but do not actually touch the area to be heated. The inductor block current induces a current in the surface of the metal, thereby providing quick heating of the outer surface.

This method of heating reduces the time for hardening of heavy parts, such as crank-shafts, from hours to seconds and gives a superior product at lower overall cost. High-frequency heating for surface hardening is particularly applicable to the heating of steel parts of cylindrical section.

## Section 19

# FORGING PRACTICE

By Eugene C. Clarke

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# FORGING PRACTICE *

By Eugene C. Clarke

Forging, the plastic forming of hot metal, is one of the older arts of civilization. The early smiths provided the energy necessary to form the metal by striking with hammers the subject held in place on anvils. Also, they were the early heat treaters; they tempered their products.

The operations and mechanisms cited below account for the production of practically all the forged products in current use. There are, of course, special operations employing modifications or different applications of the principles described, but the tonnage so produced is relatively insignificant.

Progress in the reduction of the art of forging to a scientific basis has brought about specialized development of the various elements involved in shearing, heating, plastic formation, trimming, heat treating, sizing, inspecting, and the many other operations incident to modern practice. The plastic forming of the metal, being accomplished by working the metal beyond its elastic limit, has not enjoyed the same theoretical study that is common to processes within elastic limits, and practically all formulas employed are empirical.

**ENERGY REQUIRED.**—If in mechanical forging $E$ represents the forging energy, $V$ the velocity of impact of the forging element of a mechanism, and $W$ the weight or force applied, $E = WV^2/2g$. From this it is readily appreciated that the weight or pressure applied must be greater than the affected mass of the forging and that, with the size of the forging placing a practical limitation on mechanical equipment employed, the classes of forging equipment are economically complementary rather than competitive.

The rate of plastic flow of the subject metal at forging temperature limits the amount of deformation per operation and governs the application of forging power. If deformation is accomplished beyond the rate of plastic flow, rupture of the metal occurs. The rate of plastic flow varies with the subject material, its form and density, its temperature during the forging operation, and the velocity and continuity of power application.

**LIMITATIONS.**—This discussion is confined to consideration of actual forging applications. It, therefore, cannot include details of heating, shearing, heat treatment, etc., but a few characteristics of the metal being forged exert influences common to all procedures and are briefly mentioned.

During all forging operations the metal is cooling. Hence its fluidity for plastic flow is being reduced. The duration of any forging operation thus is limited, the limitation varying further with the kinds of metal being forged. The forging dies are extracting heat from the forging in relation to the time in contact with it. Hammer dies have relatively little thermal effect, but in press dies this extraction of heat is a major consideration.

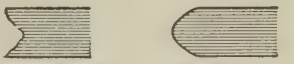
Hammered Section (Exaggerated)
Pressed Section (Exaggerated)

Fig. 1. Metal Flow—Presses and Hammers

When the forging is larger in the area to be deformed than the impinging or contacting dies, as in cogging down an ingot, the forging effect is reduced by the resistance to distortion of that part of the metal not in contact with the dies of the forging mechanism. The deformation necessarily is limited in scope to that which will not cause laps at adjoining forged areas and which does not shear the metal at those sections under the edges of the dies.

When the forging is smaller in area than the die face, a limitation on height is introduced. Generally, the greater the area upon which the dies impinge, as related to the height between impinging dies, the less is the control of plastic deformation, with the limitation that the distance between the dies at the point of impact cannot safely be greater than three times the smaller sectional dimension. When the height limitation is exceeded the forging frequently will bend, forming a lap, rather than follow the desired plastic deformation. The hammer with high impact velocity is effective on thin stock which is unworkable in a press, and the press is more effective on relatively thick sections. The difference in metal flow under the two tools can be observed at the ends of sections reduced by forging. That reduced under a hammer will have the edges extending beyond the center; that reduced under a press will have the center extending beyond the edges.

---

* *Forging* as discussed herein is defined as the craft of the smith and as such does not consider plastic deformation as accomplished in rolling mills.

**IMPACT AND PRESSURE.**—The velocity, constancy and duration of the applied forging force are best considered with relation to the mechanism employed. The first application of force must overcome the initial resistance to deformation of the metal, after which plastic flow will occur only so long as the force is continued in motion and in magnitude. Coincident with the application of the force, the density of the metal is increased, increasing its resistance to flow. That immediately worked-on area of the metal in contact with the forging dies is further restricted by friction between die and work, and this increases with the magnitude of the force.

The mechanical application of power to accomplish plastic deformation or forging has been developed along two distinctive lines: 1. Impact forging, *i.e.*, the utilization of the energy due to velocity at impact as in hammers; 2. Pressure forging, *i.e.*, the application of pressure at relatively low velocity.

Tremendous strides have been made in mechanization of forging processes. Forging equipment generally divides into three groups, namely hammers, hydraulic presses, and crank-driven machines. Likewise forged products have developed along clearly defined lines, *i.e.*, drop forgings or die forgings, made within enclosing dies controlling form and size; drawn forgings combining piercing and extruding, as shell blanks; and, that group of forgings manipulated, between open dies, to form and size, and controlled by observation or gage.

**Forging by Impact,** as by hammers, is an application of maximum force during contact of dies and metal; the forging effect continues until the resistance to flow brings the impacting die to rest, thus absorbing all the useful energy of the applied force. If the hammer blows are of sufficient rapidity, advantage is gained of the momentum of flow, and greater rate of flow per blow is attained than will be the case with blows at greater intervals.

**Forging by Pressure,** applied and maintained in motion hydraulically, is limited by the resistance to flow of the metal, the increased density of the metal between the dies, the friction between dies and work and the increasing resistance to flow caused by reduction in temperature during the operation. Forging by pressure applied through a crank mechanism is the application of a force of increasing magnitude at decreasing velocity. Such applications are almost universally applied to forgings completely enclosed in dies; the limitations on useful deformation then are the amount of metal under compression in the dies, and its density as being workable within the useful force available.

**GRAIN FLOW.**—The strength and durability of steel forgings are at their maximum values only when the grain flow lines are in the correct position. The selection of the forging mechanism, the determination of the successive forging operations, and the design of dies are governed by the required grain flow. Grain flow in steel is the fibrous structure imparted to the constituents by mechanical working as in rolling or cogging. The alignment of such fibers is parallel, and in the longitudinal direction of the working.

Forging operations should dispose the grain flow to the best advantage of the duty of the finished forging. Thus gear blanks are upset to secure radial grain flow and uniform strength in all teeth. If the blank were forged with longitudinal grain flow, half the teeth would have cross grain flow, and thus would be weaker than the remainder.

In a crank-shaft, the grain is required to flow without rupture through the cheeks, pins and bearings, and to be radial at the end when a flange or disc is employed to transmit the power of the engine or motor of which the crank-shaft is a part. The size of the crank-shaft determines the equipment employed. An automotive crank is forged from rolled stock in a drop hammer, and the end is upset in a forging machine.

Table 1 shows the characteristics of impact forging and pressure forging, and illustrates the current uses and limitations of equipment available in 1937 markets.

## 1. IMPACT OR HAMMER FORGING

Impact forging is commercially developed along two lines: 1. *Drop-forging*, or *die-forging*, *i.e.*, the accurate reproduction of metallic parts shaped by forms or cavities in enclosing dies, which dies are actuated by power hammers; 2. *Hammer-forging*, *i.e.*, the refining of billets, and the forming of billets and bars into shapes between flat dies and swages in power-driven hammers. The principal difference in the hammers employed is in construction. Die-forming or drop-hammers are rigidly constructed with respect to the alingment of the reciprocating hammer ram and the anvil, so that accurate die alignment is maintained. Forging hammers are constructed with the anvil separate from the balance of the hammer and mounted on an independent foundation.

**DROP-FORGINGS** or die-formed forgings made under drop hammers must embody in their forms, as outlined by R. E. W. Harrison, Developments in Modern Drop-forging and Their Relation to Current Machine Shop Practice, *Trans.*, A.S.M.E., 1936: 1. Minimum machining allowance consistent with proper cleaning up of finished surfaces; forgings true to die and perfectly matched; 2. Consistent and easy machinability; 3. Controlled grain flow of metal and avoidance of defects, as cold shuts, laps or seams; 4. Accurate sizing and locating of all surfaces which otherwise would require machining; 5. Minimum wastage of material, *i.e.*, small stub and flash; 6. Minimum number of actual forging and straightening operations; 7. Speed, precision, reliability and safety for the operator, in so far as the forging mechanism is concerned.

As these requirements are reduced to practice for a given material the major considerations become: 1. Die material, die design, and die setting; 2. Type of hammer; *a*, gravity drop with fixed fall, and hence the same energy available to accomplish each stage of the operation; *b*, power accelerated hammer, controlled as to power of each blow by the hammer operator. It is obvious that dies designed to operate in the one type of hammer will not be of the best form for use in the second type.

**DIE BLOCKS.**—Die blocks are made from nickel-chrome-molybdenum grain-controlled alloy steels, of which the following are typical:

| Carbon | Manganese | Chromium | Nickel | Molybdenum |
|---|---|---|---|---|
| 0.50–0.70 | 0.50–0.90 | 0.60–1.30 | 1.00–2.00 | ......... |
| 0.50–0.70 | 0.60–0.90 | 0.75–1.00 | ......... | 0.15–0.25 |
| 0.50–0.70 | 0.50–0.90 | 0.50–1.00 | 1.00–2.00 | 0.15–0.35 |
| 0.50–0.70 | 0.50–0.80 | 0.75–1.00 | 1.50–2.50 | 0.60–0.90 |

Blocks are forged on six faces—upset to destroy all original directional fibers, an essential factor, since the forging forces are exerted in all directions in a die cavity. Die block steel is never permitted to become cold from the time the ingot is cast until final tempering treatment. Blocks are normalized, quenched and tempered.

The block is machined and ground on the working face in order to secure a well-prepared surface for Brinell hardness testing. Blocks are hardened to a uniformity of 0.10 ball diameter. Converting this into scleroscope, it means approximately 4 to 5 points of hardness.

**Table 1.—Characteristics of Impact and Pressure Forging**

| Mechanism | Impact Forging | | Pressure Forging | | |
|---|---|---|---|---|---|
| | Hammers | Hammers | Hydraulic Presses | Steam Hydraulic Presses | Mechanical Forging Machines |
| Power application | Mass falling by gravity | Mass accelerated in fall by power | Continuous velocity and power through relatively long stroke | Continuous velocity and high power through relatively short stroke | Crank |
| Nature of deformation | Surface of forging worked more rapidly than interior | | Upsetting, piercing and extrusion | Interior worked more than surface which is restrained by die friction | Upsetting, piercing, extrusion |
| Practical limitations | Height of fall | Weight of anvil | Strength and flow of subject metal | Size of ingot to be forged | Size of machine elements |
| Products | Die forgings *<br>Surgical instruments<br>Shears<br>Pliers<br>Tableware<br>Automobile parts<br>Machine parts, etc. | Die forgings *<br>Billets<br>Bars<br>Shapes<br>Car axles<br>Crank-shafts<br>Airplane propellers<br>Crane hooks<br>Anchor chains<br>Automobile parts, machine parts, railroad parts, marine parts, etc. | Shell carcasses<br>Pressure containers<br>Tubes<br>Bends<br>Oxygen bottles | Reduction of<br>Ingots<br>Billets<br>Bars<br>Shapes<br>Die blocks<br>Propeller shafts<br>Massive machine parts<br>Ordnance | Die forgings *<br>Cluster gears<br>Automotive gears<br>Bolts, nuts, rivets<br>Valve tappets<br>Automotive parts<br>Machine parts |

* *Die-forgings* has been used as a more descriptive name for the products commercially known as Drop-forgings. The latter designation comes from the initial die forgings made on drop- or gravity-hammers.

The modern die block is furnished in four hardness ranges as follows:

| Hardness letter | A | B | C | D |
|---|---|---|---|---|
| Brinell ball diameter | 2.95–3.05 | 3.10–3.20 | 3.25–3.35 | 3.40–3.55 |
| Scleroscope | 60–56 | 55–52 | 51–48 | 47–44 |

The hardness is decided according to the type of impression in the block. The modern machine shop of 1937 can machine a die block as hard as 2.90 ball diameter, or approximately 62–63 scleroscope.

**DIE DESIGN.**—Drop-forgings are formed, usually of hot bar steel, between two dies, one reciprocating, in the faces of which are machined "half" impressions of the parts. Thus a parting line provides for insertion of the stock and withdrawal of the forged shape. To prevent sticking of the shape in the impression, draft is used on vertical surfaces. Hammers with slow return stroke pick-up require larger draft than those with properly accelerated return stroke, the range running from 10 deg. draft to 1 1/2 deg. draft, with 7 deg. generally employed in the slower hammers, and 3 deg. common to the more rapid double-acting hammers. The amount of deformation of the stock and its nature dictates the number and form of die impressions for drawing, fullering, roughing and finishing. The finishing impression, and sometimes the roughing impression, are surrounded by a gutter into which the flash, or excess metal, flows. Great care is essential in the design of the die for all functions to secure proper grain flow and minimum flash, since the flash, which is the thinnest section, governs the rate of plastic flow within the metal. One of the important characteristics of hammered forgings is the ability of the impact energy to sharply fill raised surfaces such as bosses and flanges. Only in rare cases is venting necessary to accomplish this purpose. A typical lower die for a drop-hammer is shown in Fig. 2. The upper die matches and is similarly machined.

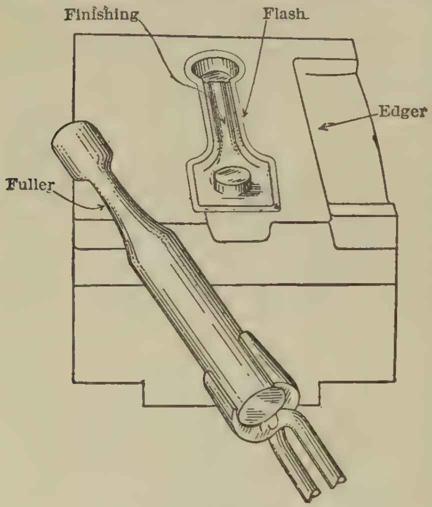

FIG. 2. Hammer Die

**DIE SETTING.**—The final blow struck on a die-forging, and producing the exact reproduction from the dies, also is that blow which has the least vertical displacement and hence absorbs the greatest amount of available energy in the least time. It, therefore, is essential in the reproduction of exact forgings that the dies be set so that the displacement load of the final impression is centered under the falling mass. Since die designing can only approximate this load distribution, it is necessary that adjustable yet rigid means of fastening the dies in the hammer be employed. Fig. 3 illustrates a typical dove-tail mounting for dies. Shims are used on the sides of the slot for left to right adjustment, and a series of dowels with graduated offsets provides adjustment from front to back.

**DROP FORGING PROCESSES.**—In the actual forging operation the hammer man manipulates the forging stock in the various impressions of the dies to secure the desired plastic flow in the finished forging. The heating of the metal is of prime importance to this manipulation, and requires: *a*, control of the rate of heating and avoidance of over-heating or of burning; *b*, forging with the stock at the proper temperature throughout; *c*, elimination of surface scale, usually blown off by blasts of air or steam between blows.

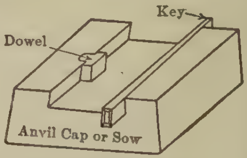

FIG. 3. Die Notch

The Subcommittee on Forging of the American Society for Metals in its Recommended Practice for Heating Steel Bars for Forging, Metals Handbook, 1936 edition, suggests practice for bars and billets up to and including 4 in. square or diameter The rate of heating suggested to bring the center of the stock to forging temperature for plain carbon steel of 0.50 carbon or less is five minutes per inch of diameter or thickness up to 3 in., with a slower rate for larger sizes. Slower rates are recommended for steels with higher carbon content. For alloy steels the rate recommended is six minutes per inch of diameter and thickness. In the "Metals Handbook" data are given on hot working stainless steels, "Fabricating and Finishing Stainless Steels," V. W. Whitmer.

Maximum forging temperatures, cited by Adam M. Steever, Factors Relating to the Production of Drop and Hammer Forgings, *Trans.* A.S.M., vol. xxv, 1937, are:

| | | | |
|---|---|---|---|
| 0.10 carbon steel | 2400° F. | 0.90 carbon steel | 2150° F. |
| 0.20 " " | 2375° F. | 1.00 " " | 2000° F. |
| 0.30 " " | 2350° F. | 1.50 " " | 1900° F. |
| 0.40 " " | 2325° F. | Monel Metal | 2100° F. |
| 0.50 " " | 2300° F. | Copper | 1800° F. |
| 0.60 " " | 2250° F. | Brass | 1400° F. |
| 0.70 " " | 2225° F. | | |

The cooling of the finished forgings requires attention in the higher carbon content steels, usually retardation of the cooling rate.

The larger drop-forgings are trimmed hot, the flash being removed by blanking in a trimming press after they are finished under the hammer. The forgings frequently are re-struck in the hammer for straightening and sizing. Hot trimming is essential when there is any hazard of introducing defects due to the tearing that occurs in cold trimming. Flash from aluminum forgings sometimes is removed by sawing.

The desirable features in a drop forge shop production line are specified in Fig. 4, which diagrams the principal processes involved. The chart is abstracted from a diagram by R. E. W. Harrison (Developments in Modern Drop Forging, and Their Relation to Current Machine Shop Practice, *Trans.* A.S.M.E., 1936). Greater detail is obtainable from Mr. Harrison's paper.

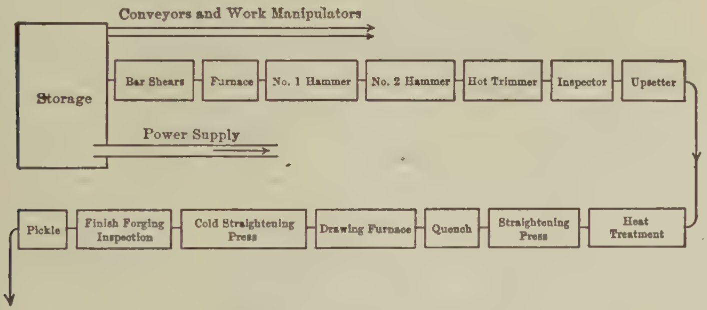

FIG. 4. Drop Forge Processes

**DROP-HAMMERS.**—Two types of hammers are available for drop forging. In the U. S., these are principally, *a*, board drop-hammers, and, *b*, steam drop-hammers. The hammers are rated by the weight of the reciprocating parts without dies. Principal dimensions are indicated in Tables 2 and 3, and Figs. 5 and 6, respectively, for the two types.

Since the hammer is the pacemaker for production, controls the accuracy of form, and represents the major equipment investment, the selection of the proper tool requires expert study. In general, shapes involving considerable drawing are not adapted for production on board drop-hammers. Relatively flat thin shapes best are produced in gravity drop-hammers and the smaller forgings usually are produced on this type. For specific recommendations the manufacturers of hammers should be consulted.

**HAMMER FOUNDATIONS.**—Typical drop-hammer foundations are shown in Figs. 5 and 6. It is essential that foundations be properly installed, that hardpan be the base of the foundation, or that piles be driven. Proper foundations insure the highest production, lowest maintenance costs, and freedom from shock transmission to surrounding structures.

**POWER REQUIRED FOR DEFORMATION.**—No generally accepted method of predetermining the power required to deform metals in forging hammers exists. Many tests on sample shapes under varying conditions have been made (R. H. Thurston, *Trans.* A.S.M.E., vol. v, p. 53, 1884, with copper cylinders; A. W. Moseley and J. L. Bacon, *Trans.* A.S.M.E., vol. xxvii, p. 605, 1906, with Bessemer steel plugs; W. T. Sears, *Am. Mach.*, March 10, 1910, with lead plugs), but the variables are so great and so little is

## Table 2.—Dimensions of Board Drop Hammers and Foundations (See Fig. 5)

Chambersburg Engineering Company, Chambersburg, Pa.

| Rated Falling Weight, lb. | Hammer Dimensions | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A*, in. | *B*, in. | *C*, in. | D, in. | *E*, in. | *H* | *I*, in. | Front to Back, in. | | Boards | |
| | | | | | | | | Ram | Cap | No. | Size, in. |
| 1000 | 17 | 48 | 27 | 21 1/4 | 4 1/2 | 72 | 35 | 14 | 21 | 3 | 1 1/2 × 3 × 81 |
| 1500 | 18 | 51 | 27 | 22 3/4 | 14 | 80 1/2 | 39 | 16 | 23 | 3 | 1 1/2 × 3 × 84 |
| 2000 | 19 | 55 | 26 | 24 | 18 1/2 | 88 | 42 | 20 | 26 | 4 | 1 1/2 × 3 × 91 |
| 2500 | 20 1/2 | 57 | 26 | 25 1/4 | 23 | 92 1/2 | 45 | 21 | 27 | 4 | 1 1/2 × 3 × 96 |
| 3000 | 23 | 59 | 26 | 28 1/2 | 26 3/4 | 99 1/2 | 46 | 23 | 30 | 5 | 1 5/8 × 3 × 102 |
| 4000 | 25 | 61 | 26 | 30 3/4 | 29 3/4 | 112 | 54 | 25 | 32 | 6 | 1 7/8 × 3 × 105 |
| 5000 | 27 1/4 | 67 | 25 | 32 1/4 | 32 1/4 | 124 | 63 | 27 | 38 | 7 | 1 7/8 × 3 × 114 |

| Rated Falling Weight, lb. | Foundation Dimensions | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | *F*, in. | *J*, in. | *K*, in. | *L*, in. | *M*, in. | *N*, in. | Timbers | |
| | | | | | | | No. | Size, in. |
| 1000 | 65 1/2 | 84 | 132 | 43 | 91 | 20 1/2 | 7 | 12 × 12 × 43 |
| 1500 | 83 3/4 | 88 1/2 | 136 1/2 | 47 | 95 | 29 3/4 | 7<br>1 | 12 × 12 × 47<br>4 1/2 × 12 × 47 |
| 2000 | 95 1/2 | 98 1/2 | 146 1/2 | 50 | 98 | 35 1/2 | 8<br>1 | 12 × 12 × 50<br>2 1/2 × 12 × 50 |
| 2500 | 108 1/4 | 104 1/2 | 152 1/2 | 53 | 101 | 42 1/4 | 8<br>1 | 12 × 12 × 53<br>8 1/2 × 12 × 53 |
| 3000 | 116 3/4 | 108 | 168 | 54 | 112 | 44 3/4 | 9 | 12 × 12 × 54 |
| 4000 | 123 3/4 | 120 | 180 | 62 | 122 | 45 3/4 | 10 | 12 × 12 × 62 |
| 5000 | 126 | 132 | 192 | 71 | 131 | 48 | 11 | 12 × 12 × 71 |

## Table 3.—Dimensions of Steam Drop Hammers and Foundations (See Fig. 6)

Chambersburg Engineering Co., Chambersburg, Pa.

| Rated Falling Weight, lb. | Hammer Dimensions | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Bore, in. | *A*, in. | *B*, in. | *C*, in. | *D*, in. | *E*, in. | *F*, in. | *G*, in. | *H*, in. | *J*, in. | *K*, in. | Front to Back, in. | |
| | | | | | | | | | | | | Ram | Cap |
| 1,000 | 7 1/2 | 16 | 17 3/4 | 22 | 4 1/2 | 5 1/2 | 36 | 75 1/2 | 15′ 3 1/4″ | 0 | 35 | 16 | 20 |
| 1,500 | 9 | 18 | 19 3/4 | 23 1/4 | 4 1/2 | 5 3/4 | 36 | 82 | 15′ 10 3/4″ | 7 1/4 | 39 | 19 | 23 |
| 2,000 | 10 1/4 | 19 | 20 3/4 | 26 | 4 1/2 | 6 1/4 | 36 | 88 1/2 | 16′ 7 1/4″ | 11 1/4 | 43 | 22 | 26 |
| 2,500 | 11 | 21 | 23 3/4 | 28 | 5 1/2 | 6 1/2 | 36 | 94 | 17′ 0 1/2″ | 15 | 47 1/2 | 23 | 27 |
| 3,000 | 11 3/4 | 23 | 25 3/4 | 30 | 5 1/2 | 7 | 36 | 101 1/2 | 17′ 7″ | 15 | 52 | 25 1/2 | 30 |
| 4,000 | 13 1/2 | 25 | 27 3/4 | 31 3/4 | 6 | 7 1/2 | 36 | 112 | 17′ 11″ | 20 | 59 | 27 1/2 | 32 |
| 5,000 | 15 1/2 | 27 1/4 | 30 3/4 | 33 3/4 | 7 | 8 1/4 | 36 | 125 1/2 | 18′ 8 3/4″ | 23 1/4 | 64 | 29 | 34 |
| 6,000 | 16 | 27 1/4 | 30 3/4 | 35 1/2 | 7 | 9 | 36 | 129 1/2 | 19′ 2″ | 29 | 68 | 31 | 36 |
| 8,000 | 18 1/4 | 28 | 31 1/2 | 37 | 8 | 10 | 36 | 137 | 19′ 9 1/2″ | 34 | 77 | 35 | 40 |
| 10,000 | 20 | 28 | 31 1/2 | 38 3/4 | 8 | 10 | 36 | 146 | 20′ 4 1/2″ | 63 | 74 | 38 | 44 |
| 12,000 | 21 | 32 | 34 | 40 1/4 | 8 | 10 1/2 | 36 | 153 | 20′ 10″ | 74 1/2 | 78 | 42 | 48 |
| 14,000 | 23 | 32 | 35 1/2 | 40 1/4 | 8 | 10 1/2 | 36 | 154 | 20′ 10″ | 90 | 78 | 42 | 48 |
| 16,000 | 25 | 34 | 37 1/2 | 42 1/4 | 7 | 11 | 36 | 163 | 22′ 4 1/2″ | 91 | 84 | 44 | 51 |

| Rated Falling Weight, lb. | Foundation Dimensions | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | *L* | *M* | *N* | *R* | *S*, in. | *T* | *O*-Timbers | | *P*-Timbers | |
| | | | | | | | No. | Size | No. | Size |
| 1,000 | 4′ 1″ | 8′ 1″ | 7′ 7″ | 11′ 7″ | 26 1/2 | 8′ 2 1/2″ | 4 | 12×12× 7′ 6″ | 7<br>1 | 12×12×4′ 0″<br>6×12×4′ 0″ |
| 1,500 | 4′ 1″ | 8′ 1″ | 8′ 1″ | 12′ 1″ | 35 1/4 | 8′ 11 1/4″ | 4 | 12×12× 8′ 0″ | 8 | 12×12×4 0″ |
| 2,000 | 4′ 7″ | 8′ 7″ | 8′ 7″ | 12′ 7″ | 41 1/4 | 9′ 5 1/4″ | 4<br>1 | 12×12× 8′ 6″<br>6×12× 8′ 6″ | 8<br>1 | 12×12×4′ 6″<br>6×12×4 6″ |
| 2,500 | 5′ 1″ | 9′ 1″ | 9′ 1″ | 13′ 1″ | 45 | 9′ 9″ | 5 | 12×12× 9′ 0″ | 9 | 12×12×5′ 0″ |
| 3,000 | 5′ 1″ | 9′ 1″ | 9′ 7″ | 13′ 7″ | 45 | 9′ 9″ | 5 | 12×12× 9′ 6″ | 9<br>1 | 12×12×5′ 0″<br>6×12×5′ 0″ |
| 4,000 | 6′ 1″ | 10′ 1″ | 10′ 7″ | 14′ 7″ | 50 | 10′ 2″ | 6 | 12×12×10′ 6″ | 1<br>10 | 6×12×6′ 0″<br>12×12×6′ 0″ |
| 5,000 | 6′ 7″ | 10′ 7″ | 11′ 7″ | 15′ 7″ | 53 | 10′ 5″ | 6<br>1 | 12×12×11′ 6″<br>6×12×11′ 6″ | 11<br>1 | 12×12×6′ 6″<br>6×12×6′ 6″ |
| 6,000 | 6′ 9″ | 10′ 9″ | 12′ 1″ | 16′ 1″ | 59 | 10′ 11″ | 6<br>1 | 12×12×12′ 0″<br>8×12×12′ 0″ | 12 | 12×12×6′ 8″ |
| 8,000 | 7′ 7″ | 11′ 7″ | 12′ 7″ | 16′ 7″ | 64 | 11′ 4″ | 7<br>1 | 12×12×12′ 6″<br>6×12×12′ 6″ | 12<br>1 | 12×12×7′ 6″<br>6×12×7′ 6″ |
| 10,000 | 7′ 1″ | 14′ 1″ | 13′ 1″ | 20′ 1″ | 117 | 19′ 9″ | 7 | 12×12×13′ 0″ | 13 | 12×12×7′ 0″ |
| 12,000 | 8′ 1″ | 15′ 1″ | 14′ 1″ | 21′ 1″ | 128 1/2 | 20′ 8 1/2″ | 8 | 12×12×14′ 0″ | 14 | 12×12×8′ 0″ |
| 14,000 | 8′ 1″ | 15′ 1″ | 14′ 1″ | 21′ 1″ | 144 | 22′ 0″ | 8 | 12×12×14′ 0″ | 14 | 12×12×8′ 0″ |
| 16,000 | 8′ 1″ | 15′ 1″ | 15′ 1″ | 22′ 1″ | 145 | 22′ 1″ | 8 | 12×12×15′ 0″ | 15 | 12×12×8′ 0″ |

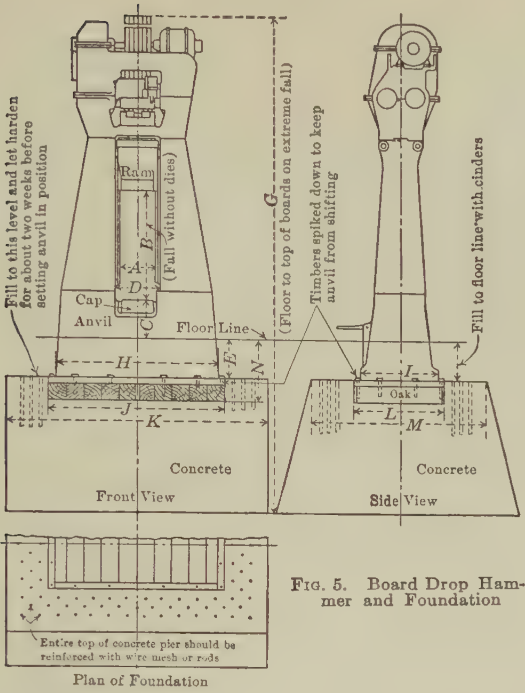
Fill to this level and let harden for about two weeks before setting anvil in position
Ram
B
(Full without dies)
A
D
Cap
Anvil
C
G
(Floor to top of boards on extreme fall)
Timbers spiked down to keep anvil from shifting
Floor Line
H
E
N
J
K
Concrete
Front View
I
Oak
L
M
Fill to floor line with cinders
Concrete
Side View
Entire top of concrete pier should be reinforced with wire mesh or rods
Plan of Foundation

FIG. 5. Board Drop Hammer and Foundation

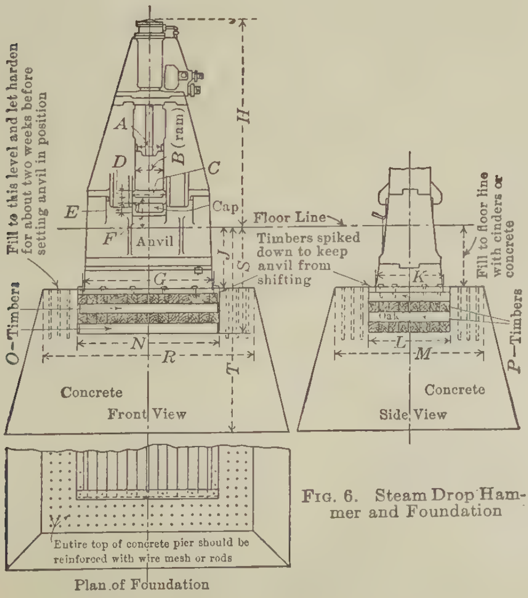
Fill to this level and let harden for about two weeks before setting anvil in position
H
A
B (ram)
D
C
E
Cap
Floor Line
F
Anvil
J
S
Timbers spiked down to keep anvil from shifting
G
O—Timbers
N
R
T
Concrete
Front View
K
Fill to floor line with cinders or concrete
Oak
Timbers
L
M
P—Timbers
Concrete
Side View
Entire top of concrete pier should be reinforced with wire mesh or rods
Plan of Foundation

FIG. 6. Steam Drop Hammer and Foundation

known of strains beyond elastic limits, that the practical application of formulas has not been developed.

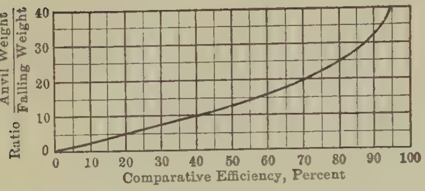

**Fig. 7.** Comparative Efficiency of Anvil Ratios

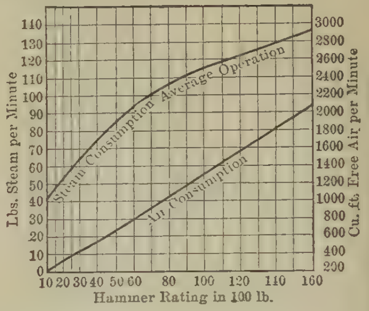

**Fig. 8.** Air and Steam Consumption of Power Drop Hammers

In the laboratories of the Chambersburg Engineering Company, Chambersburg, Pa., tests have been made in a specially constructed apparatus, which, with varying falling weights, velocities and anvils, simulates all conditions met in current practice. Not only are temperatures accurately controlled, but the rebound of the impacting die and the motion imparted to the entire structure are measurable. While useful data for designing hammers have been developed, all are empirical and incapable of reduction to formula. Three very useful points have been established, however, in the research conducted to the end of year 1936, *viz.*, 1. Within the range of present hammer velocities there is practically no difference in work accomplished by different falling weights and varying velocities to give the same available energy. Hence, within reason, it is desirable to forge on large rather than small hammers to save time and heat by shorter strokes; 2. The speeds of steam- or air-driven drop hammers as now constructed are near the economic limit; 3. The work done per blow is definitely related to the ratio of falling weight to anvil weight. Fig. 7 shows the relative efficiency of various ratios of anvil weight to falling weight. Ratio of falling weight to anvil weight of 20 : 1 or greater is standard practice for drop-hammers.

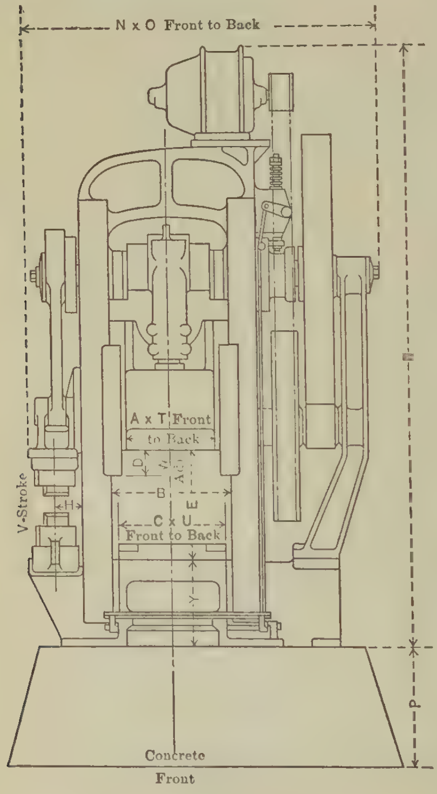

**Fig. 9.** Geared Trimming Press

**POWER CONSUMPTION OF DROP-HAMMERS.**—Gravity drop-hammers are driven by direct-connected motors, or are belt driven, and use energy only to lift the falling mass. Hence, the speed of the lift and friction within the mechanism readily indicate the power requirements. For individual drive it is common to calculate 1 Hp. per 100 lb. of rated falling weight. In group drive, the differential in operating cycles of the hammers will permit a reduction in peak load to as low as $3/4$ Hp. per 100 lb. of rated falling weight where six or more hammers of approximately one size are in the same group.

Steam- and air-driven hammer power consumption is shown in Fig. 8.

**Table 4.—Dimensions of Geared Trimming Press** (See Fig. 9)
Chambersburg Engineering Co., Chambersburg, Pa.

| Tons | 60 | 80 | 100 | 150 | 200 |
|---|---|---|---|---|---|
| Dimension *A*, in. | 11 | 15 | 19 | 22 | 26 |
| " *B*, in. | 18 | 20 | 25 | 30 | 34 |
| " *C*, in. | 15 | 18 | 22 | 27 | 31 |
| " *D*, in. | 4 | 4 | 5 | 6 | 6 |
| " *E*, in. | 20 1/2 | 22 | 26 1/2 | 28 | 31 1/2 |
| " *H*, in. | 5 | 5 | 6 1/4 | 7 | 7 |
| " *N*, in. | 60 | 63 | 83 | 92 | 100 |
| " *O*, in. | 36 | 44 | 66 | 74 | 78 |
| " *P*, in. | 30 | 30 | 30 | 30 | 30 |
| " *R*, in. | 118 | 120 | 146 | 159 | 175 |
| " *T*, in. | 12 | 16 | 20 | 24 | 27 |
| " *U*, in. | 20 | 24 | 27 | 33 | 36 |
| " *V*, in. | 3 | 4 | 5 | 5 | 5 |
| " *W*, in. | 3 | 3 | 4 | 4 | 5 |
| " *Y*, in. | 23 1/2 | 22 | 22 | 22 | 27 |
| Fly-wheel, rev. per min. | 280 | 274 | 267 | 250 | 225 |
| Motor, Hp. | 7 1/2 | 7 1/2 | 10 | 15 | 15 |
| " rev. per min. | 1150 | 1150 | 1150 | 1150 | 1150 |
| Gear ratio | 5 1/2–1 | 6–1 | 6 1/2–1 | 7.1–1 | 7 1/2–1 |
| Maximum stroke, in. | 6 | 8 | 8 | 9 | 15 |
| Standard stroke, in. | 5 | 5 1/2 | 6 | 6 | 7 |

**TRIMMING PRESSES.**—Drop-hammer forgings require trimming to remove the flash. Crank-driven power presses are employed. For hot trimming, such presses are provided with a side shear for cutting the sprue, or stub end, from a bar. A typical trimming press is shown in Fig. 9. Dimensions are given in Table 4.

## 2. POWER-HAMMER FORGING

**TYPES OF POWER HAMMERS.**—By far the majority of hammer forged products in the United States are produced in double-acting hammers, actuated by steam or compressed air. These tools, rated by the weight of the falling parts, are made in two types, single frame forging hammers (Fig. 12), and double frame forging hammers (Fig. 13). Both types are manually controlled by the hammer man to deliver a range of blows, from a squeeze to the maximum velocity of the particular size. Common sizes range from 1000 lb. falling weight to 16,000 lb. falling weight; smaller sizes listed in Table 5 are used for tool dressing and general maintenance work; larger sizes than those listed in Table 6 usually are specially designed for a particular product. The most important individual characteristic of forging hammers is that the anvil is a separate structure from the frames and is mounted on a separately constructed foundation. (See Fig. 12.)

Recommended hammer sizes for the various sizes of forging stock are shown by Fig. 10.

**POWER CONSUMPTION** of double-acting hammers varies with every job, and when several hammers are served by a single power plant the stand-by requirements are materially reduced. Fig. 11 indicates the stand-by requirements of a single hammer for nor-

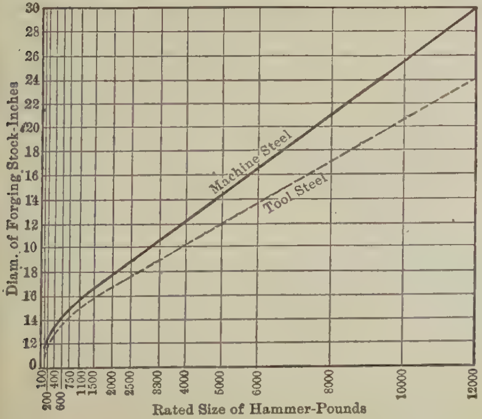

FIG. 10. Sizes of Forging Hammers for Various Sizes of Stock

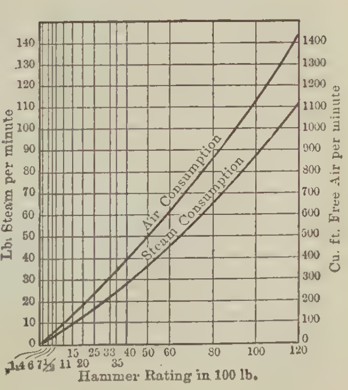

FIG. 11. Steam and Air Consumption of Forging Hammers

mal operation. For a group of six or more hammers, the stand-by requirements are frequently reduced 60%. No general data on power consumption are reliable, and the manufacturer should be consulted for specific information.

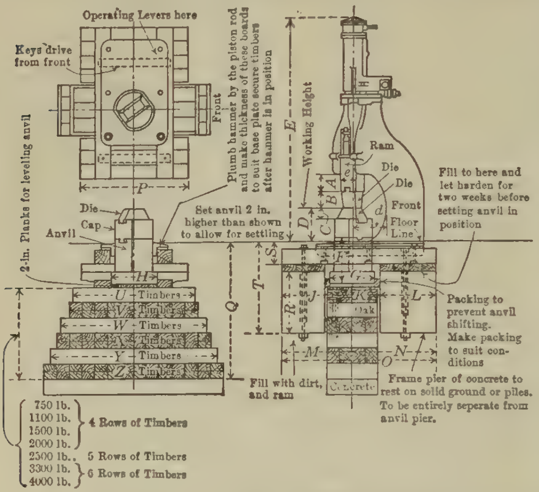
Operating Levers here
Keys drive from front
Front
Plumb hammer by the piston rod and make thickness of these boards to suit base plate secure timbers after hammer is in position
E
Working Height
Ram
Die
Die
Front
Floor Line
e
A
B
C
D
d
Fill to here and let harden for two weeks before setting anvil in position
P
2-in. Planks for leveling anvil
Die
Cap
Anvil
Set anvil 2 in. higher than shown to allow for settling
S
F
G
H
U Timbers
V Timbers
W Timbers
X Timbers
Y Timbers
Z Timbers
Q
T
R
J
K
Oak
L
Packing to prevent anvil shifting. Make packing to suit conditions
M
N
O
Fill with dirt, and ram
Concrete
Frame pier of concrete to rest on solid ground or piles. To be entirely seperate from anvil pier.
750 lb.
1100 lb.
1500 lb.
2000 lb.
4 Rows of Timbers
2500 lb.. 5 Rows of Timbers
3300 lb.
4000 lb.
6 Rows of Timbers

Fig. 12. Single-frame Forging Hammer

**WEIGHT OF THE ANVIL** as compared with the weight of the falling parts has a marked effect on the amount of forging accomplished per blow. Fig. 7 shows the relative efficiency of various ratios of anvil weight to falling weight. It has been found that a 15 : 1 anvil is the most economical for this type of hammer.

**HELVE- AND SPRING-HAMMERS.**—In addition to double-acting forge hammers, helve hammers, spring hammers, and self-contained pneumatic hammers also are employed for occasional specialized jobs. Helve hammers, as the name implies, carry the hammer head and upper die on a crank-driven helve. These hammers principally are used for drawing, tube shaping, etc.

**Table 5.—Dimensions of Single Frame Forging Hammers and Foundations** (See Fig. 12)

Chambersburg Engineering Co., Chambersburg, Pa.

| Rated Falling Weight, lb. | Hammer Dimensions | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Bore, in. | Stroke, in. | *A*, in. | *B*, in. | *C*, in. | *D*, in. | *d*, in. | *E* | *e*,* in. | *F*, in. | *G*, in. | *H*, in. |
| 750 | 7 | 23 | 8 | 10 | 7 | 26 | 18 | 11′ 0 3/4″ | 12 | 34 1/2 | 28 | 60 |
| 1100 | 10 | 28 | 11 | 13 | 7 | 26 | 23 | 13′ 3 1/2″ | 13 1/2 | 39 | 32 | 60 |
| 1500 | 12 | 33 | 13 1/2 | 13 | 8 | 27 | 25 | 14′ 4 1/2″ | 16 | 43 | 36 | 72 |
| 2000 | 13 | 36 | 12 | 16 | 8 | 27 | 28 | 15′ 2 3/4″ | 16 | 52 1/2 | 40 | 72 |
| 2500 | 14 | 39 | 16 | 16 | 9 | 27 | 32 | 16′ 7 1/2″ | 16 1/4 | 46 | 50 | 96 |
| 3300 | 15 | 42 | 20 | 16 | 9 | 27 | 38 | 17′ 2 1/2″ | 16 1/2 | 54 1/2 | 54 | 96 |
| 4000 | 16 | 42 | 20 | 16 | 9 | 27 | 38 | 17′ 5 1/2″ | 18 1/2 | 58 1/2 | 54 | 96 |

| Rated Falling Weight, lb. | Foundation Dimensions | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *J*, in. | *K*, in. | *L*, in. | *M*, in. | *N*, in. | *O* | *P* | *Q* | *R*, in. | *S*, in. | *T*, in. |
| 750 | 36 | 40 | 48 | 56 | 68 | 10′ 4″ | 7′ 0″ | 7′ 0 1/2″ | 36 | 18 | 58 |
| 1100 | 36 | 40 | 48 | 56 | 68 | 10′ 4″ | 7′ 0″ | 7′ 5″ | 40 | 18 | 62 1/4 |
| 1500 | 36 | 52 | 48 | 62 | 74 | 11′ 4″ | 7′ 0″ | 7′ 9″ | 48 | 18 | 70 1/2 |
| 2000 | 36 | 52 | 48 | 62 | 74 | 11′ 4″ | 7′ 0″ | 8′ 6 1/2″ | 48 | 18 | 70 1/2 |
| 2500 | 36 | 58 | 48 | 65 | 77 | 11′ 10″ | 8′ 0″ | 9′ 0″ | 54 | 18 | 77 |
| 3300 | 36 | 70 | 48 | 71 | 83 | 12′ 10″ | 9′ 0″ | 10′ 8 1/2″ | 60 | 24 | 89 1/2 |
| 4000 | 36 | 70 | 48 | 71 | 83 | 12′ 10″ | 9′ 0″ | 11′ 0 1/2″ | 60 | 24 | 89 1/2 |

| Rated Falling Weight, lb. | Foundation Timbers | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *U*-Timbers | | *V*-Timbers | | *W*-Timbers | | *X*-Timbers | | *Y*-Timbers | | *Z*-Timbers | |
| | No. | Size | No. | Size | No. | Size | No. | Size | No. | Size | No. | Size |
| 750 | 3 | 12×12×7′ 6″ | 8 | 12×12×3′ 0″ | 3 | 12×12× 9′ 0″ | 10 | 12×12×3′ 0″ | .. | .......... | .. | ...... |
| 1100 | 3 | 12×12×7′ 6″ | 8 | 12×12×3′ 0″ | 3 | 12×12× 9′ 0″ | 10 | 12×12×3′ 0″ | .. | .......... | .. | ...... |
| 1500 | 4 | 12×12×8′ 0″ | 9 | 12×12×4′ 0″ | 4 | 12×12× 9′ 6″ | 10 | 12×12×4′ 0″ | .. | .......... | .. | ...... |
| 2000 | 4 | 12×12×8′ 0″ | 9 | 12×12×4′ 0″ | 4 | 12×12× 9′ 9″ | 11 | 12×12×4′ 0″ | .. | .......... | .. | ...... |
| 2500 | 4<br>1 | 12×12×9′ 0″<br>12× 6×9′ 0″ | 10 | 12×12×4′ 6″ | 4<br>1 | 12×12×10′ 6″<br>12× 6×10′ 6″ | 11 | 12×12×4′ 6″ | 4<br>1 | 12×12×11′ 6″<br>12× 6×11′ 6″ | ..<br>.. | ......<br>...... |
| 3300 | 5<br>1 | 12×12×9′ 0″<br>12× 6×9′ 0″ | 10 | 12×12×5′ 6″ | 5<br>1 | 12×12×11′ 0″<br>12× 6×11′ 0″ | 12 | 12×12×5′ 6″ | 5<br>1 | 12×12×12′ 6″<br>12× 6×12′ 6″ | 13 | 12×12 ×5′ 6″ |
| 4000 | 5<br>1 | 12×12×9′ 0″<br>12× 6×9′ 0″ | 10 | 12×12×5′ 6″ | 5<br>1 | 12×12×11′ 0″<br>12× 6×11′ 0″ | 12 | 12×12×5′ 6″ | 5<br>1 | 12×12×12′ 6″<br>12× 6×12′ 6″ | 13 | 12×12 ×5′ 6″ |

*Between guides.

Self-contained pneumatic hammers have a crank-driven compressor piston, the chamber of which is connected to the hammer cylinder by proper valving. Limited in speed to that of the compressor crank, such hammers have characteristics which confine their use to certain specialized jobs.

**THE PRODUCTS OF THE HAMMER SHOP** cover a wide range of use, from machine tool bits to large crank-shafts and special shapes. In almost every case the subject material is steel, and in finishing, allowance must be made for the contraction occurring as cooling takes place from the forging temperature to atmospheric temperature.

**COMMERCIAL TOLERANCES** on size are given in Table 7 by E. J. Poole, Jr., Forging in a Tool and Alloy Steel Plant, *Heat Treating and Forging*, March, 1931. Forgings usually are made full to forging size and a tolerance allowed from the new size.

**FORGING METHODS.**—The selection and heating of forging stock, and its annealing and inspection, are beyond the scope of this publication. For such detail it is suggested that reference be made to Recommended Practice for Large Forgings, by John L. Cox, Metals Handbook, 1936 edition.

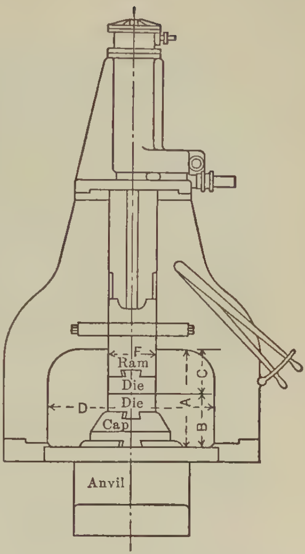

Fig. 13. Double-frame Forging Hammer

In forging operations, ingot reduction should be slight at the start and uniform on every face to properly consolidate the ingot structure. Too rapid operation will cause transverse fractures in the center, and may produce mechanical piping by working the surface metal faster than the center. It is important to cause the metal to flow in proper relation to the stress to be imposed on the subject. In shaping forgings to the desired form the preservation of proper grain flow is important.

**THE HAMMER FORGE SHOP** employs auxiliary equipment such as furnaces, work-holding and forming tools, crane and hoist material-handling means, and, for larger hammers, manipulators for the stock. A typical layout for the manufacture and maintenance of railroad forgings is shown in Fig. 14, a reproduction of the layout of the shops of the Norfolk and Western Railroad at Roanoke, Va.

**Table 6.—Dimensions of Double-frame Forging Hammers** (See Fig. 13)

| Rated Falling Weight, lb. | Dimensions | | | | | |
|---|---|---|---|---|---|---|
| | *A*, in. | *B*, in. | C, in. | *D* | *E* in. | Die Face, in. |
| 1000 | 41 | 27 | 14 | 4′ 6″ | 15 3/4 | 6 × 13 1/4 |
| 1500 | 43 1/2 | 27 | 16 1/2 | 5′ 0″ | 17 3/4 | 7 × 15 |
| 2000 | 43 1/2 | 27 | 16 1/2 | 5′ 6″ | 18 1/4 | 7 1/2 × 16 |
| 2500 | 47 | 27 | 20 | 6′ 0″ | 19 3/4 | 8 × 17 |
| 3000 | 47 | 27 | 20 | 6′ 6″ | 22 | 8 × 18 |
| 3500 | 47 | 27 | 20 | 6′ 6″ | 23 | 8 1/2 × 19 |
| 4000 | 50 | 27 | 23 | 7′ 0″ | 23 1/4 | 9 × 20 |
| 5000 | 50 | 27 | 23 | 7′ 10 1/2″ | 25 3/4 | 11 × 22 |
| 6000 | 50 | 27 | 23 | 8′ 6″ | 29 | 12 × 25 |
| 8000 | 51 | 27 | 24 | 9′ 0″ | 32 | 14 × 26 |
| 10000 | 52 | 28 | 24 | 10′ 0″ | 32 1/2 | 15 × 27 |
| 12000 | 63 | 30 | 33 | 10′ 0″ | 33 1/2 | 16 × 28 |
| 14000 | 63 | 30 | 33 | 10′ 0″ | 33 1/2 | 17 × 29 |
| 16000 | 63 | 30 | 33 | 10′ 6″ | 34 1/2 | 18 × 30 |
| 20000 | 63 | 30 | 33 | 10′ 9″ | 37 1/2 | 19 × 32 |
| 24000 | 63 | 30 | 33 | 10′ 9″ | 37 1/2 | 20 × 33 |

**Table 7.—Commercial Tolerances for Forgings**

| Specified Dimension, in. | Addition for Forging Size, in. | Tolerance Allowed from Forging Size, in. |
|---|---|---|
| Up to 1 1/4 | 1/32 | ±0.015 |
| Over 1 1/4 to 2 | 3/64 | ± .015 |
| Over 2 to 3 | 5/64 | ± .025 |
| Over 3 to 4 | 3/32 | ±1/32 |
| Over 4 to 5 | 7/64 | ±3/64 |
| Over 5 to 6 | 9/64 | ±1/16 |
| Over 6 to 7 | 5/32 | ±5/64 |
| Over 7 to 8 | 3/16 | ±3/32 |
| Over 8 | 1/4 | ±1/8 |

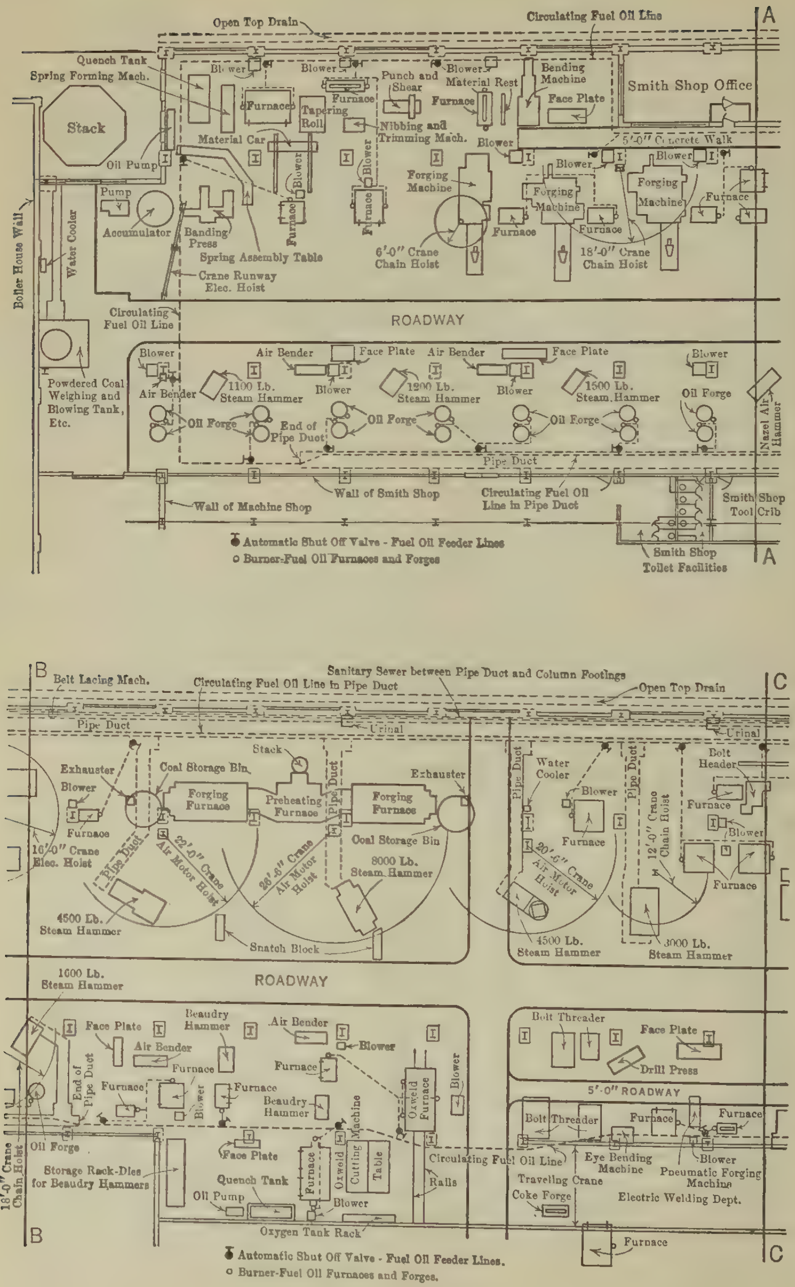

FIG. 14. Railroad Forge Shop Layout

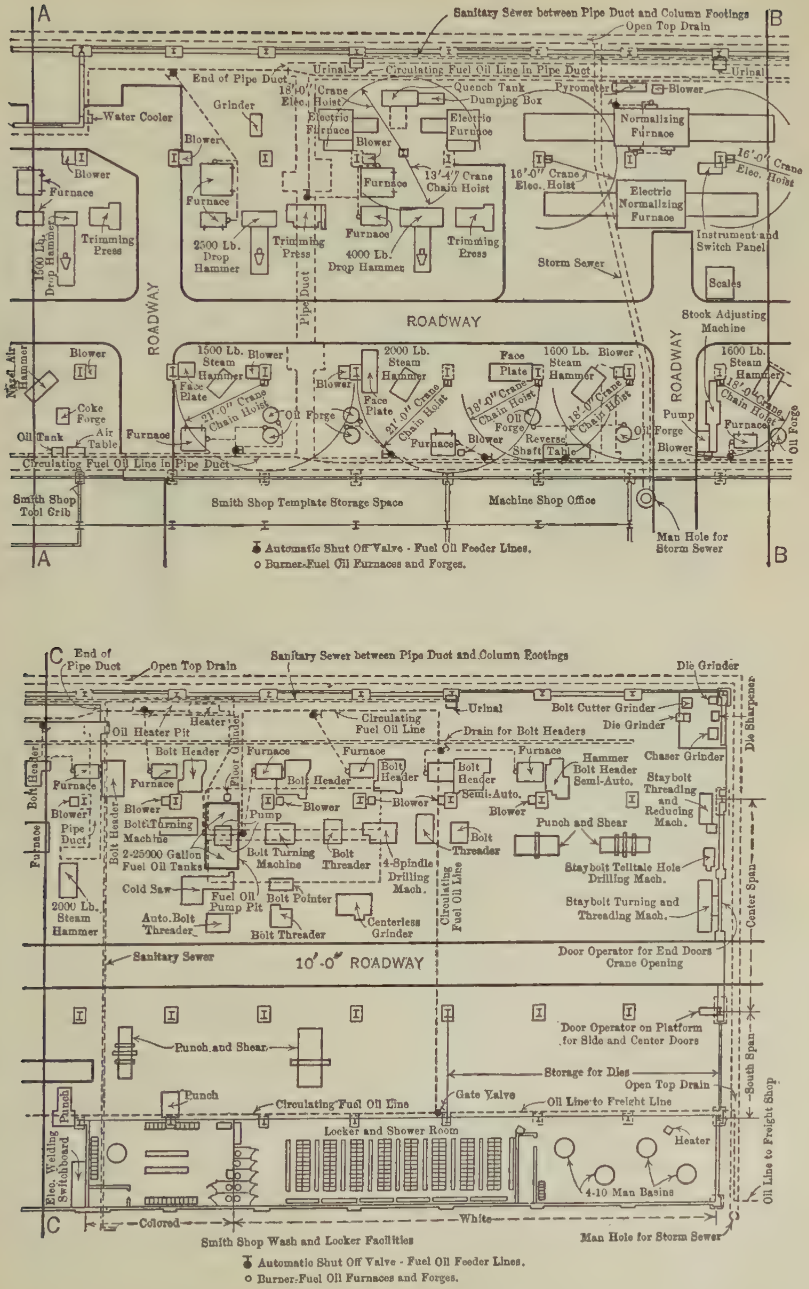

FIG. 14—*Continued.* Railroad Forge Shop Layout

## 3. HYDRAULIC FORGING *

Forging in hydraulic presses particularly lends itself to piercing, drawing, and extruding. The plastic deformations which occur in these operations are diagrammatically indicated in Fig. 15.

**PIERCING.**—In piercing, the shape of the punch has an influence on the flow. If the punch tip is pointed the layers are ruptured in a relatively shorter depth of piercing.

**Pressure of Fluidity,** that is, the point at which flow begins, increases as temperature decreases. Assume that at the temperature of the point of incipient fusion the pressure of fluidity at low velocities equals zero. From observation, and accepting that the pressure increases by a straight line law as the temperature falls, the pressure of fluidity may be calculated. For 0.45% carbon steel, observed pressure of fluidity at 1470° F. is $11\,^3/_4$ tons per sq. in., and at 2470° F., the region of incipient infusion, the pressure of fluidity is 0. Then $P_f$, the pressure of fluidity between these points, the velocity being slow is $P_f = 0.0118\ (2460 - T)$, where $T$ = temperature of the metal. Variations in carbon content in pearlite steel are insufficiently great to seriously affect this formula.

Pressure to produce flow, $P_m$, when the upward velocity of the pierced metal appreciably exceeds the downward velocity of the punch, can be determined within practical working limits by the formula $P_m = P_f\,[\{d^2/(D^2 - d^2)\} + 1]$, where $d$ and $D$ = diameter of piercing tool, and of the die, respectively. These conclusions will be modified if the velocity of the piercing tool deviates greatly from the normal speed of 3 in. per second common to shell piercing operations. Appreciably slower speeds will result in reduced flow, because of the cooling of the metal. Allowance will be required if the tools are not in normally good condition with regard to surface defects. If the piercing tool is tapered, greater pressure is required, because of frictional resistance on the tapered section. See chapter on Mechanical Treatment, in Bower, Hydraulic Forging, for formulas reflecting varying conditions.

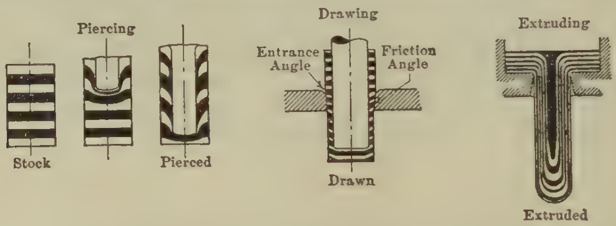

FIG. 15. Plastic Flow in Hydraulic Press Operations

**DRAWING** of shells or tubes generally is limited by the stress in the walls of the shell in tension. It is undesirable to stress steel walls more than 50% of their strength at 1650° F. A difference in temperature of 200° F. easily may exist on the two sides of the drawing die.

**EXTRUDING** † is limited in its usefulness in so far as hydraulic forging of steel is concerned. It is a major operation in the production of various cross-sectional shapes of lead, tin, zinc, copper, aluminum, etc. Such shapes in the form of rods or bars are made by forcing the metal through an aperture die by high pressure, producing smooth surfaces, free from flaws, to close dimensions, and generally imparting higher physical properties to the subject.

**PUNCHES AND DIES.**—Punches must be rigid to maintain concentricity, and tough to withstand abrasive action. Chrome-nickel steels generally have the longest life. Since scoring due to abrasion starts at the tip, renewable tough cast-iron tips frequently are used, in order to reduce the expense of renewals. Cooling and lubrication are required between piercing operations.

Dies for piercing operations commonly are constructed in the form of a steel jacket completely enclosing a renewable cast-iron liner, bored to suit the outside diameter required of the forging. Such dies have stripper plates to strip the forging from the punch, and the larger sizes are slidably mounted on the lower platen of the press to permit mechanical handling.

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* The material here presented is principally an abstract of Hydraulic Forging, by Clifford O. Bower, printed by The Library Press, Ltd., London, England.

† Extrusion of Metals, by D. K. Crampton, *Trans.*, A.S.M., vol. xxv, 1937, contains much detailed information on extrusion of non-ferrous metals.

The mean average life of punches and die liners for forgings over 6 in. diameter is approximately 600 forgings. Smaller forgings may result in tool life of double or triple this amount.

Drawing dies are made of the hardest close-grained iron. It is common practice to use 20 deg. as an entrance angle, and 12 deg. as the friction angle of the die. See Fig. 15. Some dies have a curved entrance. Dies may be mounted in series for multiple drawing operations. The larger drawn forgings are produced on horizontal presses called *draw benches.*

**HYDRAULIC FORGING PRESSES.**—The piercing and drawing of steel forgings requires high pressures, which are most satisfactorily applied by means of hydraulically operated presses. The simplest forms are driven from constant-pressure systems, pressures of 1500 lb. or 2000 lb. per sq. in. being most general. The return stroke of the press ram usually is accomplished by return rams. Double-acting cylinders are uncommon, because of the inconvenience of packing. The simplicity of such presses permits construction adaptable to all conditions, and standards of working space, stroke, etc., are not established. Fig. 16 shows a typical shell drawing press.

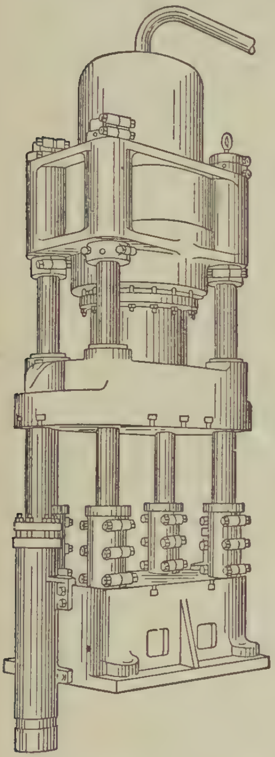
Fig. 16. Shell Drawing Press

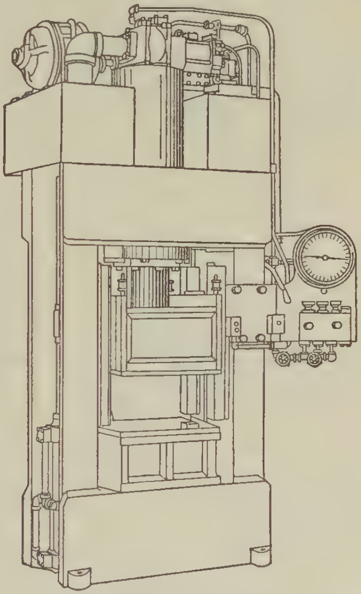
Fig. 17. Self-contained Drawing Press

Multi-pressure systems, using low pressure for non-working traverse motions, embody more complicated piping and valves. Such systems sometimes are more economical, but the economies should be proven before these systems are adopted.

For light operations and for non-ferrous metals, variable pressure systems are applicable. Self-contained presses, utilizing motor-driven rotary pumps and greater volumes of lower pressure fluid, are employed on light drawing and piercing work. Fig. 17 illustrates this type of press.

## 4. STEAM HYDRAULIC PRESSES

The reduction of ingots, the manufacture of large forgings and the refining of large sections, such as hammer die blocks, all require deep penetration into the mass of the forging. The steam hydraulic press is used for such work.

As the name implies, the presses are steam powered. The *press head* is lowered to the work by gravity, water being forced into the cylinder by air pressure in the storage tank. Steam admitted to the intensifier raises the water pressure during forging operations to as high as 6000 lb. per sq. in. Steam also is used to return the ram to its upper position.

The working spaces available in standardized steam hydraulic presses of the four-

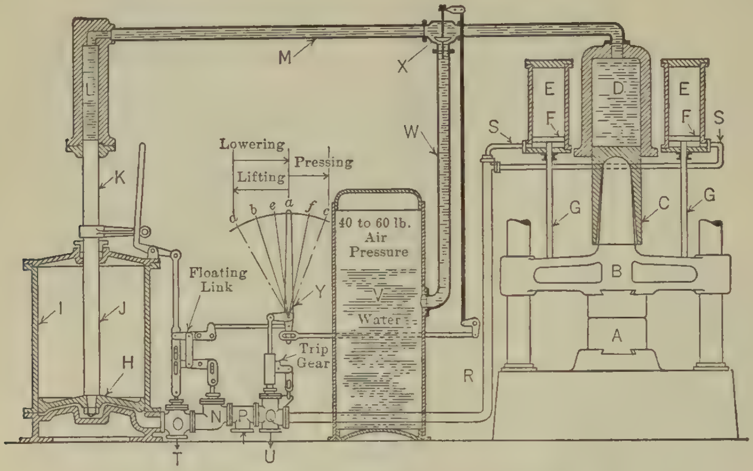

FIG. 18. Diagrammatic Arrangement of Steam Hydraulic Press

| | |
|---|---|
| *A* | Bottom Die |
| *B* | Press Head |
| *C* | Main Hydraulic Ram |
| *D* | Hydraulic Ram Cylinder |
| *E* | Steam Pull-back Cylinder |
| *F* | Steam Pull-back Pistons |
| *G* | Pull-back Piston Rods |
| *H* | Intensifier Piston |
| *I* | Intensifier Cylinder |
| *J* | Intensifier Piston Rod |
| *K* | Intensifier Ram Cylinder |
| *L* | Intensifier Ram Cylinder |
| *M* | Hydraulic High-pressure Pipe |
| *N* | Intensifier Inlet Valve |
| *O* | Intensifier Exhaust Valve |
| *P* | Inlet Boiler Connection |
| *Q* | Pull-back Valve |
| *R* | Pull-back Pipe Connection |
| *S* | Pull-back Cylinder Connections |
| *T* | Intensifier Exhaust Connection |
| *U* | Pull-back Exhaust Connection |
| *V* | Prefiller |
| *W* | Prefiller High-pressure Connection |
| *X* | Hydraulic Valve to Prefiller |
| *Y* | Operating Hand Lever |

**Table 8.—Dimensions of Four-column Forging Press** (See Fig. 19)

Chambersburg Engineering Co., Chambersburg, Pa.

| Nominal Rating of Press, tons | 500 | 750 | 1000 | 1500 | 2000 |
|---|---|---|---|---|---|
| Movement of Presshead, per stroke of intensifier, in.*.... | 5.43 | 6.00 | 6.64 | 7.50 | 8.21 |
| Approximate size of ingot, in.. | 18 | 24 | 28 | 40 | 48 |
| Dimension A............... | 2′ 4″ | 3′ 0″ | 3′ 8″ | 4′ 5″ | 5′ 8″ |
| " *B*............... | 5′ 0″ | 6′ 0″ | 7′ 0″ | 9′ 0″ | 10′ 6″ |
| " *C*............... | 2′ 6″ | 3′ 0″ | 3′ 6″ | 4′ 0″ | 4′ 0″ |
| " *D*............... | 2′ 8″ | 3′ 3″ | 3′ 11″ | 5′ 4″ | 6′ 5″ |
| " *E*............... | 10′ 8″ | 11′ 9″ | 13′ 6″ | 17′ 0″ | 21′ 3″ |
| " *F*............... | 14′ 8″ | 17′ 5″ | 19′ 5″ | 22′ 11″ | 25′ 1″ |
| " *G*............... | 22′ 8″ | 25′ 11″ | 28′ 2″ | 33′ 3″ | 37′ 3″ |
| " *H*............... | 7′ 7″ | 8′ 9″ | 9′ 8″ | 11′ 7″ | 13 3″ |
| " *J*............... | 2′ 0″ | 3′ 0″ | 3′ 4″ | 4′ 0″ | 4′ 6″ |
| " *K*............... | 9′ 9″ | 11′ 1″ | 12′ 3″ | 14′ 1″ | 15′ 9″ |
| " *L*............... | 11′ 0″ | 12′ 0″ | 13′ 0″ | 14′ 0″ | 15′ 9″ |
| " *M*............... | 12′ 8″ | 14′ 6″ | 16′ 4″ | 19′ 4″ | 23′ 6″ |
| " *N*............... | 14′ 0″ | 17′ 0″ | 19′ 6″ | 20′ 8″ | 22′ 0″ |
| " *O*............... | 19′ 10″ | 23′ 2″ | 25′ 2″ | 30′ 6″ | 33′ 11″ |
| " *P*............... | 13′ 0″ | 15′ 0″ | 16′ 0″ | 19′ 0″ | 20′ 6″ |
| " *Q*............... | 4′ 11″ | 5′ 10″ | 6′ 4″ | 7′ 3″ | 8′ 6″ |
| " *R*............... | 7′ 4″ | 8′ 2″ | 8′ 9″ | 9′ 11″ | 11′ 1″ |
| " *S*............... | 4′ 6″ | 4′ 11″ | 5′ 10″ | 5′ 3″ | 4′ 2″ |
| " *T*............... | 4′ 9″ | 5′ 9″ | 6′ 4″ | 8′ 2″ | 9′ 9″ |

* Without resistance. At full load, actual movement is about 15% less.

column type together with dimensions are given in Table 8 and Fig. 19. Similar information for open-frame presses is given in Table 9 and Fig. 20.

**TOOLS.**—The mass of the forgings worked in steam hydraulic presses generally requires material handling equipment to manipulate the stock. This handling usually is done either by overhead cranes with power-driven chains for rotating the work, or by *manipulators* which grasp the forging and position it between the dies of the press. Mechanical means for handling cutting off tools and the like also is desirable with the very large presses.

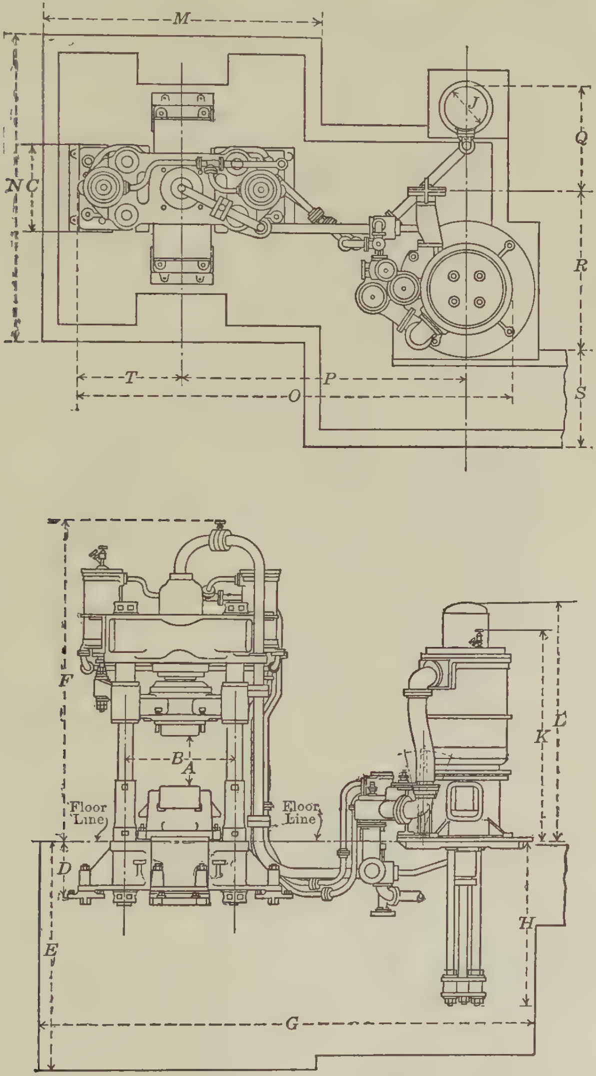

Fig. 19. Four-column Steam-hydraulic Press

**METHODS.**—Refer to Forging Methods, Hammer Forging, p. 19–04. The same methods are applicable to press forging.

**POWER CONSUMPTION.**—Neither the water nor the air employed in the steam hydraulic press are lost in operation. The steam is used only as forging pressure is applied

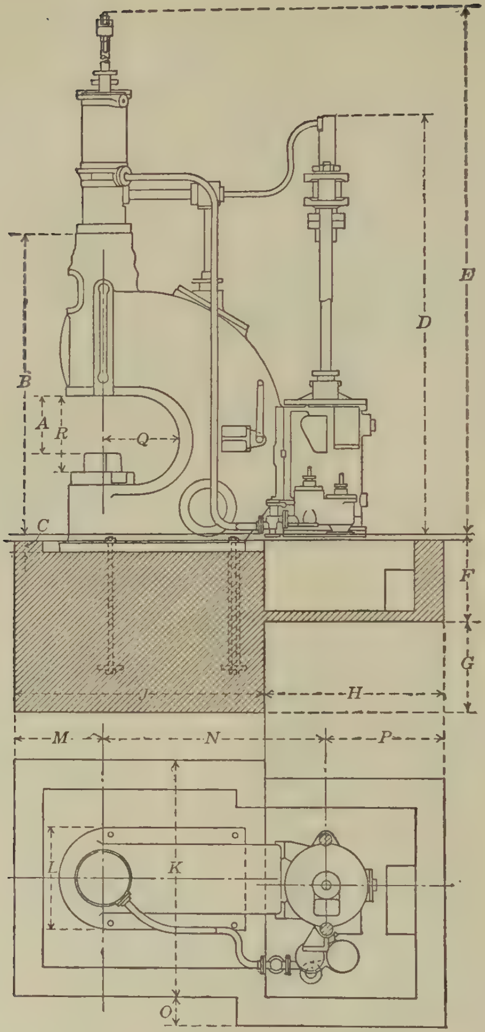

Fig. 20. Open-frame Steam-hydraulic Press

**Table 9.—Dimensions of Open-frame Steam Hydraulic Presses** (See Fig. 20)
Chambersburg Engineering Co., Chambersburg, Pa.

| 150-ton Press | | | | | | 300-ton Press | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Stroke, 18 in.<br>Presshead moves 3 in. per stroke of intensifier<br>Size of ingot (approx.), 10 in. | | | | | | Stroke, 24 in.<br>Presshead moves 4 in. per stroke of intensifier<br>Size of ingot (approx.), 14 in. | | | | | |
| Dimensions, in. | | | | | | Dimensions, in. | | | | | |
| *A* | *B* | *C* | *D* | *E* | *F* | *A* | *B* | *C* | *D* | *E* | *F* |
| 17 7/8 | 95 | 11 1/2 | 135 | 176 3/4 | 27 | 24 1/4 | 119 1/4 | 19 1/4 | 178 | 220 3/4 | 27 |
| *G* | *H* | *J* | *K* | *L* | *M* | *G* | *H* | *J* | *K* | *L* | *M* |
| 35 1/2 | 60 | 79 | 74 | 32 | 28 | 64 1/4 | 65 | 114 | 96 | 39 | 40 |
| *N* | *O* | *P* | *Q* | *R* | | *N* | *O* | *P* | *Q* | *R* | |
| 72 | 11 | 39 | 24 | 30 | | 93 | 3 | 46 | 33 | 38 1/4 | |

or released, and consequently is in direct proportion to the work done. The nature of the work, pressure required, temperature and characteristics of the subject material are so variable that a general statement of steam consumption is impractical.

## 5. MACHINE FORGING

The displacement of metal in closed dies, one of which is reciprocated by a crank mechanism, is plastic flow induced principally by upsetting and by piercing, or by both, for the mass production of identical shapes. Since each operation fills the die cavity under high hydraulic-like pressure, the stationary, or *grip*, dies are split to permit placing and extraction of the stock. The grip dies hold the stock, by frictional grip, against the forging force of the heading tool. Bar stock normally is used, and successive deformations to complete the shape are done in multi-impression dies. Positioning stops for each impression, and a *cut-off* to shear the finished forging from the bar are provided.

The plastic flow engendered in machine forging is illustrated by Fig. 21, wherein both upsetting and piercing have contributed to the desirable internal structure. In properly-designed dies, the stock is plastically flowing under compression in all impressions, resulting in a radial grain flow and refinement that is essential to a great variety of forgings, such as gear blanks, bearing races, etc.

**DIES.**—In a paper, Laws Governing Forging Machine Die Design, before the Am. Drop Forge Assoc., E. R. Frost gives three rules as follows: Let $d$ = diam. of bar.

**1.**—The limit of length of unsupported stock that can be gathered or upset in one blow without injurious buckling is not more than $3d$.

**2.**—Lengths of stock more than $3d$ can be successfully upset in one blow, provided the diameter of the upset made is not more than $1\,{}^1/_2\,d$.

**3.** In an upset requiring more than three diameters of stock in length, and in which the diameter of the upset is $1\,{}^1/_2\,d$, the amount of unsupported stock beyond the face of the die must not exceed $1d$.

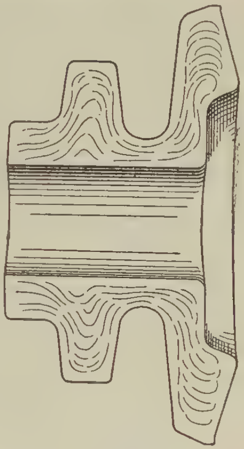

Fig. 21. Grain Flow in Machine Forging

The dies in a forging machine must resist pressure, abrasion and heat. In automatically-fed machines, the drawing of the temper of the dies generally reduces the forgings per dressing well below the number of forgings secured per dressing when hand fed. Cold-heading operations, as used in forming rivet heads, enjoy the longest die life. High chrome-nickel or vanadium steels, or other alloys, which retain their characteristics at relatively high temperatures are desirable when large radiation surfaces occur in the forming operation. Before being put in service, dies should be checked for hardness, which should conform to the requirements of the particular operation while avoiding excess hardness that may introduce brittleness. Fig. 22 indicating practical selection of die steels is reproduced from The Selection of Steels for Forging Machine Dies, E. R. Frost. Metals Handbook, 1936 edition, should be consulted for details regarding the selection and heat treatment of dies for machine forging, cold heading, etc.

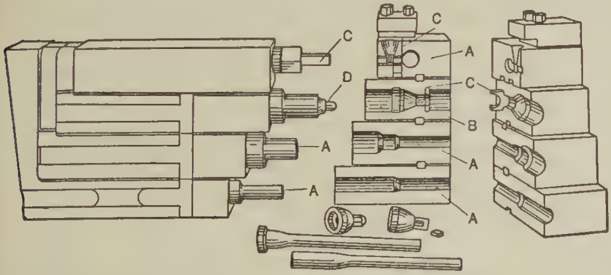

Fig. 22. Die Set-up for Forging Machine

*A* Straight Carbon Steel
*B* Chrome Steel
*C* High Tungsten Steel
*D* Low Tungsten Steel

**FORGING MACHINES.**—Requirements of the modern forging machine are: Grip sufficiently tight to prevent die separation and consequent fins; maintenance of alignment in both the gripping and upsetting mechanism to insure concentricity and to avoid breakage of punches; rapid starting and stopping; ease of control by the operator. Of major importance is the rigidity of the frame enclosing the mechanism. This must be extremely massive, and currently (1937) is made of a single steel casting, heat treated, and prestressed with heavy steel bars, shrunk in place. Commercial forging machines are nominally rated in inches of stock, which rating usually requires interpretation by the builder.

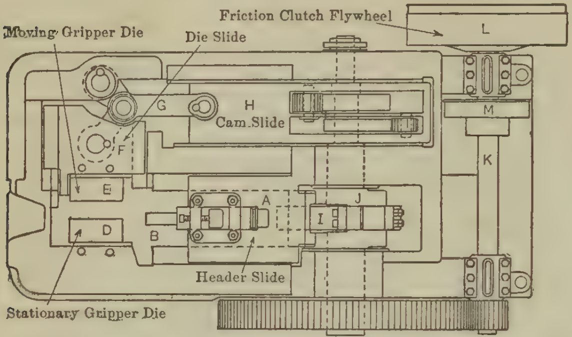

Fig. 23. Diagrammatic Arrangement of Horizontal Upset Forging Machine

Fig. 23 is a diagrammatic arrangement of a horizontal upset forging machine. The header-slide *A* carries the tool *B*. The bar from which the forging is to be made is gripped between the holding dies *D* and *E*, *E* being moved horizontally by the slide *F* at right angles to the movement of slide *A*. Movement of *F* is effected by the toggle *G*, which is actuated from the cam slide *H*, by complementary cams on the main crank-shaft. The header-slide is driven by the pitman *I* from the crank-shaft *J*. The driving shaft *K* is driven by a motor through the clutch flywheel *L*. In this forging machine, the clutch is a pneumatically operated friction clutch. Air is admitted to the clutch by a foot-controlled valve at the operator's end of the machine. When the crank-shaft has completed one revolution, a cam opens the valve and actuates a band brake *M* to bring the machine to rest. The die blocks *D* and *E*, actually may carry two or more impressions, and the header-slide two or more working tools, one above the other, each of which performs one of the successive forging operations necessary to complete the work.

Table 10 gives the specifications of typical heavy-duty horizontal forging machines.

**Table 10.—Specifications of High Duty Forging Machines**

National Machinery Co., Tiffin, O.

| Designated size, in. | 1 | 1 1/2 | 2 | 2 1/2 | 3 | 4 | 5 | 6 | 7 1/2 |
|---|---|---|---|---|---|---|---|---|---|
| Speed, r.p.m. | 90 | 75 | 60 | 55 | 45 | 35 | 30 | 27 | 25 |
| Hp. of Motor | 7 1/2 | 10 | 15 | 20 | 25 | 35 | 50 | 75 | 150 |
| Speed of Motor, r.p.m. | 900 | 720 or 900 | 720 or 900 | 720 | 720 | 720 | 720 | 600 | 600 |
| Floor Space, Length | 5′ 0″ | 5′ 11″ | 9′ 6″ | 9′ 8″ | 11′ 2″ | 13′ 0″ | 14′ 2″ | 16′ 3″ | 19′ 5″ |
| Floor Space, Width | 6′ 0″ | 6′ 9″ | 8′ 5″ | 8′ 8″ | 9′ 10″ | 10′ 8″ | 11′ 7″ | 14′ 2″ | 16′ 8″ |
| Net Weight, lb. | 13,000 | 23,500 | 37,000 | 43,000 | 75,000 | 120,000 | 170,000 | 250,000 | 375,000 |

**MECHANICAL FORGING PRESSES.**—For accurate sizing, coining, and the production of relatively simple symmetrical forgings, the mechanical forging press is a valuable contribution to processes, in that it reduces subsequent machining operations. The excessive rigidity and freedom from stretch built into the frame is of primary importance. The stationary die employed is not parted, as in the forging machine, but is solid. In other respects the die requirements are similar to those of the forging machine.

**GENERAL.**—The application of crank-driven forging machines to production of bolts, nuts, rivets, as well as to the manufacture of mechanical parts, has encouraged the development of many special machines for which specific information cannot be given. The manufacturers of such machines should be consulted.

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# Section 20

## FOUNDRY PRACTICE

By D. M. Avey

## MALLEABLE CASTINGS

By Enrique Touceda

## NON-FERROUS FOUNDRY PRACTICE

By N. K. B. Patch

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# FOUNDRY PRACTICE

By D. M. Avey

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## 1. GRAY IRON CASTINGS

**DEFINITION.**—Gray cast iron covers a wide range of alloys. Cast iron is a combination of iron with carbon, and other elements as silicon, manganese, phosphorus and sulphur. Nickel, chromium, molybdenum and other elements are used to obtain certain effects in the cast product. The fundamental governing material is graphitic carbon, associated with pearlite, ferrite, or both. Most commercial gray irons contain 2.50 to 3.75% total carbon (T.C.).

**USES.**—Cast iron is extremely important in the manufacture of machinery. Where bulk and inertia are important, its uses are well known. Machine tool frames and bases are made of cast iron in intricate forms, with high accuracy to finished dimension without undue machining. Machining time is a minor consideration in ordinary gray iron, as the material cuts rapidly, and, when properly made, has a smooth accurate surface which shows regularity and fine grain under the microscope. Its low cost recommends it for general purposes, limited by its physical properties and by the detailed method of manufacture.

**PHYSICAL PROPERTIES.**—In gray cast iron, carbon is held in solution in the molten state, but is precipitated in flat flakes like mica when the metal freezes in the mold. These flakes form weakening planes which tend to lower strength when subjected to stress. To the casting buyer, this condition is important only when a part is subjected to stress.

**Strength Properties.**—Tensile strength of gray cast iron ranges from 20,000 to 60,000 lb. per sq.in.; 25,000 to 35,000 lb. is a fair average, unless special, high-strength cast iron is specified. Transverse strength may be made to range from 1500 to 6900 lb.on a 1.2 in. diam. bar, 18 in. between supports. Hardness varies inversely with machinability, although in direct proportion; it may be varied from 100 to 400 Brinell by increasing or decreasing different alloy elements or by hastening or retarding cooling of the iron in the mold. The ultimate compressive strength of iron ranges from 70,000 lb. to 200,000 lb. per sq. in.

**Wear Resistance.**—The early explanation that particles of free graphite form a lubricating element which prevents binding or galling when gray iron is used with similar material, or with steel, is not now generally accepted. J. W. Bolton offers a theory that the heterogeneous structure provides an interspersing of soft and hard materials, like bearing metals and bronzes. Wear resistance of gray cast iron under certain conditions is extraordinary. Examples are: automobile cylinders, subjected to sliding friction of cast-iron piston rings; slides on machine tools. For more severe wear, so-called chilled iron, resulting from rapidly cooling the surface of gray iron castings, finds wide use. Cast-iron car wheels are examples. Chilled cast iron also is used for die blocks, gears for certain services, scrapers, conveyor buckets, etc. White cast iron, low in Si, having a white fracture throughout, also resists wear extremely well, and is used for balls and liners for grinding machinery.

**Corrosion Resistance.**—Gray iron does not deteriorate rapidly under normal corrosion conditions. Special alloy cast irons are resistant to commercial acids, mine waters, and the extraordinary hazards of machinery in the chemical, tanning, fruit preserving and allied industries. See p. 3–24. Strong dense irons, low in total carbon and silicon are acid resistant. High-silicon irons, about 14% Si, resist all acids except hydrofluoric and hot concentrated hydrochloric acids. These irons are brittle and practically unmachinable, but are useful in chemical industry, notably the production of nitric acid. Gray iron has many applications in production of caustic or alkaline products, where high-nickel, austenitic types are preferred, particularly for boiling kettles.

## Table 1 —Summary of Physical Properties of Gray Cast Iron in Relation to Impact Tests

| Type of Furnace | Chemical Analysis | | | | | | | | | Transverse Tests, 1.20 in. Diam. × 18 in. Bars[7] | | | Tensile Tests, Specimen, 0.505 in. Diam. Load in 1000 lb. per sq. in. | Compression Tests, Specimen, 1.20 Diam. × 3 in. Long. Load in 1000 lb. per sq. in. | Brinell Hardness, Average, Rim to Center | Impact Tests | | | | | | Resilience × Modulus of Rupture ÷ 1000 See Note [8] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | | | | | Pendulum | | | Falling Hammer 1.20 in. Diam. Bar | | | |
| | | | | | | | | | | | | | | | | 1.20 in. Diam. × 18 in. Span | | 1.20 in. Diam. × 8 in. Span | | | | |
| | T.C. | G.C. | Si | P | Mn | S | Ni | Cr | Mo | Load, lb. | Deflection, in. | Resilience, ft-lb. | | | | Charpy, ft-lb. | Russell, ft-lb. | Russell, ft-lb. | 25 lb. Hammer, 6 in. Span, 2-in. First Drop 1/2 in. Increments | 11 lb. Hammer, 5 in. Span, 1-in. First Drop 1-in. Increments | 25 lb. Hammer, 6 in. Span, 1-in. First Drop 1-in. Increments | |
| Cupola...... | 3.49 | 2.72 | 1.68 | 0.74 | 0.54 | 0.074 | ...... | 0.01 | ..... | 2457 | 0.284 | 35.4 | 29.47 | 111.0 | 191 | 47.3 | 46.0 | 29.2 | 10.5 | 20 | 17.3 | 231 |
| Cupola...... | 3.53 | 2.69 | 1.67 | 0.68 | 0.56 | 0.076 | ...... | ..... | ..... | 2342 | .293 | 34.7 | 28.50 | 107.2 | 184 | 54.7 | 41.3 | 28.8 | 9.2 | 20 | 16.7 | 216 |
| Cupola...... | 3.95 | 3.09 | 1.90 | 0.07 | 0.65 | 0.060 | 0.01 | ..... | ..... | 1420 | .352 | 27.1 | 18.32 | 65.2 | 132 | 42.2 | 39.4 | 24.9 | 7.0 | 15.3 | 12.0 | 102 |
| Cupola...... | 3.25 | 2.66 | 2.08 | 1.99 | 0.53 | 0.050 | 0.03 | ..... | ..... | 1254 | .141 | 8.3 | 21.28 | 90.9 | 185 | 17.6 | 12.6 | 12.6 | 4.3 | 11 | 6.7 | 28 |
| Cupola...... | 3.12 | 2.44 | 2.18 | 0.63 | 0.44 | 0.096 | ...... | ..... | ..... | 2758 | .301 | 42.0 | 32.50 | 120.8 | 195 | 60.9 | 53.4 | 34.7 | 11.2 | 22 | 20.3 | 307 |
| Cupola...... | 3.61 | 2.93 | 2.06 | 0.75 | 0.52 | 0.062 | ...... | ..... | ..... | 1857 | .251 | 23.3 | 22.42 | 91.0 | 169 | 38.5 | 35.8 | 23.2 | 7.5 | 15.7 | 11.7 | 115 |
| Cupola...... | 3.49 | 2.78 | 2.08 | 0.19 | 0.55 | 0.078 | 1.17 | 0.65 | ..... | 2651 | .331 | 44.3 | 29.98 | 108.8 | 188 | 56.6 | 48.9 | 31.9 | 10.8 | 23 | 17.7 | 311 |
| Cupola...... | 3.39 | 2.75 | 2.50 | 0.43 | 0.74 | 0.062 | 0.01 | ..... | ..... | 2407 | .245 | 29.3 | 31.50 | 116.1 | 195 | 41.1 | 38.4 | 28.3 | 9.2 | 19 | 14.0 | 187 |
| Cupola...... | 3.41 | 2.85 | 2.44 | 0.63 | 0.57 | 0.070 | 0.01 | 0.02 | ..... | 1840 | .274 | 25.7 | 20.00 | 87.2 | 159 | 40.3 | 35.6 | 24.2 | 7.0 | 16 | 11.7 | 125 |
| Cupola...... | 2.61 | 1.73 | 2.38 | 0.06 | 0.77 | 0.105 | 1.08 | 0.09 | ..... | 3489 | .272 | 45.6 | 51.44 | 156.6 | 270 | 67.0 | 80.5 | 42.6 | 15.2 | 31 | 23.3 | 422 |
| Arc Elec.[1]... | 3.14 | 2.30 | 2.69 | 0.18 | 0.82 | 0.094 | 0.25 | 0.09 | 0.07 | 2590 | .293 | 38.4 | 35.65 | 115.9 | 195 | 57.6 | 53.4 | 33.4 | 10.2 | 24.7 | 15.3 | 268 |
| Arc Elec.[2]... | 3.05 | 2.20 | 2.70 | 0.16 | 0.79 | 0.089 | 0.60 | 0.24 | 0.45 | 3083 | .301 | 45.5 | 39.20 | 131.2 | 228 | 67.4 | 67.2 | 40.6 | 13.5 | 31.3 | 22.3 | 372 |
| Cupola[3]..... | 3.46 | 2.82 | 2.35 | 0.15 | 0.69 | 0.092 | 0.65 | 0.02 | 0.02 | 2200 | .335 | 39.0 | 24.45 | 90.8 | 160 | 61.7 | 56.8 | 35.1 | 9.8 | 21.3 | 16.3 | 227 |
| Arc Elec.[4]... | 2.53 | 2.03 | 3.26 | 0.02 | 1.01 | 0.035 | 10.59 | 1.55 | ..... | 2414 | .471 | 61.1 | 29.73 | 134.2 | 220 | 139.8 | 158.9 | 76.8 | 13.5 | 30.7 | 16.7 | 394 |
| Cupola...... | 3.43 | 2.75 | 2.35 | 0.20 | 0.73 | 0.092 | 0.01 | 0.01 | ..... | 2218 | .341 | 41.2 | 25.02 | 95.0 | 164 | 61.1 | 51.2 | 36.1 | 9.5 | 21 | 14.7 | 242 |
| Rocking Arc. | 2.88 | 2.25 | 1.99 | 0.43 | 0.51 | 0.098 | 0.01 | 0.01 | ..... | 3177 | .439 | 78.9 | 40.92 | 119.1 | 189 | 84.2 | 80.3 | 48.5 | 13.8 | 29.3 | 24.0 | 664 |
| Air......... | 3.24 | 2.41 | 1.63 | 0.42 | 0.56 | 0.084 | 0.06 | 0.01 | ..... | 2905 | .326 | 48.7 | 35.08 | 120.8 | 195 | 63.3 | 58.9 | 43.1 | 12.2 | 27 | 18.7 | 375 |
| Cupola...... | 3.38 | 2.59 | 1.18 | 0.14 | 0.82 | 0.077 | 0.95 | 0.01 | ..... | 2455 | .267 | 33.1 | 30.65 | 112.0 | 192 | 61.7 | 50.2 | 36.7 | 10.8 | 25 | 18.0 | 215 |
| Cupola[5]..... | 3.49 | 2.16 | 0.53 | 0.33 | 0.42 | 0.133 | ...... | 0.01 | ..... | 2835 | .294 | 45.3 | 34.95 | 123.5 | 212 | 50.7 | 48.8 | 39.8 | 12.6 | 28.3 | 16.5 | 341 |
| Arc Elec..... | 3.52 | 2.68 | 2.94 | 0.04 | 0.66 | 0.042 | 0.01 | 0.01 | ..... | 1651 | .370 | 32.5 | 19.43 | 72.8 | 150 | 42.4 | 41.8 | 28.3 | 7.8 | 17.3 | 13.7 | 143 |
| Cupola ..... | 2.50 | 1.50 | 2.20 | 0.04 | 0.74 | 0.088 | ...... | 0.08 | ..... | 3472 | .230 | 37.3 | 47.23 | 159.0 | 266 | 55.1 | 48.3 | 32.6 | 13.2 | 30.3 | 20.3 | 343 |
| Cupola...... | 3.79 | 2.99 | 1.47 | 0.12 | 0.73 | 0.083 | 0.01 | 0.01 | ..... | 1771 | .336 | 31.6 | 18.27 | 77.9 | 142 | 51.2 | 42.5 | 31.3 | 8.2 | 18 | 12.7 | 149 |
| Arc Elec.[6]... | 3.08 | 2.32 | 2.07 | 0.04 | 1.21 | 0.047 | 16.29 | 1.89 | ..... | 1970 | .745 | 94.5 | 16.70 | 86.0 | 121 | 128.7 | 93.3 | 87.8 | 12.3 | 28 | 24.0 | 493 |
| Rotary Coal. | 3.07 | 2.22 | 1.97 | 0.40 | 0.63 | 0.072 | 0.01 | ..... | ..... | 2776 | .308 | 43.2 | 29.20 | 124.5 | 210 | 56.8 | 46.9 | 29.8 | 12.0 | 23 | 18.3 | 318 |
| Arc. Elec.... | 2.79 | 1.94 | 2.44 | 0.03 | 0.50 | 0.060 | 0.50 | 0.09 | ..... | 4200 | .490 | 114.2 | 56.70 | 135 7 | 234 | 138 8 | 136.0 | 77.9 | 21.4 | 47 | 28.0 | 1276 |

[1] Mo = 0.07. [2] Mo = 0.45. [3] Not suitable for heavy section. [4] Cu = 4.07; a martensitic iron not at present (1937) commercially used. [5] Not suitable for light section. [6] Cu = 6.12; an austenitic iron for high temperature and corrosion resistance. [7] Bars cast to 1.4 in. and machined to 1.20 in. diameter. [8] Modulus of Rupture = (Transverse breaking load of 1.20 × 18 in. bar) × 26.5.

**Impact Resistance.**—The engineering conception of gray cast iron generally is a brittle material, incapable of shock or impact resistance. However, gray iron castings are used in many places where knowledge of impact resistance is important. Examples: Pistons for internal combustion motors, and car wheels, which are specified and purchased on impact tests.

A far-reaching investigation undertaken by A.S.T.M. Committee A–3 on the relation of impact properties to chemical and physical properties of cast-iron was completed and reported in 1933. See *Proc.* A.S.T.M., 1933. The developed data of the investigation are given in Table 1. Each test includes an investigation of 3 to 6 sets of specimens with varying chemical analyses and different types of foundry conditions. Beside impact values, transverse, tensile and compression tests were made and hardness values established.

In Table 1, summarized from data presented, average values are given. Brinell hardness is an average of values taken at the rim, center and midway on the cross-section of test specimens. The final columns include impact values established on unnotched specimens by the Charpy and Russell pendulum type testing apparatus, the specimen being supported rigidly near the ends and struck at the center. Three of the columns are variations of the falling hammer test on a specimen 1.20 in. diam. supported at 6 in., 5 in., and 6 in., respectively. The first of these falling hammer tests used a 25-lb. weight from an elevation of 2 in., with increasing increments of $1/2$ in. The table gives average height from which hammer dropped when the bar broke. The second of the falling hammer tests was on specimens supported at 5 in. centers with an 11-lb. weight starting at 1 in., with increasing increments of 1 in. The third summarizes average tests at 6-in. centers with 25-lb. weights starting at 1 in. with increasing increments of 1 in.

The impact test, essentially a measure of energy absorbed in breaking bars, may be expressed by the work done, namely force times distance through which force moves. Results of Charpy and Russell tests are in foot-pounds. The others may be computed from the drop and weight of hammer.

**Electrical Properties.**—Castings may have a wide range of electrical properties, depending on alloying elements. Non-magnetic cast iron is obtainable, an important factor in signal equipment or electrical control machinery. Non-magnetic iron contains a relatively large amount of Ni and Mn. The electrical resistivity of pure iron according to Dr. T. D. Yensen is 9.87 micro-ohms per cu.cm. at 20° C. Interruptions in resistivity are provided by other elements, especially carbon. The resistivity of soft gray iron is 74.62 micro-ohms, and of hard iron 98.04 micro-ohms per cu.cm. (J. E. Hurst, Metallurgy of Cast Iron, p. 270.)

According to Hatfield (Cast Iron in the Light of Recent Research) magnetic properties of cast iron may vary between wide limits ranging from low permeability and high coercive force, for permanent magnets, to high permeability, low coercive force and low hysteresis loss for electrical machinery.

**Thermal Properties.**—Specific heat of gray iron has not been determined accurately for all temperatures, but the International Critical Tables give it as 0.13 between 0 and 100° C. Hurst supplies accumulated data to show that it varies in pure iron from 0.1111 at 100° C. to 0.1948 at 1600° C. The addition of various elements appears to raise the specific heat.

Thermal conductivity according to Donaldson, *Proc.* I. M. E., 1928, varies from 0.110 to 0.137 calories per cm. per sec., and decreases with temperature increase. Values obtained depend on composition and structure of gray iron. Si lowers conductivity as does Ni, Mn and V. Chromium and tungsten tend to raise conductivity.

**Growth.**—Gray cast iron expands when heated, and unless certain limits are exceeded, returns to its original size when cooled. Growth is the permanent increase in volume that occurs in some grades of gray iron under repeated heating and cooling. The critical point which influences growth is about 1200° F., but in the presence of steam or corrosive elements, the change occurs at lower temperature. Different irons show varying susceptibility to this change, which is accompanied by loss in strength. Rugan and Carpenter (report to British Iron & Steel Inst.) showed that growth is proportionate to Si content in the range from 1.0 to 3.5% Si; that P, S, and Mn all are elements that retard this action. Piwowarsky (*Trans.* Am. Fdymen's Assoc., 1926) shows that superheating molten iron tends to reduce growth in resulting castings. This is ascribed by Bolton to extremely fine graphite formation. Hurst (*Fdry. Trade Jour.*, Aug. 12, 1926) states that growth tendency is inversely proportional to maximum strength, and directly proportional to modulus of elasticity.

**Fatigue Resistance.**—Fatigue tests conducted at the Universities of Illinois and Wisconsin (J. B. Kommers, *Proc.* A.S.T.M., vol. xxix, Pt. 2, p. 100, 1929) included tension, compression, transverse, Russell impact, and rotating beam fatigue. These show ratio of endurance limit to modulus of rupture to average 0.26. See also p. 7–12.

**EFFECTS OF ALLOYING ELEMENTS.**—In making calculations of the common elements in gray cast iron, the Am. Fdymen's Assoc. suggests:

**Silicon** softens iron, opens the grain, lowers density; in excess, it weakens and hardens iron and causes sponginess in heavy sections.

Table 2.—Minimum Breaking Loads for Transverse Tests of Cast-iron Test Bars

| Test Bar | | Class No. | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Diam., in. | Distance between supports, in. | 20 | 25 | 30 | 35 | 40 | 50 | 60 |
| | | Minimum Breaking Load at Center, lb. | | | | | | |
| 0.875 | 12 | 900 | 1025 | 1150 | 1275 | 1400 | 1,675 | 1925 |
| 1.2 | 18 | 1800 | 2000 | 2200 | 2400 | 2600 | 3,000 | 3400 |
| 2.0 | 24 | 6000 | 6800 | 7600 | 8300 | 9100 | 10,300 | ....... |

**Sulphur** hardens iron, closes the grain, increases density; in excess, it causes defects and may weaken the metal. It increases tendency to chill white in thin sections.

**Phosphorus** increases fluidity of iron; in excess, it lowers impact value. It fosters formation of porosity in heavy sections.

**Manganese** increases density of iron; it has strengthening effect up to around 1.00 Mn; in very small amounts it has a softening effect. Over 1.00, Mn has a tendency to harden iron.

**Combined Carbon** (CC) increases hardness and toughness of iron; it increases strength below 0.80 C, decreases strength when in excess, and closes the grain.

**Graphitic Carbon** (GC) increases softness and machinability in iron, weakens it, opens the grain, and decreases density.

## 2. SPECIFICATIONS FOR GRAY IRON CASTINGS

**CLASSIFICATION.**—A.S.T.M. tentative specification A48–36 covers general gray iron castings not covered by other existing A.S.T.M. specifications, and applies to castings where strength is a consideration. Its purpose is to list and classify cast irons and gray iron castings according to minimum strengths of tensile test bars as follows:

| Class No. | 20 | 25 | 30 | 35 | 40 | 50 | 60 |
|---|---|---|---|---|---|---|---|
| Min. Tensile Str., lb. per sq. in. ........ | 20,000 | 25,000 | 30,000 | 35,000 | 40,000 | 50,000 | 60,000 |

Transverse tests are optional and minimum breaking loads are given in Table 2. By agreement between manufacturer and purchaser deflection values in transverse test may be specified. Transverse tests are made on bars as cast, with central loading between supports. To correct test result to standard diameter, breaking load obtained in testing the bar is divided by correction factor, given in Table 3.

Controlling dimension is diameter of bar at fracture. Rate of application of load should

Table 3.—Correction Factors for Transverse Cast-iron Test Bars

| 0.875 in. Diameter Bar | | | 1.20 in. Diameter Bar | | | 2.00 in. Diameter Bar | | |
|---|---|---|---|---|---|---|---|---|
| Diameter of Test Bars, in. | Correction Factor | | Diameter of Test Bars, in. | Correction Factor | | Diameter of Test Bars, in. | Correction Factor | |
| | Load | Deflection | | Load | Deflection | | Load | Deflection |
| 0.825 | 0.838 | 1.061 | 1.10 | 0.770 | 1.091 | 1.90 | 0.857 | 1.053 |
| .830 | .853 | 1.054 | 1.11 | .791 | 1.081 | 1.91 | .871 | 1.047 |
| .835 | .869 | 1.048 | 1.12 | .813 | 1.071 | 1.92 | .885 | 1.042 |
| .840 | .885 | 1.042 | 1.13 | .835 | 1.062 | 1.93 | .899 | 1.037 |
| .845 | .900 | 1.036 | 1.14 | .857 | 1.053 | 1.94 | .913 | 1.032 |
| .850 | .916 | 1.029 | 1.15 | .880 | 1.043 | 1.95 | .927 | 1.026 |
| .855 | .933 | 1.023 | 1.16 | .903 | 1.034 | 1.96 | .941 | 1.021 |
| .860 | .949 | 1.017 | 1.17 | .927 | 1.026 | 1.97 | .955 | 1.015 |
| .865 | .966 | 1.012 | 1.18 | .951 | 1.017 | 1.98 | .970 | 1.010 |
| .870 | .983 | 1.006 | 1.19 | .975 | 1.009 | 1.99 | .985 | 1.005 |
| .875 | 1.000 | 1.000 | 1.20 | 1.000 | 1.000 | 2.00 | 1.000 | 1.000 |
| .880 | 1.017 | .994 | 1.21 | 1.025 | .992 | 2.01 | 1.015 | .995 |
| .885 | 1.034 | .989 | 1.22 | 1.051 | .984 | 2.02 | 1.030 | .990 |
| .890 | 1.051 | .983 | 1.23 | 1.077 | .976 | 2.03 | 1.046 | .985 |
| .895 | 1.069 | .978 | 1.24 | 1.103 | .968 | 2.04 | 1.061 | .980 |
| .900 | 1.087 | .972 | 1.25 | 1.130 | .960 | 2.05 | 1.076 | .976 |
| .905 | 1.106 | .967 | 1.26 | 1.158 | .952 | 2.06 | 1.092 | .972 |
| .910 | 1.125 | .962 | 1.27 | 1.185 | .945 | 2.07 | 1.109 | .967 |
| .915 | 1.143 | .956 | 1.28 | 1.214 | .938 | 2.08 | 1.125 | .962 |
| .920 | 1.162 | .951 | 1.29 | 1.242 | .930 | 2.09 | 1.141 | .957 |
| .925 | 1.181 | .946 | 1.30 | 1.271 | .923 | 2.10 | 1.158 | .952 |

**Table 4.—Dimensions of Test Bar and Controlling Section of Casting for Transverse Tests**

| Controlling Section of Casting | Test Bar | Dimensions of Test Bar, in. | | Distance between Supports in Transverse Test |
|---|---|---|---|---|
| | | Diameter | Length | |
| 0.50 in. max. | A | 0.875 | 15 | 12 in. |
| 0.51 to 1.00 in. | B | 1.20 | 21 | 18 in. |
| 1.01 to 2.00 in.* | C | 2.00 | 27 | 24 in. |

* For sections over 2.00 in., test bar C or a larger bar may be used by mutual agreement.

produce fracture in not less than 15 seconds for the 0.875-in. bar; 20 sec. for the 1.2-in. bar; 40 sec. for the 2.0-in. bar.

Tension and transverse test bars are cast separately from the casting; 3 tension, 3 optional tension, and 3 transverse test bars are required. Size of test bars depends on controlling section of the casting, and the corresponding transverse test bar is used, as shown in Table 4. Form of tension test bars is shown in Fig. 1, and dimensions in Table 5. Instead of using the separately cast tension test bar, it is optional to use test specimens machined from the transverse test bars. Form and dimensions of the optional tensile test bars then shall give a minimum section in the breaking area as follows:

| | Test Bar A | Test Bar B | Test Bar C |
|---|---|---|---|
| Diameter | 0.505 in. | 0.800 in. | 1.25 in. |
| Length of reduced area | Shall be equal to or greater than diameter | | |

For sections of castings over 2 in., test bar C or a larger test bar may be used by agreement between manufacturer and purchaser.

The previous correlation is only approximate and may need modification for complicated castings. Tension test specimens must so fit holders of testing machine that the load is axial; self-aligning shackles are recommended.

There is no definite factor for correlation between tensile and transverse strengths for various types of gray cast iron. Table 3 shows minimum values only. If manufacturer and purchaser agree on transverse tests to govern, the proper range of transverse values may be taken from the table or determined by pouring transverse and tension bars to establish a correlation for that particular cast iron. Test bars should be made under the same sand conditions as the casting, poured from the same ladles of iron as the castings, and as nearly as possible at the same temperature. They should receive the same mechanical and thermal treatment as the castings, and in heat treatment, should be treated adjacent to the castings they represent.

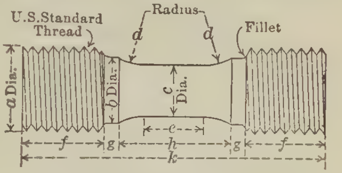

Fig. 1. Standard Test Specimen

**GRAY IRON CASTING COMPOSITION.**—The compositions of castings given in Table 6 are not absolute, but are a guide for general practice. Designers and purchasers of castings should rely on specifications of desired physical properties and should not attempt to designate chemical analyses. Questions of technique and skill permit competent foundrymen to produce a satisfactory product which may vary from these empirical compositions, and individual practice may offer a superior product which does not accord with the compositions recommended in Table 6.

**Shrinkage of Castings**—The allowance necessary for shrinkage varies for different kinds of metal, and the different conditions under which they are cast. For castings

(*Continued on p.* 20–10)

**Table 5.—Dimensions of Cast-iron Tensile Test Bars**

See Fig. 1.

| Dimension | Test Bar A | Test Bar B | Test Bar C |
|---|---|---|---|
| *a* | 3/4 in. | 1 1/8 in. | 1 7/8 in. |
| *b* | 5/8 in. | 15/16 in. | 1 7/16 in. |
| *c* | 0.505 in. | 0.800 in. | 1.25 in. |
| *d* | 1.00 in. | 1.00 in. | 2.00 in. |
| *e** | Shall be equal or greater than *c* (diameter) | | |
| *f* | 1.00 in. | 1.00 in. | 1 3/4 in. |
| *g* | 0.25 in. | 0.25 in. | 5/16 in. |
| *h*, minimum | 1.25 in. | 1.50 in. | 2 1/4 in. |
| *k*, minimum | 3.75 in. | 4.00 in. | 6 3/8 in. |

* Length of parallel.

### Table 6.—Recommended Analyses for Gray Iron Castings

| Type of Casting | Silicon, Percent | Sulphur, Percent | Phosphorus, Percent | Manganese, Percent | Total Carbon, Percent |
|---|---|---|---|---|---|
| Acid-resistant | 1 to 2 | Max. 0.05 | Max. 0.40 | 1 to 1.5 | Low |
| Acid stills and eggs | 1 to 1.5 | Max. 0.05 | Max. 0.40 | 1 to 1.25 | Low |
| Agricultural implement (medium) | 2.10 | 0.10 | 0.50 | 0.60 | 3.40 |
| Agricultural implement (light) | 2.20 | 0.10 | 0.70 | 0.55 | 3.50 |
| Annealing boxes for malleable work | 0.50 to 0.80 | Max. 0.10 | 0.10 to 0.20 | 0.10 to 0.30 | 2.60 to 2.00 |
| Annealing boxes, pots and pans | 0.60 to 1.20 | Max. 0.06 | Max. 0.35 | 0.35 to 0.60 | 3.00 to 3.30 |
| Anvil blocks | 0.80 | ..... | 0.30 | 0.70 | 3.20 |
| Automobile castings (soft) | 2.60 | 0.10 | 0.30 | 0.65 | 3.40 |
| Ball mill liners | 1.00 | ..... | Max. 0.35 | 0.70 | 3.00 |
| Balls for ball mills | 0.90 to 1.20 | Max. 0.08 | 0.20 | 0.60 to 1.0 | 3.20 |
| Bed plates | 1.25 to 2.20 | Max. 0.10 | 0.30 to 0.60 | 0.60 to 0.80 | 3.30 to 3.50 |
| Bedstead work | 2.60 to 2.80 | Max. 0.09 | 0.90 to 1.10 | 0.30 to 0.60 | ..... |
| Bells and hoppers (blast furnace) | 1.25 to 1.50 | Max. 0.08 | Under 0.30 | 0.80 to 1.0 | Low |
| Binders | 2 to 2.50 | 0.06 to 0.08 | 0.60 to 0.80 | 0.60 to 0.80 | ..... |
| Blast furnace castings | 1.25 to 1.75 | Max. 0.09 | 0.30 to 0.60 | 0.50 to 0.80 | ..... |
| Boiler fronts | 2.25 to 2.75 | Max. 0.09 | 0.50 to 0.70 | 0.40 to 0.60 | 3.40 |
| Boiler sections | 2 to 2.50 | Max. 0.08 | 0.15 to 0.30 | 0.60 to 1.0 | 3.40 |
| Brake drums (auto) [1] | 1.90 | 0.08 | 0.15 | 0.65 | 3.30 |
| Brake shoes | 1.40 to 1.60 | 0.08 to 0.10 | Max. 0.50 | 0.50 to 0.70 | Low |
| Car castings, gray iron | 1.50 to 2.25 | Max. 0.08 | 0.40 to 0.60 | 0.60 to 0.80 | ..... |
| Car wheels, chilled | 0.65 | 0.12 | 0.35 | 0.60 | 3.35<br>(0.90 C.C.) |
| Caustic pots | 1.00 | 0.07 | 0.20 | 0.75 | 3.60 |
| Chemical castings | 1 to 2 | Max. 0.06 | Max. 0.25 | 0.90 to 1.50 | 2.75 to 3.40 |
| Chilled castings | 0.75 to 1.00 | 0.08 to 0.10 | 0.20 to 0.40 | 0.70 to 0.80 | 3.30 to 3.40 |
| Chills | 1.75 to 2.25 | Max. 0.07 | 0.10 to 0.20 | 0.60 to 0.90 | ..... |
| Collars and couplings (shafting) | 1.75 to 2 | Max. 0.08 | 0.40 to 0.60 | 0.50 to 0.70 | 3.40 |
| Cone pulleys | 2 to 2.25 | Max. 0.08 | 0.50 to 0.80 | 0.50 to 0.70 | ..... |
| Cotton machinery | 2 to 2.25 | Max. 0.08 | 0.60 to 0.80 | 0.50 to 0.80 | 3.50 |
| Crusher castings | 1 to 2 | Max. 0.09 | Max. 0.40 | 0.60 to 0.80 | Low |
| Crusher jaws | 0.80 to 1 | 0.80 to 0.10 | 0.20 to 0.35 | 0.80 to 1.20 | 3.00 to 3.20 |
| Cutting tools, chilled iron | 1 to 1.25 | Max. 0.08 | 0.20 to 0.30 | 0.60 to 0.80 | 3.30 |
| Cylinders | 1 to 2 | Max. 0.10 | 0.30 to 0.60 | 0.60 to 1.0 | 3.30 |
| " air | 1.25 | 0.10 | 0.20 | 0.65 | 3.25 |
| " air-cooled, airplane | 2.25 | Max. 0.08 | Max. 0.22 | 0.80 | 3.30 |
| " ammonia | 1.25 | 0.10 | 0.20 | 0.65 | 3.25 |
| " automobile [2] | 2.25 | 0.10 | 0.15 | 0.65 | 3.25 |
| " automobile, high nickel [3] | 1.80 | 0.10 | 0.15 | 0.70 | 3.25 |
| " Diesel | 0.80 | Max. 0.08 | Max. 0.22 | 1.00 | 3.00 |
| " gas engine (light) | 1.75 | 0.09 | 0.35 | 0.80 | 3.40 |
| " " " (medium) | 1.50 | 0.09 | 0.35 | 0.80 | 3.40 |
| " " " (heavy) | 1.25 | 0.09 | 0.35 | 0.80 | 3.40 |
| " hydraulic (heavy) | 0.80 to 1.20 | 0.06 to 0.10 | Max. 0.22 | 0.80 to 1 | 3.20 |
| " " (medium) | 1.20 | 0.06 to 1.0 | Max. 0.22 | 0.80 | 3.30 |
| " locomotive | 1.25 | Max. 0.08 | Max. 0.30 | 0.80 | 3.30 |
| " marine | 1.20 to 1.80 | Max. 0.08 | Max. 0.22 | 0.80 | 3.20 to 3.40 |
| " steam (heavy) | 1.0 to 1.25 | Max. 0.10 | 0.20 to 0.40 | 0.80 to 1.0 | Low |
| " " (medium) | 1.25 to 1.75 | Max. 0.09 | 0.30 to 0.50 | 0.70 to 0.90 | ..... |
| Cylinder bushings, locomotive | 1.25 to 1.50 | Max. 0.08 | 0.30 to 0.50 | 0.70 to 0.90 | ..... |
| Diamond polishing wheels | 2.50 to 3 | Max. 0.07 | 0.20 to 0.40 | 0.30 to 0.50 | 2.97 |
| Dies for drop hammers | 1.00 to 1.25 | Max. 0.07 | Max. 0.30 | 0.60 to 0.80 | 3.20 |
| Dynamo frames, bases and spiders, large and medium | 1.40 to 1.80 | Max. 0.08 | 0.50 to 0.80 | 0.40 to 0.60 | 3.20 to 3.30<br>(C.C. 1.60) |
| Dynamo frames, bases and spiders, small | 2.20 | Max. 0.08 | Max. 0.70 | 0.70 | Max. 3.40 |
| Eccentric straps | 1.75 to 2 | Max. 0.09 | 0.40 to 0.60 | 0.60 to 0.80 | ..... |
| Electrical castings | 2 to 2.60 | Max. 0.08 | 0.50 to 0.80 | 0.40 to 0.60 | 3.30 to 3.40 |
| Embossing heads | 1.25 to 1.50 | Max. 0.09 | 0.30 to 0.50 | 0.70 to 1.0 | ..... |
| Engine frames | 1.70 to 2 | Max. 0.08 | Max. 0.60 | 0.70 | 3.30 to 3.40 |
| Fan and blower casings | 2.20 | Max. 0.08 | 0.60 to 0.80 | 0.60 | Max. 3.35 |
| Ferrocyanide pots | 3 to 4 | Max. 0.08 | 0.40 to 0.80 | 0.40 to 0.80 | ..... |
| Fire bars | 1.40 | ..... | Max. 0.40 | 0.70 | 3.20 |
| Fire pots (light) | 2.25 | 0.06 | Max. 0.20 | 0.80 | 3.40 |
| " " (heavy) | 1.80 | 0.06 | Max. 0.20 | 0.80 | 3.30 |
| Fly-wheels (auto) | 2.25 to 2.50 | Max. 0.07 | 0.40 to 0.50 | 0.50 to 0.70 | ..... |
| " (machinery) | 1.20 to 2.00 | Max. 0.08 | Max. 0.80 | 0.70 | 3.20 to 3.40 |
| Friction clutches | 1.60 to 2 | 0.08 | Max. 0.22 | 0.50 to 0.70 | Max. 3.50 |
| Gears (heavy | 1.20 | 0.08 to 0.10 | Max. 0.40 | 0.90 | 3.20 |
| " (medium) | 1.70 | Max. 0.09 | Max. 0.50 | 0.80 | 3.30 |
| " (light) | 2.00 | Max. 0.08 | Max. 0.60 | 0.70 | 3.40 |
| Grate bars | 1.70 | Max. 0.06 | Max. 0.30 | 0.80 | 3.30 |
| Grinding burrs | 0.60 to 0.90 | Max. 0.12 | 0.20 to 0.40 | 0.40 to 0.60 | 3.00 |

[1] Ladle additions of Ni, 1.25, Cr, 0.30, are used.

[2] For auto cylinders general practice is to run a Brinell hardness test in the bore; a desired hardness is 180 Brinell. Emphasis is further directed to obtaining a machinable iron (no hard corners or spots). This analysis with Ni, 0.60, Cr, 0.25, Brinell 190, is a low-nickel mixture.

[3] An addition of 1.25% nickel is added to the ladle. Brinell 190.

**Table 6.—Recommended Analyses for Gray Iron Castings—*Continued***

| Type of Casting | Silicon, Percent | Sulphur, Percent | Phosphorus, Percent | Manganese, Percent | Total Carbon, Percent |
|---|---|---|---|---|---|
| Gun Carriages | 1 to 1.25 | Max. 0.06 | 0.20 to 0.30 | 0.80 to 1.0 | ..... |
| Hangers (shafting) | 1.50 to 2 | Max. 0.09 | 0.40 to 0.60 | 0.60 to 0.80 | ..... |
| Hardening pots | 0.60 to 1 | Max. 0.06 | Max. 0.20 | 0.40 to 0.60 | ..... |
| Hardware, light | 2.25 to 2.75 | Max. 0.08 | Max. 0.80 | 0.40 to 0.60 | 3.50 |
| Heat-resistant iron | 1.15 | 0.07 | 0.10 | 0.80 | 3.50 |
| High-test iron | 2.25 | 0.09 | 0.10 | 0.80 | 2.75 |
| " " " (nickel)[4] | 1.25 | 0.09 | 0.10 | 0.85 | 3.10 |
| Hollow ware | 2.25 to 2.75 | Max. 0.08 | 0.60 to 0.90 | 0.40 to 0.60 | 3.45 |
| Housings (rolling mill) | 1.0 to 1.25 | Max. 0.10 | Max. 0.30 | 0.70 | 3.20 |
| Ingot molds and stools | 1.00 | 0.07 | 0.20 | 0.90 | 3.50 |
| Locks and hinges | 2.50 to 2.75 | Max. 0.08 | 0.70 to 1.0 | 0.40 to 0.60 | ..... |
| Locomotive castings, heavy | 1.25 to 1.50 | Max. 0.10 | 0.30 to 0.50 | 0.70 to 0.90 | ..... |
| " " light | 1.50 to 2 | Max. 0.09 | 0.40 to 0.60 | 0.60 to 0.80 | ..... |
| Machinery castings, heavy (heavy service) | 1.25 | 0.10 | 0.50 | 0.50 | 3.25 |
| Machinery castings, heavy (medium service) | 1.75 | 0.10 | 0.50 | 0.50 | 3.25 |
| Machinery castings, light | 2.25 | 0.09 | 0.50 | 0.50 | 3.25 |
| Magneto and armature parts | 2.75 | Max. 0.08 | Max. 0.70 | 0.50 | 3.55 |
| Mine-car wheels | 0.75 to 1.25 | Max. 0.10 | 0.30 to 0.50 | 0.40 to 0.60 | ..... |
| Motor frames, bases and spiders, large | 2 to 2.50 | Max. 0.08 | 0.50 to 0.80 | 0.40 to 0.60 | Low |
| Motor frames, bases and spiders, small | 2.50 to 3 | Max. 0.08 | 0.50 to 0.80 | 0.30 to 0.40 | Low |
| Mowers | 2 to 2.50 | Max. 0.09 | 0.60 to 0.80 | 0.50 to 0.80 | ..... |
| Municipal | 2.00 | ..... | Max. 0.40 | 0.70 | 3.30 |
| Niter pots | 1 to 1.50 | Max. 0.06 | Max. 0.20 | 1.0 to 1.50 | Low |
| Novelty work | 2.50 to 3 | Max. 0.08 | 0.80 to 1 | 0.40 to 0.60 | ..... |
| Ornamental | 2.25 to 2.75 | Max. 0.06 | Max. 0.90 | 0.60 | 3.50 |
| Permanent molds | 1.70 | 0.08 | Max. 0.20 | 1.25 | 3.50 |
| Permanent mold castings | 1.50 to 3 | Max. 0.06 | 0.30 to 0.80 | Max. 0.40 | ..... |
| Piano plates | 2 to 2.25 | Max. 0.08 | 0.40 to 0.60 | 0.60 to 0.80 | 3.40 |
| Pig molds | 2.00 | 0.07 | Max. 0.30 | 0.90 | 3.40 |
| Pillow blocks | 1.50 to 1.75 | Max. 0.08 | 0.40 to 0.60 | 0.60 to 0.80 | 3.50 |
| Pipe (light and medium) | 1.75 | 0.08 | 0.80 | 0.50 | 3.60 |
| " (heavy) | 1.40 | 0.08 | 0.80 | 0.50 | 3.40 |
| Pipe fittings | 1.75 to 2.50 | Max. 0.08 | 0.50 to 0.80 | 0.60 to 0.80 | ..... |
| " " (superheated steam) | 1.50 to 1.75 | Max. 0.08 | 0.20 to 0.40 | 0.70 to 0.90 | Low |
| Pistons (plain iron) | 2.25 | 0.10 | 0.15 | 0.65 | 3.35 |
| " (Mo, 0.35%) | 2.25 | 0.08 | 0.15 | 0.65 | 3.35 |
| Piston rings (individual) | 2.90 | 0.06 | 0.50 | 0.65 | 3.50 |
| Plow mold boards | 0.85 | 0.10 | 0.40 | 0.50 | 3.75 |
| Plow shares, chilled | 1.25 | 0.10 | 0.40 | 0.55 | 3.60 |
| Plugs for piercing billets | 1 to 1.25 | Max. 0.08 | 0.20 to 0.30 | 0.40 to 0.60 | ..... |
| Pressure castings | 1.25 | 0.10 | 0.20 | 0.65 | 3.25 |
| Printing press | 1.75 to 2.25 | Max. 0.09 | 0.50 to 0.70 | 0.50 to 0.70 | ..... |
| Projectiles and shell[5] | 1.00 to 1.40 | Max. 0.08 | Max. 0.30 | 0.70 | 3.10 to 3.35 |
| Propeller wheels | 1 to 1.75 | Max. 0.10 | 0.20 to 0.40 | 0.60 to 1.0 | ..... |
| Pulleys, medium | 1.70 | Max. 0.08 | Max. 0.70 | 0.70 | 3.40 |
| " small | 2.30 | Max. 0.08 | Max. 0.70 | 0.70 | 3.50 |
| Pumps, hand | 2 to 2.25 | Max. 0.08 | 0.60 to 0.80 | 0.60 | 3.40 |
| Radiators | 2.20 | Max. 0.08 | Max. 0.80 | 0.60 | 3.45 |
| Railroad castings | 1.75 | Max. 0.08 | Max. 0.30 | 0.70 | Max. 3.45 |
| Refrigerator | 2.20 | Max. 0.08 | Max. 0.22 | 0.70 | Max. 3.40 |
| Retorts | 1.75 to 2 | Max. 0.06 | Max. 0.20 | 0.60 to 1 | Low |
| Rolls, chilled | 0.60 to 0.90 | 0.06 to 0.08 | 0.20 to 0.40 | 0.55 to 0.80 | 3.20 to 3.40 |
| Scale, castings | 2.00 | Max. 0.08 | Max. 0.90 | 0.80 | 3.40 |
| School desks and furniture | 2.60 | ..... | Max. 0.90 | 0.60 | 3.40 |
| Slag car bowls | 1.50 to 1.75 | Max. 0.07 | Max. 0.20 | 0.60 to 1.0 | Low |
| Smoke stacks (locomotive) | 1.75 to 2 | Max. 0.10 | 0.30 to 0.60 | 0.60 to 0.80 | ..... |
| Soil pipe and fittings | 1.40 | 0.08 | 0.80 | 0.50 | 3.40 |
| Stove plate | 2.60 | Max. 0.08 | 0.60 to 0.90 | 0.60 | 3.50 |
| Textile machinery | 2.70 | Max. 0.08 | Max. 0.80 | 0.70 | Max. 3.50 |
| Toys | 2.50 to 3 | Max. 0.08 | 0.80 to 1.0 | 0.40 to 0.60 | ..... |
| Typewriter castings | 2.50 to 2.75 | Max. 0.08 | 0.60 to 0.80 | 0.50 to 0.60 | ..... |
| Valves, large | 1.20 | Max. 0.08 | 0.22 | 0.90 | 3.00 |
| " small | 2.20 | Max. 0.08 | 0.30 | 0.80 | 3.30 |
| Water heaters | 2 to 2.25 | Max. 0.08 | 0.30 to 0.50 | 0.60 | 3.40 |
| Wearing plates | 1.10 | 0.17 | 0.35 | 0.85 | 3.20 |
| Weaving machinery | 2.30 | Max. 0.08 | Max. 0.80 | 0.60 | 3.50 |
| Welding rod | 2.90 | Max. 0.08 | 0.60 | 0.60 | 3.50 |
| Wheels, large | 1.50 to 2 | Max. 0.09 | 0.30 to 0.50 | 0.60 to 0.80 | ..... |
| " small | 1.75 to 2.25 | Max. 0.08 | 0.40 to 0.60 | 0.50 to 0.70 | ..... |
| Wheel centers | 1.25 to 1.50 | Max. 0.10 | 0.30 to 0.50 | 0.70 to 0.90 | ..... |
| White iron castings | 0.80 | 0.08 to 0.10 | Max. 0.40 | 0.40 | 3.00 |
| Woodworking machinery | 1.70 | Max. 0.08 | Max. 0.60 | 0.60 | 3.20 |

[4] Ladle addition of Ni, 1.00%.

[5] Light 3- to 5-in. diam., higher Si and C; heavy, 10 to 12 in. diam., lower Si and C.

where the thickness runs about 1 in., cast under ordinary conditions, the following allowance can be made:

| | | | |
|---|---|---|---|
| For cast iron, | 1/8 inch per foot. | For zinc, | 5/16 inch per foot. |
| " brass, | 3/16 " " " | " tin, | 1/12 " " " |
| " steel, | 1/4 " " " | " aluminum, | 3/16 " " " |
| " malleable iron, | 1/8 " " " | " britannia, | 1/32 " " " |

Thicker castings, under the same conditions, will shrink less, and thinner ones more, than this standard. The quality of the material and the manner of molding and of cooling the casting also will make a difference. W. J. Keep (*Trans.* A. S. M. E., xvi, p. 561, 1895) gives the following "approximate key for regulating foundry mixtures" so as to produce a shrinkage of 1/8 inch per foot in castings of different cross-sectional area:

| Size of casting, in. sq. | 1/2 | 1 | 2 | 3 | 4 |
|---|---|---|---|---|---|
| Silicon required, percent | 3.25 | 2.75 | 2.25 | 1.75 | 1.25 |
| Shrinkage of a 1/2-in. test-bar, in. per ft. | 0.125 | 0.135 | 0.145 | 0.155 | 0.165 |

**Table 7.—To Determine Weight of Casting from Weight of Pattern**

(Allowance must be made for the metal in the pattern)

| A Pattern Weighing 1 lb. When Made of | Will Weigh When Cast in | | | | | |
|---|---|---|---|---|---|---|
| | Cast Iron | Yellow Brass | Gun Metal | Zinc | Copper | Aluminum |
| Baywood | 8.8 lb. | 9.9 lb. | 10.3 lb. | 8.5 lb. | 10.5 lb. | 3.2 lb. |
| Beech | 8.5 | 9.5 | 10. | 8.2 | 10.1 | 3.1 |
| Cedar | 16.1 | 18. | 18.9 | 15.6 | 19.2 | 5.8 |
| Cherry | 10.7 | 12. | 12.6 | 10.4 | 12.8 | 3.9 |
| Linden | 12. | 13.5 | 14.1 | 11.6 | 14.3 | 4.3 |
| Mahogany | 8.5 | 9.5 | 10. | 8.2 | 10.1 | 3.1 |
| Maple | 9.2 | 10.3 | 10.8 | 8.9 | 11. | 3.2 |
| Oak | 9.4 | 10.5 | 11. | 9.1 | 11.2 | 3.4 |
| Pear | 10.9 | 12.2 | 12.8 | 10.6 | 13. | 3.9 |
| Pine, white | 14.7 | 16.5 | 17.3 | 14.3 | 17.5 | 5.3 |
| Pine, yellow | 13.1 | 14.7 | 15.4 | 12.7 | 15.6 | 4.7 |
| Whitewood | 16.4 | 18.4 | 19.3 | 15.9 | 19.5 | 5.9 |

## 3. FOUNDRY MELTING PRACTICE

### Cupola Practice

The American cupola furnace is a vertical, cylindrical stack of riveted mild-steel plate, elevated on columns above the floor level, with semi-circular drop doors at the bottom. The stack has small openings at front and back, close to the bottom, and a suitable opening above at a predetermined level for charging metal and fuel. It is surrounded by an enclosed, steel-plate wind belt, continuous about the lower portion of the stack, connected with a blower and communicating with the interior of the cupola through a series of openings or tuyeres. An inner lining of refractory brick or blocks is built from top to bottom inside the stack. The lining is 4 1/2 to 10 1/2 in. thick, depending on overall diameter of the cupola.

In operation the bottom doors are closed and a molding sand bottom 3 to 10 in. deep, is sloped properly to direct molten iron to the tap hole. Special refractory shapes close the breast or front opening, and a special shape inserted at the back of the cupola at a higher elevation drains slag. Air is delivered at pressures of 10 to 25 oz. per sq. in. by blower or fan to the wind belt and enters the cupola through the tuyeres.

Special types of cupolas admit blast in a variety of ways. The simpler forms, as in the standard cupola of Whiting Corp., Harvey, Ill., have one or two rows of tuyeres. The bottom row is rectangular, expanding inward to the inside of the cupola, the discharge area being approximately double the inlet area. The upper row, of similar construction, supplies air to consume gases generated by the coke at the lower level. Appropriate dampers or blast pipe gates govern the amount of entering air, or blowers can be regulated to give correct pressure and velocity of air. Table 8 shows a simple calculation of tuyere areas based on inner lining of the cupola.

At a melting ratio of 8 lb. iron to 1 lb. coke, 30,000 cu. ft. of air usually is assumed to be required for melting 1 ton of iron. This ratio is not arbitrary, and a 10 : 1 melting ratio is common. Table 9 summarizes the actual performance of a number of cupolas, based on studies by Dr. E. E. Marbaker.

### Table 8.—Calculation of Tuyere Ratios

| Effective Diam. Inside Lining, in. | Area Inside Lining, sq. in. | No. of Tuyeres | | Dimensions of Primary Tuyeres, in. | | Total Area of Primary Tuyeres, sq. in. | | Total Tuyere Area, sq. in. | | Ratio of Total Tuyere Area to Area Inside Lining | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Primary | Secondary | At Shell | At Lining | At Shell | At Lining | At Shell | At Lining | At Shell | At Lining |
| 43 | 1452 | 6 | 6 | 10×12 | 5×18 | 720 | 540 | 810 | 630 | 1 : 1.8 | 1 : 2.3 |
| 62 | 3019 | 8 | ...... | 7×21 | 7×21 | 1176 | 1176 | 1176 | 1176 | 1 : 2.6 | 1 : 2.6 |
| 72 | 4072 | 8 | ...... | 9×6 | 9×9 | 432 | 648 | 432 | 648 | 1 : 9.5 | 1 : 6.3 |
| 69 | 3729 | 15 | ...... | 7 (diam.) | 6×10 | 578 | 900 | 578 | 900 | 1 : 6.5 | 1 : 4.2 |
| 63 | 3117 | 8 | ...... | 9 1/2×14 | 9 1/2×14 | 1064 | 1064 | 1064 | 1064 | 1 : 2.9 | 1 : 2.9 |

### Table 9.—Examples of Cupola Practice

| Example No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Size of cupola, No. | 8 | 9 1/2 | 12 | 10 | 9 1/2 |
| Diameter, inside lining, in. | 60 | 72 | 72 | 72 | 70 |
| Bosh diameter, in. | 40–46–30 | 62 | ......... | 66 | 63 |
| Bosh height, in. | ......... | 30 | ......... | 36 | ......... |
| Normal melting rate (10 : 1 melting ratio), tons per hr. | 6–7 | 14–18 | 18–21 | 19–20 | 14–18 |
| Total iron charged, tons | 107 | 220 | 160 | 150 | 139.5 |
| Time of melting, hr. | 9 3/4 | 11 | 10 | 8 | 9 |
| Melting rate, tons per hr. | 11 | 20 | 16 | 18.75 | 17.5 |
| Bed coke, lb. | 1,500 | 4,000 | 4,500 | 4,500 | 4,200 |
| First coke charge, lb. | 510 | ......... | ......... | ......... | ......... |
| Intermediate coke charge, lb. | 13,125 | 49,050 | 32,000 | 20,550 | 39,525 |
| Total coke charged, lb. | 15,135 | 53,050 | 36,500 | 25,050 | 43,725 |
| Coke in drop, lb. | 1,400 | 2,000 | 2,000 | 2,500 | 1,400 |
| Total coke consumed, lb. | 14,735 | 51,050 | 34,500 | 22,550 | 42,325 |
| Melting ratio (iron-coke) | 14.5 : 1 | 8.6 : 1 | 9.25 : 1 | 13.3 : 1 | 6.6 : 1 |
| Temperature of iron at spout, ° F. | 2,640 | 2,800 | 2,300 | 2,500–2,550 | 2,650 |
| Air volume, cu. ft. per min. | 3,200 | 9,750 | 8,000 | 9,700 | 7,400 |
| Blast pressure, oz. | 12–13 | 20 | 8–10 | 16 | 24–25 |
| EFFICIENCY CALCULATION BASED ON HOURLY MELTING RATE | | | | | |
| Iron melted, tons | 11 | 20 | 16 | 18.75 | 17.5 |
| Total heat to melt and superheat to pouring temperature, B.t.u.* | 12,034,000 | 23,360,000 | 15,008,000 | 19,762,500 | 19,250,000 |
| Carbon content of coke, percent | 90 | 88 | 87 | 90.4 | 89.1 |
| Coke consumed, lb. | 1,510 | 4,650 | 3,450 | 2,820 | 4,700 |
| Carbon content of coke consumed, lb. | 1,360 | 4,090 | 3,000 | 2,540 | 4,210 |
| Total heat evolved (14,550 B.t.u. per lb. carbon), B.t.u. | 19,788,000 | 59,510,000 | 43,650,000 | 36,957,000 | 61,255,500 |
| Melting efficiency, percent, $= \left\{ \frac{\text{(B.t.u. required} \times 100)}{\text{B.t.u. evolved}} \right\}$ | 60.8 | 39.2 | 34.4 | 53.5 | 31.4 |
| Theoretical air necessary, cu. ft. (Carbon consumed × 143 cu. ft. per lb.) | 194,480 | 585,000 | 429,000 | 363,320 | 602,030 |
| Total air supplied, cu. ft. (Cu. ft. per min. × 60) | 192,000 | 585,000 | 480,000 | 582,000 | 444,000 |
| Excess or deficiency of air, percent | − 1.00 | 0.00 | + 11.9 | * See note | − 26.2 |

* Calculation of B.t.u. necessary to melt and superheat iron to pouring temperature: Let $Q$ = heat required to melt and superheat 1 lb. of iron; $T_m$ = melting temperature = 2260° F, average; $T_n$ = average temperature of blast air = 60° F; $T_p$ = temperature of iron at spout, deg. F; $H$ = specific heat of iron up to 2260° F = 0.19; $h$ = specific heat of iron up to 2800° F = 0.23; $L$ = latent heat of fusion of iron = 41.5 (say 42) B.t.u. per lb. Then

$$Q = H(T_m - T_n) + h(T_p - T_m) + L = \text{B.t.u. required} = \text{pounds melted} \times Q.$$

For example, as in No. 5: Heat for melting = 0.19 (2260 − 60) = 0.19 × 2200 = 418
Heat for superheating 0.23 (2650 − 2260) = 0.23 × 390 = 90
Latent heat of fusion = 42
Total = $Q$ = 550

Total B.t.u. = 35,000 × 550 = 19,250,000.

Efficient cupola operation comprises more than obtaining hot iron with the lowest coke consumption. Smoothness of operation, uniformity, freedom from bridging and tap hole difficulties, and economy by avoiding frequent lining troubles are factors. The

A. F. A. Committee on Gray Iron Castings (*Trans.* A. F. A., vol xxxix, p. 49, 1931) has formulated recommendations for operating a 54-in. cupola, summarized below.

**RECOMMENDATIONS FOR OPERATION OF A 54-IN. CUPOLA. Lining.**—To line a new cupola or reline an old one, use good grade cupola block of proper dimensions, laid with close joints filled with fireclay mixture. Typical mixtures are: *a.* 1 part fireclay, 1 part molding sand, 1 part sharp core sand; *b.* 1 part blue clay, 1 part finely-ground firebrick, 2 parts fire sand or heavy molding sand; *c.* 1 part fireclay, 2 parts molding sand. The shape of lining depends on practice, and may be classified under two types: 1. Straight side, *i.e.*, straight from bottom doors to charging door. 2. Boshed, *i.e.*, a 3 to 6 in. projection at top of tuyeres, tapering to the straight side just above melting zone. After each heat, chip out badly burned brick, and patch with new brick, maintaining original outlines as closely as possible. A new breast brick or tap hole should be used if tap hole has become too large for desired flow of metal. Tap hole should be about 1 in. diam. at the shell for intermittent tapping and not over 2 or 3 in. long; for continuous operation size should be adjusted to melting rate. Its location is just at the level of the sand bottom. The slag hole should be located 2 to 4 in. below bottom of tuyeres. Its size depends on thickness of lining and size of cupola, and usually is 1 to 2 in., either round or square.

**Tuyeres.**—Area of tuyeres should be approximately $^1/_3$ area of cupola at the tuyeres. Height of tuyeres should be 10 to 20 in. above center of the sand bottom, depending on amount of iron required at one tap. They should be as nearly continuous as possible with a vertical opening 3 to 5 in. high. For ordinary conditions, use only one row of tuyeres.

**Bottom.**—Bottom doors should be securely propped and rigid, to eliminate possible cracking of sand bottom and running out of iron. Bottom sand should be riddled rich heap sand, at same temper as for molding, carefully rammed and sloped with a 1 to 2 in. pitch from all directions to the tap hole. All-new, or too wet sand should not be used. Make bottom 5 to 6 in. thick at tap hole.

**Lighting.**—Flat wood is laid to protect the bottom while charging heavier wood. The wood usually is ignited through the tuyeres, or by ignited oil-soaked rags dropped from the charging door; sufficient wood then is added to ignite the coke thoroughly. From 65 to 75% of the coke for the bed next is charged and allowed to burn until the whole bed is cherry red. The remaining coke is added and leveled to receive the charge of iron. The coke bed should be high enough to melt iron in 8 to 10 min. after blast is put on. This usually requires 2,000 to 2,400 lb., depending particularly on height of tuyeres above the bottom and shape of cupola well. The above amount should raise the bed 40 to 50 in. above the tuyeres. In starting a new cupola, it is safe to use a coke bed 45 in. above the upper edge of the tuyeres. The upper edge of the melting zone is easily located by examining the lining after the first heat. It is wise to determine height of bed coke with a measuring rod. At least one hour should elapse between lighting of the cupola and putting on of the blast. A period of 2 hr. is better to thoroughly warm the charge before melting begins.

**Charges and Charging.**—All materials should be weighed and loaded systematically according to size of pieces and composition. The proper system is best determined by experiment and practice. Good practice charges the heavy pieces around the lining of the cupola; this tends to better blast penetration. Under ordinary conditions, the charges should be 2,000 to 3,000 lb., of the same weight, and in the approximate ratio of 10 lb. of iron per lb. of coke. Charging may be either mechanical or by hand. It is not good practice to use in the same heat, mixtures of widely different composition. If this is necessary, the mixtures may be partly separated by an extra charge of coke.

**Flux** should be a good grade of limestone. For long heats and continuous melting, the limestone should be 2 to 4 % of weight of the iron charge, with 4 to 5 lb. of fluorspar, per charge.

**Blast.**—Volume of air required depends on rate of melting, coke ratio and quality of coke; 30,000 cu. ft. of air per ton of iron melted is generally assumed. Blast pressure varies with style and size of tuyeres, height of stack, size of coke and metal used in the charge. Pressure ranges from 6 oz. to 24 oz., depending on the above conditions. For uniform melting, volume should be constant and pressure varied. Every cupola should have volume and pressure gages and means to adjust the volume of air delivered to the wind box. For a given cupola and blower, melting rate increases as the square root of the pressure. Power required varies as cube of melting rate.

For a 54-in. cupola, one blower manufacturer recommends a blower of 22, 25, or 29 cu. ft. displacement. To obtain 6000 cu. ft. per min. in accordance with the recommendation, it will be necessary to operate the blowers at 272, 240, or 206 r.p.m., respectively. With 10% slippage allowance the speeds will be 299, 264, or 227 r.p.m. To use the blower as a volume meter, it is necessary only to know its displacement and its speed. In calculations, the 10% slip correction should be included. For example, with 25 cu. ft. blower displacement and 250 r.p.m., tachometer reading, the volume supplied, allowing 10% slip, will be $0.90 \times 250 \times 25 = 5625$ cu. ft. per min., which is less than desired; hence the blower should run at 264 r.p.m. While the blower can be used as a volume meter, slippage is difficult to determine accurately; a standard orifice is recommended for accurate measurement of air supply.

CUPOLA COMBUSTION.—Since the combustion of 1 lb. of carbon to $CO_2$ or CO evolves 14,600 and 4,450 B.t.u. respectively, cupola economy demands that operation shall produce a minimum amount of CO. A clearly defined melting zone is established in a completely charged cupola, which should be slightly above the tuyere area, as no melting occurs below this level. If air passes through the cupola in uniform quantity and velocity and encounters a cylindrical coke column, uniform in shape, size and area exposed, combustion occurs uniformly. This condition is ideal rather than practical. The U. S. Bureau of Mines, by analysis of gas samples obtained from the center to the outside edge at different levels, has located the melting zone as regards its upper level. In a cupola lined to 27 in. diam. at the tuyere level, with 1000 cu. ft. of air per min. at

**Table 10.—Capacity Table of Steel Pressure Blowers for Cupola Service***

| Cupola | | Tuyeres | | Iron | Air | | Blower | | | | | Pipe | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diam. of inside lining, in. | Area of inside lining, sq. in. | Number | Total area, sq. in. | Melting capacity, lb. per hr. | Cu. ft. air per min. required | Pressure, oz. per sq. in. | Speed, r.p.m. | Pressure produced, oz. per sq. in. | Capacity of blower, cu. ft. free air per min. | Brake Hp. to drive blower | Diam. of outlet, in. | Diam., in. | Air velocity in pipe, ft. per min. |
| 18 | 254 | 2 | 51 | 1,775 | 444 | 7.5 | .... | .... | .... | .... | .... | .. | .... |
| 23 | 415 | 4 | 83 | 3,090 | 773 | 8.5 | 4150 | 9 | 752 | 4.20 | 5 3/4 | 9 | 1800 |
| 27 | 573 | 8 | 115 | 4,420 | 1,105 | 9.2 | 3740 | 10 | 1,093 | 6.78 | 6 3/4 | 10 | 1900 |
| 32 | 804 | 8 | 161 | 6,480 | 1,620 | 10.0 | 2880 | 10 | 1,840 | 11.40 | 8 3/4 | 13 | 1900 |
| 37 | 1075 | 8 | 215 | 8,980 | 2,245 | 10.76 | 2595 | 10 | 2,280 | 14.13 | 9 3/4 | 15 | 1900 |
| 42 | 1385 | 12 | 277 | 11,960 | 2,990 | 11.46 | 2470 | 11 | 2,910 | 19.85 | 10 3/4 | 16 | 2000 |
| 45 | 1590 | 12 | 318 | 13,960 | 3,490 | 11.88 | 2080 | 11 | 3,930 | 26.80 | 12 1/2 | 19 | 2000 |
| 48 | 1810 | 12 | 362 | 16,120 | 4,040 | 12.25 | 2170 | 12 | 4,110 | 30.60 | 12 1/2 | 19 | 2100 |
| 54 | 2290 | 12 | 458 | 21,050 | 5,260 | 13.00 | 1995 | 13 | 5,360 | 43.25 | 14 | 21 | 2200 |
| 60 | 2827 | 12 | 566 | 26,630 | 6,650 | 13.70 | 1728 | 13 | 7,010 | 56.66 | 16 | 24 | 2200 |
| 66 | 3421 | 12 | 685 | 33,000 | 8,250 | 14.33 | 1540 | 13 | 8,390 | 67.66 | 17 1/2 | 26 | 2200 |
| 72 | 4071 | 12 | 814 | 40,150 | 10,020 | 15.00 | 1393 | 13 | 10,140 | 82.00 | 19 1/4 | 29 | 2200 |
| 78 | 4778 | 12 | 955 | 48,000 | 12,000 | 15.66 | 1318 | 14 | 12,520 | 109.00 | 21 | 32 | 2300 |
| 84 | 5542 | 12 | 1109 | 56,750 | 14,200 | 16.20 | 1410 | 16 | 13,380 | 132.75 | 21 | 33 | 2400 |
| 87 | 5945 | 16 | 1189 | 61,600 | 15,400 | 16.50 | 1340 | 16 | 15,675 | 155.66 | 22 3/4 | 35 | 2400 |

* Air pressure less than given in the table indicates too much air flowing, with more power required to drive the blower than is specified. Blast gate in pipe between blower and cupola should be closed sufficiently to reduce the Hp. to that given in table. Volume then will be correct for highest melting efficiency obtainable with cupola, regardless of proportions or resistance, table being based on the greatest resistance usually encountered.

5.3 oz. per sq. in. pressure, the percentage of free oxygen at any given point was taken to locate the area of complete combustion. The upper level of the melting zone thus was shown to be more or less conical, extending from a height of 7 in. at the lining to about 28 in. at the center.

**AIR SUPPLY TO CUPOLAS.**—Two common types of blowing units are used to supply blast pressure to cupolas. The positive pressure type delivers a fixed volume of air, regardless of the pressure against which it operates. Cupola blowers of this type are driven by motors coupled to the main shaft of the blower, or by multiple V-belt drives. Table 10 gives capacities of positive pressure blowers for cupola service.

The other common type of unit used is the centrifugal compressor. Certain centrifugal compressors have an auxiliary blast gate, with suitable motor-driven gear control and electrical operating devices, to permit delivery of a fixed weight of air into the cupola blast pipe. Based on using coke of 88% fixed carbon, and allowing 5% air leakage, the rate of air delivery for various melting ratios is

| Melting ratio............ | 12 to 1 | 11 to 1 | 10 to 1 | 9 to 1 | 8 to 1 | 7 to 1 |
|---|---|---|---|---|---|---|
| Lb. of air per min......... | 25.3 | 26.0 | 26.8 | 27.8 | 29.0 | 30.6 |

See Kent's Mechanical Engineers' Handbook—Power, pp. 1–55 to 1–88, forming Vol. 2 of this series, for additional information.

The blast gate control automatically compensates for changes in cupola restriction and changes in atmospheric conditions, and maintains the flow of a constant weight of air into the cupola.

The usual relation of size of cupola, quantity of iron melted and air supply is shown in Table 11. The diameter of blast pipes for various volumes of air is given in Table 12.

**Table 11.—Air Supply Required for Cupolas**

| Tons of Iron per hr. | Inside Diam. of Cupola, in. | Cu. Ft. of Air per min. | Tons of Iron per hr. | Inside Diam. of Cupola, in. | Cu. Ft. of Air per min. | Tons of Iron per hr. | Inside Diam. of Cupola, in. | Cu. ft. of Air per min. |
|---|---|---|---|---|---|---|---|---|
| 1 | 23 | 500 | 11 | 54 | 5,500 | 21 | 78 | 10,500 |
| 2 | 27 | 1000 | 12 | 60 | 6,000 | 22 | 78 | 11,000 |
| 3 | 30 | 1500 | 13 | 60 | 6,500 | 23 | 78 | 11,500 |
| 4 | 32 | 2000 | 14 | 60 | 7,000 | 24 | 84 | 12,000 |
| 5 | 36 | 2500 | 15 | 66 | 7,500 | 25 | 84 | 12,500 |
| 6 | 39 | 3000 | 16 | 66 | 8,000 | 26 | 84 | 13,000 |
| 7 | 42 | 3500 | 17 | 66 | 8,500 | 27 | 90 | 13,500 |
| 8 | 45 | 4000 | 18 | 72 | 9,000 | 28 | 90 | 14,000 |
| 9 | 48 | 4500 | 19 | 72 | 9,500 | 29 | 90 | 14,500 |
| 10 | 54 | 5000 | 20 | 72 | 10,000 | 30 | 90 | 15,000 |

### Table 12.—Diameters of Blast Pipes *

| Tons of Iron per hr. | Inside Diam. of Cupola, in. | Cubic Feet of Air per min. | Length of Pipe, ft. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 20 | | 40 | | 60 | | 80 | | 100 | | 120 | | 140 | |
| | | | Pressure Drop, oz. | | | | | | | | | | | | | |
| | | | 1/4 | 1/2 | 1/4 | 1/2 | 1/4 | 1/2 | 1/4 | 1/2 | 1/4 | 1/2 | 1/4 | 1/2 | 1/4 | 1/2 |
| | | | Diameter of Pipe, in. | | | | | | | | | | | | | |
| 1 | 23 | 500 | 6 | 5 | 7 | 6 | 7 | 6 | 8 | 7 | 9 | 8 | 9 | 8 | 9 | 8 |
| 2 | 27 | 1,000 | 8 | 7 | 9 | 8 | 10 | 9 | 11 | 9 | 11 | 10 | 12 | 11 | 12 | 11 |
| 3 | 30 | 1,500 | 10 | 8 | 11 | 10 | 11 | 10 | 12 | 11 | 13 | 11 | 13 | 12 | 14 | 12 |
| 4 | 32 | 2,000 | 11 | 9 | 12 | 11 | 13 | 12 | 14 | 12 | 15 | 13 | 15 | 14 | 16 | 14 |
| 5 | 36 | 2,500 | 12 | 10 | 14 | 12 | 15 | 13 | 15 | 14 | 16 | 14 | 17 | 15 | 17 | 15 |
| 6 | 39 | 3,000 | 13 | 11 | 15 | 13 | 16 | 14 | 17 | 15 | 18 | 15 | 18 | 16 | 18 | 16 |
| 7 | 42 | 3,500 | 13 | 12 | 15 | 13 | 17 | 15 | 17 | 15 | 18 | 16 | 19 | 17 | 20 | 18 |
| 8 | 45 | 4,000 | 15 | 12 | 16 | 15 | 18 | 15 | 18 | 16 | 19 | 17 | 20 | 18 | 21 | 18 |
| 9 | 48 | 4,500 | 15 | 13 | 17 | 15 | 18 | 16 | 19 | 17 | 20 | 18 | 21 | 19 | 22 | 19 |
| 10 | 54 | 5,000 | 15 | 13 | 18 | 15 | 19 | 17 | 20 | 18 | 21 | 18 | 22 | 19 | 23 | 20 |
| 11 | 54 | 5,500 | 16 | 14 | 18 | 16 | 20 | 17 | 21 | 18 | 22 | 19 | 23 | 20 | 23 | 20 |
| 12 | 60 | 6,000 | 17 | 14 | 19 | 17 | 20 | 17 | 21 | 19 | 22 | 20 | 23 | 21 | 24 | 21 |
| 13 | 60 | 6,500 | 17 | 14 | 19 | 17 | 21 | 18 | 23 | 19 | 23 | 20 | 24 | 21 | 25 | 22 |
| 14 | 60 | 7,000 | 18 | 15 | 20 | 18 | 22 | 19 | 23 | 20 | 24 | 21 | 25 | 22 | 26 | 23 |
| 15 | 66 | 7,500 | 18 | 16 | 21 | 18 | 22 | 19 | 24 | 21 | 25 | 22 | 26 | 22 | 27 | 23 |
| 16 | 66 | 8,000 | 18 | 16 | 22 | 18 | 23 | 20 | 24 | 22 | 26 | 22 | 26 | 23 | 27 | 24 |
| 17 | 66 | 8,500 | 18 | 16 | 22 | 18 | 23 | 20 | 24 | 22 | 26 | 22 | 27 | 24 | 28 | 24 |
| 18 | 72 | 9,000 | 18 | 17 | 22 | 18 | 24 | 21 | 25 | 22 | 27 | 23 | 27 | 24 | 28 | 25 |
| 19 | 72 | 9,500 | 20 | 17 | 23 | 20 | 24 | 22 | 26 | 23 | 28 | 23 | 28 | 25 | 29 | 26 |
| 20 | 72 | 10,000 | 20 | 18 | 23 | 20 | 25 | 22 | 27 | 23 | 28 | 24 | 29 | 25 | 30 | 26 |
| 21 | 78 | 10,500 | 21 | 18 | 24 | 21 | 26 | 23 | 27 | 23 | 29 | 25 | 30 | 26 | 30 | 26 |
| 22 | 78 | 11,000 | 21 | 18 | 24 | 21 | 27 | 23 | 28 | 24 | 29 | 26 | 30 | 27 | 31 | 27 |
| 23 | 78 | 11,500 | 21 | 19 | 25 | 21 | 27 | 24 | 28 | 25 | 30 | 26 | 30 | 27 | 31 | 27 |
| 24 | 84 | 12,000 | 22 | 19 | 25 | 22 | 28 | 24 | 28 | 25 | 31 | 26 | 31 | 27 | 32 | 28 |
| 25 | 84 | 12,500 | 22 | 19 | 26 | 22 | 28 | 24 | 29 | 26 | 31 | 27 | 32 | 28 | 33 | 28 |
| 26 | 84 | 13,000 | 22 | 19 | 26 | 22 | 28 | 24 | 29 | 26 | 31 | 27 | 32 | 28 | 33 | 28 |
| 27 | 90 | 13,500 | 23 | 20 | 26 | 23 | 28 | 24 | 30 | 26 | 31 | 27 | 32 | 28 | 34 | 28 |
| 28 | 90 | 14,000 | 23 | 20 | 27 | 23 | 29 | 25 | 30 | 27 | 32 | 28 | 33 | 29 | 34 | 29 |
| 29 | 90 | 14,500 | 23 | 20 | 27 | 23 | 29 | 26 | 31 | 27 | 32 | 28 | 33 | 29 | 34 | 30 |
| 30 | 90 | 15,000 | 24 | 21 | 27 | 24 | 29 | 26 | 31 | 27 | 32 | 28 | 34 | 30 | 35 | 30 |

* The minimum radius of each turn should be equal to the diameter of the pipe. For each turn thus made add three feet in length, when using this table. If the turns are of less radius, the length added should be increased proportionately.

**OPERATING RECOMMENDATIONS.**—Cupolas should be operated on a fixed schedule, with a definite time for lighting, for start of charging, for starting blast, for tapping out and for opening slag hole. The tap hole should be carefully gaged to give desired rate of flow of iron, whether tapping is continuous or intermittent. Blast should be maintained according to established foundry practice. All materials used should be checked and recorded daily. Successful cupola operation depends on the adoption of a standard practice. Variations are necessary only when standard practice can be improved.

**CALCULATING CUPOLA MIXTURES.**—No set rule can be given for calculating cupola mixtures, because of variations in cupola operation. The factors influencing control of some elements are: Blast pressure; volume; moisture and temperature; amount and character of coke used; depth of zone or well.

**Silicon.**—Allow 10% loss under normal conditions. When more highly concentrated Si alloys are added, the loss will be higher. Loss of Si in cast iron is due to oxidation; it is controlled principally by blast pressure. A high pressure produces the maximum, and a low pressure the minimum loss, other conditions being constant.

**Sulphur.**—Allow for 0.03 to 0.05 point increase. The increase of S is controlled principally by the coke used. A low-sulphur coke (0.5 to 0.7%) will produce minimum, and a high-sulphur coke (0.7 to 0.9%) maximum S in the iron.

**Phosphorus** remains practically constant; no allowances need be made.

**Manganese**—Allow for a 20 to 30% loss. Mn loss is due to oxidation. Mixtures containing highly-concentrated Mn alloys probably will show maximum Mn loss.

**Carbon.**—On account of the many factors involved, calculation of total carbon (T.C.) is difficult. Steel additions affect T.C. in cast iron. The character and amount of coke, as well as the characteristics of the cupola, also influence T.C.

**Methods of Calculating Mixtures.**—Assume that a 3000-lb. charge containing 25% steel is sought for heavy machinery castings, composition to be: Si, 1.35–1.40; S, 0.07–0.09;

P, 0.25–0.27; Mn, 0.50–0.55. The following classes of stock are available in the yard:

| | Si | S | P | Mn |
|---|---|---|---|---|
| No. 2 pig-iron | 2.10 | 0.032 | 0.505 | 0.64 |
| Malleable iron | 1.80 | .033 | .165 | .75 |
| No. 1 machinery scrap | 1.90 | .085 | .405 | .60 |
| Steel scrap | 0.30 | .050 | .050 | .50 |
| No. 1 pig-iron | 2.90 | .030 | .530 | .84 |
| Returned scrap, gates, sprues, etc. | | | | |

Bearing in mind the increases and decreases of the elements in melting, the following method of calculation can be used: Draw from the stock piles quantities of the various irons as shown in Table 13. Opposite each iron enter the percentage content of each element as Si, S, etc., expressed as points per lb. Multiply the points per lb. of each element in each lot of iron by the number of pounds in that lot, and enter the product as total points. Divide the sum total points of each element by the total weight of all the lots of iron in the mixture. Add to or subtract from the total points of each element, the gain or loss of that element in melting. The sum or difference as the case may be is the percentage of each element in the iron resulting from the mixture. Note that the gain in S is expressed in points and not in percentage as is the case with the other elements.

A daily record should be kept of all mixtures, on a form similar to Table 13.

**ALLOYS.**—At times it is desired to cast alloy-irons. Alloys of Cr, Ni, Mo, Vn, and Ti can be had in a form for introduction in spout or ladle. High-silicon irons, as silvery iron (8.00 to 10.00% Si) and Bessemer ferro-silicon (12.00 to 15.00% Si), come in pigs and are mixed with the cupola charge when higher Si is wanted than is available from stock piles. Fe-Si alloy (50% and 80% Si), both lump and ground, also is used to increase Si in cast iron. It generally is ground for ladle additions. High-Mn irons, as spiegel (19.00 to 21.00% Mn), in pig form, is added to the charge in the cupola. Fe-Mn (80.00% Mn) in lumps is added in the cupola; the ground is form added to the ladle. These alloys are used when higher Mn is wanted than is available from stock piles. Fe-Si and Fe-Mn may be obtained in briquets which are added in the cupola.

**MELTING LOSSES.**—Melting loss is the difference in weight between iron charged and that recovered in the form of castings, sprues and other scrap. It is due to oxidation of the metal during melting and to a reduction in the silicon and manganese in the iron, also by oxidation. Melting loss varies with class of work and with cupola practice. Table 14 calculated from performances of cupolas, given by Payne, shows the average melting loss in 39 foundries doing a variety of work.

The melting loss frequently appears to be higher than it actually is, due to the weight of adhered sand on the pig iron, sprues, etc., and rust attached to purchased scrap. Moldenke gives the following percentage figures as the average melting loss of the usual

**Table 13.—Method of Calculating Cupola Mixtures**

| Lb. | Iron | Silicon | | Sulphur | | Phosphorus | | Manganese | |
|---|---|---|---|---|---|---|---|---|---|
| | | Points per lb. | Total Points | Points per lb. | Total Points | Points per lb. | Total Points | Points per lb. | Total Points |
| 600 | No. 2 Pig | 2.10 | 1260 | 0.032 | 19.20 | 0.505 | 303.00 | 0.64 | 384 |
| 900 | Malleable | 1.80 | 1620 | .033 | 29.70 | .165 | 148.50 | .75 | 675 |
| 750 | No. 1 Mchy. scrap | 1.90 | 1425 | .085 | 63.75 | .405 | 303.75 | .60 | 450 |
| 750 | Steel scrap | 0.30 | 225 | .050 | 37.50 | .050 | 37.50 | .50 | 375 |
| 3000 = Totals = | | 4530 | | 150.15 | | 792.75 | | 1884 | |
| Total ÷ 3000 | | 1.51 | | 0.05005 | | 0.264 | | 0.628 | |
| Gain or loss | | −10% = .151 | | +0.040 | | 0 | | −20% = 0.1256 | |
| Net percentage in iron | | 1.359 | | 0.09005 | | 0.264 | | 0.5024 | |

**Table 14.—Melting Loss in Various Types of Foundries**

| | General Jobbing | Light Jobbing | Light Machinery | Heavy Machinery | Stove Plate | Sanitary Ware | Agricultural | Railroad Castings | Floor Plates, Grate Bars |
|---|---|---|---|---|---|---|---|---|---|
| Number of records | 4 | 4 | 5 | 4 | 3 | 8 | 5 | 3 | 2 |
| Percent melt in castings | 83.0 | 74.4 | 73.9 | 73.6 | 62.3 | 71.4 | 68.5 | 72.3 | 79.6 |
| Percent melt in returns | 8.8 | 20.0 | 18.7 | 20.6 | 34.3 | 23.6 | 26.7 | 22.6 | 13.2 |
| Percent melting loss | 8.2 | 5.6 | 7.4 | 5.8 | 3.4 | 5.0 | 4.8 | 5.1 | 7.2 |

foundry metals: Machine-cast pig iron, 0.30; sand-cast pig iron, 1.00; car wheels, 2.00; first quality machinery scrap, 2.50; light machinery scrap, 3.50; stove plate scrap, 8.00.

## Special Types of Cupolas

During the ten years, 1925–1935, remarkable strides were made in increasing the efficiency, flexibility and application of the foundry cupola. A few successful types are described below.

**THE GRIFFIN HOT BLAST PROCESS** was developed by the Griffin Wheel Co., Chicago. The distinguishing feature of the system is the application of the recuperative principle to cupola operation. Cupola gases at about 1700° F., and containing CO are drawn into a duct surrounding the cupola a short distance below the charging door. They are burned in a refractory-lined heating chamber in the presence of excess air bled from the blast line, raising the temperature to about 1800° F. The burned gases pass into either a recuperative or a regenerative preheater and are discharged through a supplementary stack. Blast air from a positive pressure blower passes around and across the ducts conveying the burned gases from the combustion chamber, and thence into the main hot blast pipe to the tuyeres. The arrangement of the Griffin cupola embodies a continuous pouring spout with an inverted weir for front slagging. The efficiency of the hot blast cupola is claimed to be 55%, as compared with a normal efficiency of 37%.

**THE MOORE HOT BLAST CUPOLA** is a development of the hot-blast principle made necessary by individual requirements, and installed by the American Cast Iron Pipe Co., Birmingham, Ala. This comprises 8 vertical pipes in the wall of the cupola above the melting zone. These pipes are of low-silicon, high-carbon, low-phosphorus cast iron designed to resist heat and abrasion of the coke and iron charges. Blast from a bustle pipe around the cupola enters the upper end of the preheating pipes, and passes downward, to enter the cupola at the tuyeres. The blast temperature entering the tuyeres is estimated at 300° F.

**THE POUMAY CUPOLA.** (R. T. Rolfe, Foundry Work and Metallurgy.)—Attempts have been made by means of auxiliary upper tuyeres to attain more effective combustion by burning the CO formed by the reduction of the initial $CO_2$. In the Poumay system, the production of CO in the melting zone is considered advantageous. It aims to induce incomplete combustion in the lower part of the cupola, thus avoiding oxidation. The arrangement of the regenerative tuyeres in the Poumay cupola causes the fractional combustion of CO higher in the shaft to preheat the descending charges, and yield hot metal, with low coke consumption. Poumay states that success or failure depends on the position, arrangement, and number of regenerative tuyeres. He also claims: 1. All useful gases are burned in the cupola below the level of the charging door; 2. Oxidation is reduced to a minimum with resulting low metal loss; 3. Melting is rapid; 4. Sulphur

**Table 15.—Dimensions and Operating Characteristics of Cupolas**

(Whiting Foundry Equipment Co., Harvey, Ill.)

| Diam. Inside of Lining, in. | Thickness of Lining, in. | Diam. of Shell, in. | No. of Tuyeres | Total Area of Tuyeres, sq. in. | Capacity, tons per hour | Bed Coke, lb. | | Other Charges, lb. | | | Air | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | With Upper Tuyeres | Without Upper Tuyeres | Coke | Iron * | Limestone | Blast Pressure, oz. | Cubic Feet per Min. | Blast Pipe Diam., in. |
| 18 | 4 1/2 | 27 | 2 | 32 | 1/4–1/2 | 150 | ...... | 40 | 400 | 6 | 7–8 | 300 | 4† |
| 23 | 4 1/2 | 32 | 4 | 88 | 1/2–1 | 250 | ...... | 60 | 600 | 9 | 8–10 | 600 | 6 3/8 |
| 27 | 4 1/2 | 36 | 8 | 118 | 1–2 | 400 | 350 | 80 | 800 | 12 | 8–10 | 1,000 | 10 |
| 27 | 7 | 41 | 8 | 121 | 1–2 | 400 | 350 | 80 | 800 | 12 | 8–10 | 1,000 | 11 3/4 |
| 32 | 4 1/2 | 41 | 8 | 121 | 3–5 | 600 | 500 | 100–120 | 1200 | 18 | 10–12 | 2,500 | 11 3/4 |
| 32 | 7 | 46 | 8 | 161 | 3–5 | 600 | 500 | 100–120 | 1200 | 18 | 10–12 | 2,500 | 14 |
| 37 | 7 | 51 | 8 | 182 | 5–6 | 850 | 650 | 140–160 | 1600 | 25 | 10–12 | 3,000 | 14 |
| 42 | 7 | 56 | 12 | 263 | 6–7 | 1100 | 900 | 180–200 | 2000 | 30 | 12–14 | 3,500 | 14 |
| 45 | 9 | 63 | 12 | 314 | 7–9 | 1250 | 1100 | 200–230 | 2300 | 35 | 12–14 | 4,500 | 17 |
| 48 | 9 | 66 | 12 | 347 | 9–10 | 1450 | 1300 | 225–250 | 2500 | 40 | 12–14 | 5,000 | 17 |
| 54 | 9 | 72 | 12 | 466 | 10–12 | 1900 | 1700 | 300–350 | 3500 | 50 | 12–14 | 6,000 | 18 1/8 |
| 60 | 9 | 78 | 12 | 546 | 12–14 | 2450 | 2100 | 350–400 | 4000 | 60 | 14–16 | 7,000 | 18 1/8 |
| 66 | 9 | 84 | 12 | 640 | 14–18 | 3000 | 2750 | 450–500 | 5000 | 75 | 14–16 | 9,000 | 24 7/8 |
| 72 | 9 | 90 | 12 | 802 | 18–21 | 3750 | 3400 | 550–600 | 6000 | 90 | 14–16 | 10,000 | 24 7/8 |
| 78 | 9 | 96 | 12 | 919 | 21–24 | 4400 | 3900 | 650–700 | 7000 | 100 | 14–16 | 12,000 | 24 7/8 |
| 84 | 9 | 102 | 12 | 919 | 24–27 | 5100 | 4650 | 750–800 | 8000 | 120 | 14–16 | 13,500 | 24 7/8 |
| 87 | 10 1/2 | 108 | 16 | 860 | 27–31 | 5500 | 5000 | 850–900 | 9000 | 130 | 14–16 | 15,000 | 30 |

* Bed charge of iron the same weight as the other charges. † Two blast pipes.

content of metal is reduced; 5. Fuel consumption is reduced. The essential principle of the cupola is a row of main tuyeres and a secondary row entering the cupola at varying heights. The air for the upper, or auxiliary, tuyeres is taken from the main bustle pipe and the volume of air entering each tuyere depends on pressure and resistance.

**THE BALANCED-BLAST CUPOLA.**—The balanced-blast cupola, developed by J. E. Fletcher and patented for use of British Cast Iron Research Assoc. members, differs from the Poumay cupola in that primary control is provided for the lower row of tuyeres. The upper row is proportioned according to predetermined conditions to provide the proper volume of entering air. The main row of tuyeres is placed below the bustle pipe, each provided with hand-adjusted valves. The aperture into the cupola slants down to reduce the velocity of the entering blast. Closing the valves on the main tuyeres raises the pressure of the blast in two or more rows of upper tuyeres. One setting of the main valves is said to be suitable for one particular type of charge, size of scrap, pig iron, fuel, etc. J. E. Hurst reports that on a 54-in. balanced-blast cupola, a coke ratio of 18 to 1 was obtained with an increase in melting rate from 10 to 14 tons per hr.

**Briquetting Foundry Borings.** (A. L. Stillman, *Trans.* A. F. A., xxvi, p. 493, 1917.)—Cast-iron borings charged directly into the cupola have carbon and silicon burned out to such an extent that the product approaches low quality white iron. Briquetting the borings removes most of the objections to their use in the cupola. A series of tests were made at the Baldwin Locomotive Works at Philadelphia, Pa., using the Ronay metal briquetting press to consolidate the borings. The borings were compressed under a pressure of 30,000 lb. per sq. in. into briquettes 5 in. diameter and 7 to 8 in. high, weighing about 28 lb. Their density was about 80% of the normal density of iron. Three test mixtures were made for miscellaneous castings, and three for castings of locomotive cylinders. The regular mixtures in each case were composed of pig iron, scrap and car wheels in varying percentages. The object of the tests was to determine the effect of substituting briquettes for varying percentages of these several constituents. The composition, chemical analyses and strength of castings from the several mixtures are given in Table 16, the charge in each case being 3000 lb. From the foundry point of view, and considering the work for which the iron was designed, the mixtures containing briquettes were as well, if not better, adapted to that work than the regular mixtures. In loading the charge, the briquettes were placed on top so that they would not be subject to heavy impact in charging.

The Southwark Foundry & Machine Co., Philadelphia, has developed an automatic hydraulic press to exert a pressure of 600,000 lb. One of these, installed by Dodge Brothers, Detroit, was followed by installations in other plants operating cupolas and having quantities of borings or chips. A 75-Hp. motor, drives the machine at 18 r.p.m., producing 18 7-lb. briquettes per min. or 3.78 net tons per hr. Borings are fed to the machine from a storage hopper by conveyors. Briquettes are removed from the machine to the cupola charging deck by conveyors. The cost of briquettes so made in 1929 was $1.91 per net ton. Briquettes used are 16.67% of the total melt, being placed on top of each successive charge, followed by the coke for the next charge. The briquette melting loss is slightly under 10%. The briquettes do not disintegrate in the melting zone, many partly melted being found in the bottom drop at the end of the heat. No difference in percentage of good castings made followed the introduction of briquettes.

## The Air Furnace

**THE AIR FURNACE** is a hearth furnace with separate fire box and melting chamber. The flame passes over the bridge wall, through the melting chamber, the gases sweeping

Table 16.—Composition of Cupola Charges Using Cast-iron Briquettes

| Mixture | Charge | | | | Analysis | | | Tensile Strength, lb. per sq. in. |
|---|---|---|---|---|---|---|---|---|
| | Briquettes, Percent | Scrap, Percent | Car Wheels, Percent | Pig-Iron, Percent | Comb. Carbon, Percent | S, Percent | Si, Percent | |
| MISCELLANEOUS CASTINGS: | | | | | | | | |
| Regular | ....... | 60 | ....... | 40 | 0.56 | 0.061 | 2.25 | 31,140 |
| Test No. 1 | 10 | 60 | ....... | 30 | 0.61 | 0.093 | 2.12 | 31,030 |
| Test No. 2 | 10 | 50 | ....... | 40 | 0.62 | 0.084 | 1.64 | 23,770 |
| Test No. 3 | 20 | 50 | ....... | 30 | 0.69 | 0.087 | 1.86 | 30,850 |
| LOCOMOTIVE CYLINDERS: | | | | | | | | |
| Regular | ....... | 50 | 10 | 40 | 0.69 | 0.106 | 1.24 | 32,880 |
| Test No. 1 | 10 | 50 | 10 | 30 | 0.77 | 0.108 | 1.24 | 37,700 |
| Test No. 2 | 10 | 40 | 10 | 40 | 1.04 | 0.103 | 1.19 | 35,310 |
| Test No. 3 | 20 | 40 | 10 | 30 | 1.32 | 0.097 | 1.15 | 32,840 |

the hearth, and thence to the chimney. The gases partially melt the stock, but principally heat the walls and roof of the melting chamber to incandescence, radiation from which does most of the work of melting and superheating. Fuel and metal are not in direct contact, excepting gases and fine ash carried over. Hence, absorption of sulphur is not as great as in the cupola.

The type of air furnace generally used in gray iron foundries for production of high grade metal, is short, fairly wide and high, and is charged from the end. It is possible to introduce entire cannon, rolls and heavy pieces of scrap. The furnace is built for heavy tonnages, 25 tons being an ordinary, and 40 tons an occasional, charge. Natural draft generally is used, although powdered coal blown in under pressure also is used. Chimney height for natural draft ranges from 60 to 85 ft., 75 ft. being a good average. Good draft is necessary for combustion and to carry the gases to the far end of the melting chamber. Too much draft causes excessive temperatures in flue and chimney walls, and exposes the thin layer of molten iron to oxidation and consequent damage.

Products generally made in the air furnace include gun iron, chilled and sand rolls, steam and gas engine cylinders, and work requiring special physical qualities and great strength.

**Coal Consumption** of the air furnace ranges from 40 to 55 lb. per sq. ft. of grate per hr. At the latter figure the fires are forced, and unless stack capacity is ample, incomplete combustion and considerable waste of fuel result. For example, of two American air furnaces, one, burning 53.5 lb. of coal per sq. ft. of grate surface per hr., melted 3.65 lb. of iron per lb. of coal; the other burned 51.3 lb. of coal per sq. ft. of grate per hr. and melted 2.30 lb. of iron per lb. of coal. The same furnaces, each melting 22-ton heats, and burning 44 lb. of coal per sq. ft. of grate per hr., had ratios of 4.00 and 2.75 lb. of iron per lb. of coal, respectively.

**COMPARISON OF AIR FURNACE AND CUPOLA.**—Moldenke (Principles of Iron Founding) compares the air furnace and cupola as follows: 1. Air furnace iron is stronger and better than cupola iron. 2. With good coal, air furnace iron will be lower in sulphur. Compared with the best cupola practice, the air furnace has no advantage in oxidation; silicon drop in the cupola is 0.20, against 0.30 in the air furnace. The cupola usually is not run on low-silicon mixtures except for malleable castings, with great attention to melting to avoid oxidation. 3. The air furnace permits changing composition of the iron before tapping, an impossibility with the cupola. 4. The air furnace permits reduction of total carbon by addition of steel to the charge, up to 30%. This is not possible with the cupola, in which the steel absorbs carbon, the resulting total carbon in the iron being uncertain. 5. A large tonnage is available in one tap from the air furnace; several cupolas operated at the same time are necessary for the same tonnage. 6. The air furnace can use large heavy pieces of scrap which cannot be handled and melted in the cupola.

## Powdered Coal Furnaces

**THE BRACKELSBURG FURNACE** is a horizontal cylinder, with removable conical ends, giving access for relining and repair. It rotates at 15 ft. per min. in a tilting cradle, tires on the drum engaging power-driven rollers on the cradle. For charging or pouring it may be tilted 50°. The furnace burns 170-mesh powdered coal, in a burner directed into the end of the furnace when in melting position. The air pressure is 10 to 13 in. of water. A stack carries combustion products and ash out of the building. Recuperators may be used with this furnace. The advantages claimed are temperatures of 2700° to 2850° F., capable of advance to 3100°, with a preheater; rotation of the melting charge promotes thorough mixing, uniformity, and high quality. The ratio of air to fuel on a 2-ton furnace is given as 3 to 1 when cold, increasing to 5 to 1. A ratio of 6 to 1 is claimed for larger furnaces.

**THE SESCI FURNACE**, developed by Societe des Entreprises Speciales de Chauffage Industriel, Paris, is a rotating unit using a powdered fuel. The cylindrical furnace, of steel plates, angles, cast steel rings and end plates, rotates on rollers, driven at one end by a reversible electric motor through a chain drive. Rotation in either direction is from 20 to 360°. The fuel is 200-mesh powdered coal, fed by a screw distributor into a hot air pipe. The fuel, intimately mixed with preheated air, enters the burner, and thence the end of the furnace. Hot gases from the furnace preheat the air to 1100° F. An operating economy of 20% due to preheating is claimed. A melting temperature of 3270° F. is attainable, which permits pouring metal at 2900°. Fuel consumption is given as 16% for pig iron of 2.5% C, 1% Si. An English operator of several Sesci furnaces uses bituminous coal at the start of the heat, and changes to anthracite when the temperature of the entering blast is 390° F. Four furnaces supply 22 tons of iron, using an 80% scrap charge. A 5-ton furnace melts in 110 min. while 10-ton furnaces require 2 hr. The capacity of the battery of four furnaces is 80 tons per 8-hr. shift. The high temperature, and, in effect, superheating, together with production of low carbon iron from steel scrap by control of carbon, recommends this furnace for close chemical requirements.

## The Electric Furnace

**THE ELECTRIC FURNACE** for melting gray iron permits closer control of analyses and superheating; it also allows small heats to be made quickly and intermittently. Direct and indirect arc-type furnaces are used, both to melt down from cold stock and to refine molten metal from the cupola. Availability of inexpensive scrap, as borings and turnings, power rates and other economic factors influence the use of electric furnaces. The main advantages of electric furnace melting, according to L. J. Barton (Refining Metals Electrically) are: *a.* Superheating: lowers percentage of lost castings; improves appearance and solidity of castings; decreases size of heads, due to greater fluidity of metal; saves iron and labor by decreasing pig and ladle skulls; causes cleaner metal to enter the mold, as impurities tend rapidly to clear themselves from fluid iron, resulting in lower losses from sand and slag spots; gives iron of higher quality and strength. *b.* Refining: deoxidizes the iron, an advantage as great as if steel were being cast; lowers sulphur to any desired point, thus requiring less manganese. *c.* Composition: iron of analyses of absolute regularity can be made; low phosphorus irons can be melted and easily cast; large proportions of steel can be used to give irons of widely varying characteristics. *d.* Economic: pig iron can be eliminated from the charges; scrap, such as borings, may be used with insignificant losses; foundry is independent of quality of coke supply; large pieces impossible of cupola melting may be charged.

**Costs.**—N. L. Turner (*Trans.* A.F.A., vol. xxxviii, p. 749, 1930) reports on the operation of a 3-ton, direct-arc furnace utilizing cast-iron borings, shop scrap and steel clippings with alloy additions, with a cost of iron at the spout of $23.71 per ton. Of this cost $11.32 covers metal and alloys; $1.14 labor; $5.25 power; $2.09 electrodes; $0.60 refractories; $1.13 melting loss. The remainder covers water, laboratory investment and depreciation. The metal is poured into stove plate of comparatively high C and Si content.

**Reclamation of Sheet Scrap Material** by a plant producing gray iron castings is a factor recommending the electric furnace. Edwin Bremer (*The Fdry.*, Nov. 15, 1930) describes the application of a rocking type, indirect-arc furnace, utilizing soft steel scrap of 3 to 4% Si. A high grade gray iron is made by treating the charge in the furnace with petroleum coke to increase C. A wide range of C and Si analyses is possible. Tests of the castings range from 23,000 to 40,000 lb. per sq. in. tensile, with Brinell from 163 to 202.

**High Quality Electric Furnace Castings.**—An example of the use of the electric furnace induced by demand for higher quality is provided by H. E. Bromer (*Trans.* A.F.A., vol. xxxix, p. 585, 1931). A 3-ton direct-arc furnace operates in duplex or independently, as the character of castings demand and operating conditions permit. In duplexing, the electric furnace charge is a bath of 15,000 to 18,000 lb. from periodic taps of the cupola. Approximately 1 1/2 hr. after the cupola is in blast, the electric furnace is tapped; up to 1200 lb. of metal are taken out every 5 to 6 minutes. Cupola metal is charged intermittently into the electric furnace. Operation is continuous, the electric furnace acting as a refining and superheating medium for the cupola.

For restricted operation, the electric furnace melts cold stock, comprising cast-iron borings, cast scrap and steel scrap. Power consumption for continuous cold melting averages 540–550 kw-hr. per ton; graphite electrode consumption slightly over 12 lb. per ton. Refractory life, while duplexing, was a total of 800 tons of metal melted before roof was replaced; 1204 tons were melted with the side walls replaced at the same time. Average figures are 600 tons for a roof and 1000 tons for side wall linings.

The main factor in the operation of this furnace was the high quality and uniformity of automotive castings attained. Average tensile strength of unalloyed iron was approximately 40,000 lb. per sq. in. on test specimens machined from 1 1/4 in. round bars. The average for low Ni–Cr iron was practically the same, while high Ni-Cr iron showed average tensile strength of 45,000 lb. per sq. in.; a molybdenum iron averaged 50,000 lb. per sq. in. With electric furnace iron, variation in Brinell hardness taken at various points on cylinder bore, on borings and on flanges varied only approximately 15 points. Impact tests showed the electric furnace iron to have approximately twice the impact strength of cupola iron of similar analysis made for a similar class of castings.

**THE ACID PROCESS** of melting gray iron in the electric furnace offers only the qualities of superheating. It is unable to desulphurize or deoxidize by the use of slag baths, its power of eliminating bases being a function of its silicon. The regularity of composition may be considered as fair only, due to rapid changes of Si in the metal, especially at higher temperatures. It offers certain economic advantages of cheapness, and ability to use certain classes of scrap inapplicable to cupola melting. Its use is limited, and there seem to be no points which should warrant any great increase. For regular operations on iron, Barton does not recommend the acid furnace, the basic process offering far greater advantages.

## 4. CONDITIONS AFFECTING QUALITY OF CASTINGS

**HEAT TREATMENT OF GRAY IRON.**—(J. W. Bolton, *The Foundry*, March, 1933.) A knowledge of graphitization and reactions of metalloids is prerequisite to intelligent heat treatment of cast iron. For maximum softness the temperature of pearlitic cast iron must be carried above the transformation range (1325° to 1340° F.), and cooling through and to, say, 200° F. below this range must be slow. Such treatment produces a material composed almost wholly of ferrite and graphite. To harden, it is necessary to heat considerably above the transformation point and to quench rapidly.

When cooled rapidly or chilled, the cementite-austenite structure is shown. Irons of this structure, called white irons, are intensely hard as cast. Mottled irons lie between white and gray irons. They contain both cementite and graphite, as well as austenite (or at room temperatures) martensite or pearlite. To soften such irons usually requires an annealing temperature higher than is needed for straight gray iron.

Black graphite flakes, which are soft and non-tenacious, weaken the matrix of the iron. They are one of the structural features that distinguish gray iron from steel. The amount, size and distribution of graphite flakes are major controlling factors determining many physical properties of gray cast irons. Control of graphite is important.

A pearlitic matrix is characteristic of most gray irons. Alternate lamellæ of cementite and ferrite, and the general resemblance of this structure to pearlite in steel may be noted in micrographs. The ferrite is similar to that in pure iron and low-carbon steel. Martensitic structure is obtained by proper quenching.

The foregoing statement of the principles underlying heat treatment of cast iron has been confined largely to rather pure iron-carbon alloys. Commercial cast irons are alloys of Fe, C and Si, plus Mn, P, and S, and in many cases *alloying* elements, as Ni, Cr, Mo, etc. In some cases the proper use of alloys facilitates heat treatment by quenching. Lowering of critical range, some stabilization of carbides, and lowering of critical quenching speeds are definite advantages that frequently result in easier manipulation, less drastic treatment, and probability of greater toughness in the finished product. On the other hand, carbide stabilizers as chromium and molybdenum, may make annealing difficult. The influence of the various elements on heat treatment is summarized below:

**Silicon** in cast iron may range from below 1% to possibly 3%. It is a graphitizing element. For a given section the amount of graphitization, *i.e.*, the relative percentages of graphite and combined carbon, can be controlled by adjustment of Si. Si also lowers C required for the eutectic, changes the pearlitic transformation range, and in other ways modifies the properties of the iron.

**Sulphur, Phosphorus,** and **Manganese** may be neglected, as regards heat treatment, except that in sufficient amounts, Mn depresses the critical range.

**Nickel,** in amounts up to slightly over 1%, has a mild graphitizing action and does not interfere with annealing. It lowers the critical range and decreases critical quenching speed. Lowering of the critical point by nickel is so marked that at from 4 to 8% Ni, cast iron will be largely martensitic, and at 18 or 20% Ni it is largely austenitic.

**Chromium** is a carbide stabilizer, tending to inhibit graphitization. It is said to raise transformation range, but to decrease critical quenching speed.

**Molybdenum** is a carbide stabilizer, lowers transformation range and decreases critical quenching speed.

**General Precautions** in heat treating of cast iron must be observed, as follows: 1. For good results, the original castings must be close grained material, of say 2.70 to 3.20% C, with a largely pearlitic matrix, *i.e.*, combined carbon, 0.50 to 0.70. Coarse, open-grain irons do not heat treat well. High-carbon, high-silicon irons should be avoided. 2. All cast irons are relatively non-ductile. Cold castings, particularly complicated shapes, placed in a hot furnace are likely to crack. Preheating to 500° or 600° F. is advisable. 3. Drastic quenching treatments are to be avoided. Quenched castings must be drawn to maintain toughness. 4. In quenching treatments, long holding at high temperature will cause trouble.

The principal commercial heat treatments applied to gray cast irons are: 1. Normalizing, aging, or low temperature annealing, to relieve internal stresses. 2. Dead or softening annealing to increase machineability. 3. Quenching and drawing to increase hardness and wear resistance. 4. Nitriding. For chilled or mottled irons a fifth type of annealing, *Chilled iron anneal*, is used.

**Low Temperature Anneal.**—Many castings, particularly those of uneven section, have internal stresses, which, if not relieved, cause warping and, in extreme cases, cracking. In precision work, earlier practice allowed the castings to age from a few days up to several months. This practice took time and was not always effective. Other castings are used at fairly high temperatures and must maintain their dimensions under operating conditions.

Low temperature annealing or quick aging to remove internal stress comprises heating to about 900° F., holding from 1 to 8 or 10 hr., and cooling slowly to below 450° F., in the furnace. The metal is made plastic enough to relieve internal stresses without entering the temperature range of graphitization. In most close-grain irons graphitization at 900° F. is slow, hardly appreciable over a few hours. With open-grain irons it may be desirable to drop the temperature to 800° F. This type of quick aging causes little change in the physical properties of the metal. To obtain a slight softening as well as stress relief the temperature may be raised to 1000° F., or even to 1150° F., depending on time and on amount of softening desired. Softening usually results in a slight loss of tensile strength.

**Softening Anneal.**—Dead or soft annealing of gray iron requires heating above transformation point, holding long enough to allow complete penetration of temperature, and cooling slowly to well below the pearlitic transformation. Annealing temperature usually is about 1450° F., followed by slow cooling to 900° F. or under. In the presence of certain alloys it may be necessary to increase the temperature, time, or both. Dead annealed cast iron is soft, usually about 120–140 Brinell, depending on total carbon and Si. In presence of alloys, hardness may be somewhat higher. Dead annealed cast iron is easily machined. Its strength usually is 10 to 25% lower than that of the original cast metal. Too high annealing temperature, or too long annealing time, or both, may cause graphite grain growth, with consequent detriment to properties of the metal.

**QUENCHING TREATMENTS.**—Completely chilled irons usually require higher annealing temperatures (1600° to 1750° F.) than gray irons. Several hours may be required to fully break up the cementite. Castings with small chilled spots in fins, small sections, etc., often cause trouble. To eliminate hard spots it is necessary to anneal at 1450° F., and sometimes at higher temperature. The correct temperature and time is a matter of experiment. An anneal with slow cooling will break down the desirable pearlitic matrix, resulting in softness and loss of strength in the body of the casting. No definite rules can be given for retention of maximum physical properties, but it may be necessary to follow the high temperature period with a quench and draw. If a mottled iron or a white iron be properly treated, it is possible to produce so-called semi-malleable. For semi-malleable, the maximum annealing temperature may be 1700–1750° F. After soaking 3 hours, the castings are cooled in the furnace to about 1400° F. They then may be air quenched, oil quenched and drawn, or cooled in the furnace, depending on result desired.

If a low-carbon, low-silicon iron (2.00–2.50% C, 1.00–1.50% Si) is used, it is possible to produce material well over 60,000 lb. per sq. in. tensile strength, or even over 75,000 lb. per sq. in., and also to obtain some ductility. Brinell hardness may be from 150 up, depending on the iron and on the quench used.

The high temperature anneal, and in mottled iron, the presence of graphite flakes as cast, leads to carbon formations intermediate between the flakes of gray iron and the nodules of true malleable.

Quenching treatments usually comprise heating to about 1500° F., holding just long enough for complete heat penetration, and quenching, usually in oil. Air quenching usually is so slow that instead of hardening, actual softening of metal may occur. Water quenching often is too drastic and the metal may be made excessively brittle. The danger in quenching cast iron is graphitization. If the treatment produces more graphite and less combined carbon, the desired hardness will not be attained. The presence of an increased amount of graphite often both embrittles and weakens the metal.

**Precautions and Procedure** that will give best results in quenching treatments are: Metal must be close-grained, reasonably low in C and Si. Sometimes the addition of so-called carbide-forming elements, Cr and Mo, is advantageous, tending to prevent excessive graphitization during heating and quenching. These, with Ni, are said to lower critical quenching speed. As a rule, strength and particularly toughness of gray cast iron is impaired by quenching. The original strength and toughness is regained or exceeded by proper drawing or tempering. Drawing above 350° F. decreases hardness and increases strength and toughness. Both usually are regained fully by drawing at 500 to 600° F., and commonly are increased considerably by drawing to as high as 1000° F. Strength and shock resistance may be increased as much as 50%. Hardness is greatest in the as-quenched condition. It decreases progressively as drawing temperature increases beyond 350 or 400° F., and is not fully lost until drawing temperatures are quite high. Thus, an iron 2.70% C, 200 Brinell, in the as-cast condition approaches 500 Brinell when quenched in oil from 1500° F. When drawn at 1050° F. for 2 hours, it still retains about 275 Brinell.

The usual range of hardness obtained by treatment of machinable castings is 400 to 500 Brinell. Properly quenched and drawn cast iron has a fine, almost silky, grain, much lighter in color than the fracture of regular gray iron. It has excellent wear resistance, both in metal-to-metal wear and under mildly abrasive conditions. Heat-treated cast irons retain the resistance to galling and seizing characteristic of regular gray irons.

**NITRIDING** sometimes called ammonia hardening, has been applied successfully to special cast irons. The process comprises heating for 2 1/2 to 4 days at 900 to 1000° F. in an atmosphere of anhydrous ammonia. The cast iron commonly used contains about 1.5% Cr, 1.5% Al, and total C of 2.50 to 2.75%. It is claimed that if V is added Al can be eliminated. Cast irons of this Cr and C content are quite likely to air harden. The usual practice is to anneal before machining. The castings then are hardened, drawn, and nitrided. Surface hardness equivalent of 800 to over 900 Brinell is claimed.

**HARDENED CAM CASTINGS** should be made of so-called semi-steel, containing approximately 15% steel. Si should be adjusted to section of casting. Assume the cam to be a plain casting approximately 1 in. thick and to be machined. The recommended composition is: T.C., 3.2; Si, 1.0–1.2; Mn, 0.5–0.7; P, 0.15–0.2; S, under 0.08. To harden, the cam should be warmed throughout to 300° F., rapidly heated to 1530–1550° F., and held at this temperature for 30 min., or until uniformly heated. It is then carefully lowered into water at 80–90° F., cooled to about 300° F., and then immediately placed in hot oil at 400° F. It is held at 400° for 30 min. to 1 hr., to remove strains and avoid cracking. This treatment gives a Brinell hardness of 500. If greater hardness is required, the cam is quenched in clear cold water, and while still steaming, is placed in oil at 350° F., for 30 min. This treatment tends to cause warping, and unless skillfully done will cause cracking. A Brinell hardness of 400 to 450 may be obtained by quenching in cold oil, withdrawing from oil when the casting is at about 300° F. This treatment will avoid warping.

## 5. STEEL CASTINGS

**DEFINITION.**—Cast steel is an alloy of iron C, Mn, Si, and other elements, which is malleable, at least in some one range of temperature, and is cast into an initially malleable mass, or is capable of hardening greatly by sudden cooling, or is both so cast and so capable of hardening. Castings are solidified from the molten state in molds.

**USES.**—Cast steel products range in section thickness from 1/4 in. to 4 ft., and weigh from a few ounces to many tons. The extreme size recorded (1937) is over 200 tons. Within the extremes of size and weight an infinite variety of shape and dimension is possible in cast steel. A classic example is a unit casting embodying a locomotive and tender frame complete with integral cylinders, cast in one piece with an overall length of 65 ft.

**CLASSES.**—Steel castings mainly may be classified as carbon cast steels and alloy cast steels. Plain carbon cast steels (C. H. Lorig and C. E. Williams, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 114, 1932) contain 0.50 to 0.90% Mn, 0.20 to 0.70% Si, and 0.05%, or less, of S and P. Carbon cast steels are divided into three grades: regular, from 0.22 to 0.35% C; low carbon, less than 0.22% C; high carbon, containing more than 0.35% C.

**SPECIAL ALLOY STEEL CASTINGS.**—(R. A. Bull. A.S.M.E., 1932.) Many combinations of alloys commonly used are possible by introducing as many as eight different elements. Besides the range of properties attainable through these elements, a variety of heat treatments, varied with each of the different alloy additions is possible. For example, satisfactory results from steel castings may be attained when the metal contains elements as low and as high as the following: C, 0.05–3.00%; Mn, 0.05–4.00%, and 10.0–15.0%; Si, 0.08–3.00%; Ni, 0.15–35.00%; Cr, 0.15–30.00%. The interval between the low and the higher range of manganese represents unexplored territory. The lower range materials are classified as pearlitic or sorbitic and those in the higher range as austenitic manganese steel.

The proportions of Ni and Cr represent variable quantities added either for improving structural properties, or for specific avoidance of corrosion under extreme conditions of temperature and corrosive elements. The complexity of alloy cast-steel properties and applications is indicated by the fact that 15 different heat treatments may be applied in one day to the output of a foundry making numerous alloy steels. These vary from full annealing, to normalizing, drawing, tempering, quenching in different mediums, and do not include such specialized treatment as carburizing or nitriding. In alloy cast-steel production the proper application of heat treatment is a vital consideration.

**MELTING MEDIUMS.**—Cast steel may be produced in crucibles, converters, open-hearth and electric furnaces. Since 1925, but small quantities of steel castings have been made in the crucible and converter. The choice of the open-hearth or electric furnace for plain carbon steel depends on quantities. Satisfactory special alloy castings can be made from the open-hearth, but the electric furnace, due to flexibility and small size requirements, generally is used for alloy-steel castings.

**The Open-Hearth Furnace** for steel castings differs mainly in size from that used for structural steel ingots. The metal melts in a chamber into which flame from gas, oil, or in some cases, coal, is applied directly to the bath. Metal is charged through along one side of the furnace. Metal is tapped through a centrally located outlet on the opposite side. The roof arches transversely to the length of the furnace. Roof and sides are of highly refractory brick to withstand temperatures of 3000° F. or more.

Steel foundry open-hearth furnaces range from 10 to 40 tons capacity. A few are in operation as low as 5 tons, and as large as 60 tons. In acid furnaces, the lining is silica brick, with a 1- to 2-ft. bottom of silica sand under the bath. Basic linings are of magnesia brick, with a bottom of magnesite.

In acid melting, less than half the charge is pig iron; the remainder is steel. Oxidation of carbon in the charge is due to the $SiO_2$ and $Fe_2 O_3$ of iron ore which is added to increase this action. Oxidation of manganese and silicon raises the temperature. Sulphur and phosphorus are not reduced, and the use of scrap is limited to grades which do not have these elements in excess. In operation, carbon is removed below the desired final product and the total raised by additions of ferromanganese and ferrosilicon.

Basic melting is used for products in which high percentages of P and S would be deleterious. Additions of limestone with a large percentage of $CaCO_3$ provide the chemical reaction with Mn on P and S. In both basic and acid practice, ladle additions will accurately control the chemical elements.

**The Electric Furnace** uses both acid and basic linings. The character of available scrap influences the choice of lining. The acid process, due to speed and economy, is used whenever possible. The basic lining is essential in production of manganese steel and, due to its deoxidizing features, is of value in recovering costly alloys, as Cr, Mn, V, Mo and Ni. A melt may be held in the basic furnace for a longer period without change in regularity of composition.

The acid process is favored for plain carbon steel castings. Higher power input is possible with shorter melting period than in the basic furnace. Considerations of lining cost, lower electrode consumption and other factors are important.

The arc type furnace is commonly used. For special alloys, and also to make heat- and corrosion-resistant alloy castings, electric induction furnaces up to 1000 lb. capacity are used.

## Carbon Steel Castings

**LOW-CARBON STEEL CASTINGS** (C. H. Lorig and C. E. Williams, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 114, 1932), according to the previous classification, do not respond to heat treatment except in refinement of grain structure and improvement in ductility. They range in tensile strength from 42,000 to 70,000 lb. per sq. in., with an elongation from 15 to 30%. The carbon content makes this grade of casting particularly applicable to electrical equipment. Such steel also is used for annealing containers, ladles and other heavy parts subjected to high temperatures. Higher carbon grades are more suitable for structural purposes.

**REGULAR-CARBON CAST STEEL.**—Plain carbon steels, 0.22 to 0.35% C, comprise the largest portion of steel castings made. Tensile and other strength properties of plain carbon castings may be varied over a wide range by proper heat treatment. Tensile strength ranges from 60,000 to 80,000 lb. per sq. in.; elongation from 20 to 30%; reduction of area from 20 to 40%. To develop mechanical properties, castings should be heat treated by quenching and drawing if design permits. After such treatment, regular carbon steel castings often equal those of some special annealed alloy steel. Endurance properties developed by Prof. H. F. Moore show a ratio from 40 to 50%. The magnetic properties of regular carbon steel permit its use in moving parts of electrical equipment. Tensile strength of regular carbon cast steel increases to a maximum at 400° to 600° F., while elastic limit and yield point decrease. A limited number of tests show only slight embrittlement at low temperatures. This grade of steel castings is used widely in railroad service, machine building, automotive industry, decorative and ornamental work, and, to a limited extent, in chemical and refinery service at high temperatures and pressures.

**HIGH-CARBON CAST STEEL**, 0.40 to 0.90% C, develop superior properties when annealed or quenched and drawn. Tensile strength varies from 70,000 to 120,000 lb. per sq. in.; elongation from 3 to 20%; reduction in area from 8 to 40%. High-carbon steel castings, properly heat treated, may be used where high strength and toughness are desired, as in roll and die tool manufacture, machine tools, automotive, contractors' and railway equipment.

**MEDIUM MANGANESE CAST STEEL.** (Lorig and Williams, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 143, 1932.)—Additions of small percentages of Mn do not constitute alloy steel according to foundrymen's interpretation of such terms. John Howe Hall describes these lower Mn steels as divided into pearlitic Mn, intermediate Mn, and high Mn steel. Hadfield, or regular, Mn steel contains 12% or more Mn and more than 1% C. Steels of low Mn are used where high strength with great toughness is essential. The tensile strength and elastic limit increase slowly with Mn, while elongation and reduction of area decrease. According to Zuege, cast steels with 1.35 to 1.55% Mn and 0.27 to 0.37% C, have a tensile strength, 95,000 lb. per sq. in.; elongation, 20% in 2 in.; reduction in area, 40%, when normalized at 1650° F. The same steel normalized at 1650°, drawn at 1250° F., shows 85,000 lb. per sq. in. tensile strength; 24% elongation; 45% reduction of area. A slight decrease in both C and Mn increases toughness but lowers strength.

The customary range of analysis of medium Mn cast steel that can be treated by quenching in oil or water and drawing, or normalizing and drawing, is: 0.20 to 0.40% C; 0.20 to 0.50% Si, and 1.10 to 2.00% Mn. When properly heat treated, these grades are ductile, with high tensile properties and resistance to abrasion.

Addition of Mn to Ni steels, using 1.00 to 1.75% Ni and 1.00 to 1.50% Mn, with C from 0.30 to 0.40%, gave tensile strength, 80,000–90,000 lb. per sq. in.; elongation, 18–23%; reduction of area, 28–33%. These values are increased by proper heat treatment.

**NICKEL CAST STEELS** (Lorig and Williams, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 161, 1932).—Cast steel with C, 0.10 to 0.60%, and Ni, 0.5 to 5.0% are characterized by high strength and elastic limit, ductility and resistance to impact. Tensile strength may be 75,000 to 110,000 lb. per sq. in.; elongation, 7 to 26%; reduction of area, 5 to 50%, depending on heat treatment followed.

To develop high impact and fatigue resistance in steel containing 0.20% C (max.), 2.0% Ni (min.), 0.60 to 0.90% Mn, 0.15% Si (min.), the following heat treatment is adopted: Heat 2 hr. per 1-in. thickness at 1750° F., with air cooling; normalizing 1 1/2 hr., per 1-in. thickness at 1500–1550° F.; draw at 1100 to 1250° F., with furnace cooling. Such treatment gives tensile strength 75,000–85,000 lb. per sq. in.; elongation, 23 to 32%; reduction of area, 45 to 65%.

Low-carbon nickel steels are used in many locomotive castings to resist impact and fatigue, and in mining and excavating machinery, marine and rolling mill service and for bridge castings.

**NICKEL-MOLYBDENUM CAST STEEL.**—The hardening tendencies of Ni steel are increased by small quantities of Mo, (Lorig and Williams, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 163, 1932). Such steels require normalizing and tempering to develop best impact and ductility values. Castings with 0.25 to 0.30% C, 0.60 to 0.80% Mn, 1.25 to 1.50% Ni, 0.30 to 0.35% Mo, show tensile strength 85,000 to 90,000 lb. per sq. in.; elongation, 22 to 25%; reduction of area, 45 to 50%. Ni-Mo steel castings are recommended for service at high temperatures.

**MOLYBDENUM CAST STEEL.**—Molybdenum increases the tensile strength of heat-treated steels containing more than 0.30% C. Its high yield point at elevated temperatures is characteristic. Molybdenum-steel castings are used in rolls, turbine housings, and where high tensile strength is required. Due to its air-hardening characteristic molybdenum gives good results in large castings which cannot be quenched in liquid, but which must have hardness to resist abrasion.

**MANGANESE-MOLYBDENUM CAST STEEL.**—Heat-treated Mn-Mo cast steels with 0.20 to 0.40% Mo, 1.0 to 1.75% Mn, 0.30 to 0.40% C, range from 90,000 to 160,000 lb. per sq. in. tensile strength; elongation, 14 to 30%; reduction of area, 40 to 60%. The Mn-Mo steels have a marked tendency to air harden. Unless properly heat treated, ductility and toughness are decreased.

**CHROMIUM-NICKEL CAST STEELS** (Lorig and Williams, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 177, 1932), with 0.28 to 0.60% C, 0.60 to 0.87% Mn, 0.66 to 1.60% Cr, and 1.00 to 2.85% Ni, with various heat treatments, range from 100,000 to 202,000 lb. per sq. in. tensile strength; elongation, 6 to 23%; reduction of area, 10 to 52%. The distinct advantage of this class of alloy steel castings is the improvement in properties made possible by proper heat treatment. Zuege reports normalized and drawn Cr-Ni steel at slightly higher yield points and tensile strengths than normalized Mn carbon steel, but with lower ductility. Rapid quenching emphasizes the hardening and strengthening effects of Cr and Ni. Cr-Ni steel may be hardened to a higher Brinell than Mn carbon steel. Fatigue resistance is high, and Cr-Ni cast steels give satisfactory strength at high temperatures. They have good mechanical properties which may be developed by simple annealing. Proper heat treatment, however, improves these properties. Medium carbon Cr-Ni steel gives excellent service under wear. Castings are adapted to use as grousers or tracks for caterpillar treads, chain belt parts, rolling mill rolls, oil refinery and other machinery. The resistance of these steels at high temperature and their corrosion-resistant properties adapt them to use in chemical and refining industries. The addition of Mo to Cr-Ni steel does not affect its mechanical properties, but improves machinability and develops air-hardening characteristics.

**PEARLITIC STEELS** (Fred Grotts, *Proc.* A.S.T.M., vol xxxii, Pt. 2, p. 197, 1932).—Cast medium pearlitic steels, when normalized, consist principally of pearlite. They contain a small amount of special material as Ni or Si dissolved in the ferrite, which may be combined with carbon as cementite, as a double carbide of iron and of the special element, or may be partially in solution in ferrite with Mn, Cr, W, V, or Mo. Pearlitic steels have higher yield point and ductility and greater hardness than plain carbon steel. Pearlitic carbon steels containing 0.40 to 0.50% C are used in wheels, rails, cutting tools, gears, tractor shoes, dies, etc.

Adding small quantities of Mo to pearlitic steels increases yield point and toughness. Average properties of 0.30–0.40% C pearlitic steel with 0.30–0.40% Mo, fully annealed, normalized and drawn, are: Tensile strength, 83,200 lb. per sq. in.; yield point, 58,100 lb. per sq. in.; elongation, 21%; reduction of area, 33%.

Adding 0.10 to 0.20% V gives average properties of: Tensile strength, 91,500 lb. per sq. in.; yield point, 58,500 lb. per sq. in.; elongation in 2 in., 28.5%; reduction of area, 50.5%.

From 0.75 to 1.00% Cr added to the same range of carbon in pearlitic steel results in: Average tensile strength, 95,000 lb. per sq. in.; yield point, 55,000 lb. per sq. in.; elongation in 2 in., 18%; reduction of area, 31%. Additions of Cr increase hardness and are recommended for castings subject to abrasion, to water corrosion and high operating temperatures.

While Mn is added as a deoxidizer, it also alloys in pearlitic steels. Medium Mn pearlitic steels contain 0.30 to 0.40% C, and range from 1.05 to 3.00% Mn. Tensile strength range, 80,000 to 117,000 lb. per sq. in.; yield point, 40,000 to 65,000 lb. per sq. in.; elongation in 2 in., 18 to 32%; reduction of area, 31 to 60%. After drawing, medium Mn steels should be air cooled and sometimes quenched. Pearlitic Mn steels have low yield point and impact values in full annealed condition. These properties are improved by the addition of other elements such as V, Mo, Zr, etc.

**CORROSION-RESISTANT STEELS** (Jerome Strauss, *Proc.* A.S.T.M., vol xxxii, Pt. 2, p. 213, 1932) are available to resist attack of polluted atmosphere, natural water, sea water, crude petroleum and other chemicals at atmospheric and elevated temperatures, air and furnace gases and molten salt and metals. They are alloys of iron and chromium which with the introduction of C, Si, S, Mo, W, Ti, V, and Ni acquire specific properties for designated purposes. For a given content of C, corrosion resistance increases with Cr. Finely divided sulphides sometimes are added to improve machinability, while Ni up to 2% reduces grain size, improves shock resistance and slightly increases corrosion resistance. Si up to 2% improves casting qualities and adds to corrosion resistance.

Cr steels resist well oxidizing acids, and gases, and sulphur-rich gases, at high temperatures, but are unsuited to high stresses at high temperatures, or to numerous and sudden variations in stress or temperature. High Ni-Fe-Cr alloys are unsuited to sulphur-rich gases at high temperature, but may be satisfactory at moderate temperatures with reduced sulphur compounds. They are valuable for resisting great stress at high temperature and sudden changes in stress and temperature. Some of these alloys are almost immune to oxidizing acids and quite resistant to $H_2SO_4$, while others show oxidation resistance to a greater degree than plain Cr steels. Practical tests, under severe conditions, indicated prior to 1930 that a medium-carbon alloy containing Si, with about equal proportions of Fe, Cr, and Ni was the most satisfactory as regards strength and stability.

The production of special alloy, corrosion-resisting castings is highly specialized. The user should consult foundries qualified to advise as to analyses, methods of manufacture and treatment.

**AUSTENITIC OR HADFIELD'S MANGANESE-STEEL** (John Howe Hall, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 238, 1932) contains 10.00 to 14.00% Mn. It is toughened and softened by quenching in water from high temperature, is non-magnetic and becomes intensely hard if cold worked. It is made either in the basic electric furnace, or by mixing in a ladle 80% ferro-manganese and low-carbon steel from the open-hearth or Bessemer converter. As cast, Mn steel castings are brittle. Heating to 1900–1940° F. and quenching in cold water makes them strong and tough. Tensile strength ranges from 80,000 to 110,000 lb. per sq. in.; elongation in 2 in., 15 to 35%; reduction of area, 15 to 35%. As cast, Brinell hardness is slightly over 200; this is reduced slightly by heat treatment, and increased by cold working. Local pounding may raise hardness to over 500 Brinell. In services subject to heavy wear, increase in hardness is a valuable property, and recommends austenitic Mn steel for frogs, crossings, lining plates of grinding mills, steam shovel teeth and lips, delivery buckets, crane wheels, gears, pinions, racks, etc. The toughness from cold working makes Mn steel castings wear-resistant in a ratio of 20 to 1 as compared with many other materials. Hardening under severe cold work renders machining difficult, and practically all machining operations are performed by grinding.

**DESIGN FOR STEEL CASTINGS** (F. A. Lorenz, *Proc.* A.S.T.M., vol xxxii, Pt. 2, p. 58, 1932).—Steel castings may be adapted to a wide range of work by proper application of fundamental design principles. The range increases as the designer works with the foundry. The designer should initiate co-operation, and consult with the foundry to effect adjustments that will reduce cost and increase life and economy of the casting.

Steel castings are poured at temperatures above 2800° F. The contact of metal with the chilled surface of the mold instantaneously forms a skin of solidified metal. Steel as it solidifies forms crystals, which formation involves contraction from all directions toward the center. The center, solidifying last, continues to supply the demands for metal and a shrinkage cavity may result. Adequate feeding and the location of risers to supply heavy sections with surplus metal is advisable.

The point of junction between light and heavy sections and the progressive freezing of heavy sections requires provision for the surrounding sand to carry away heat. Ample radiating surface and suitable radii between adjoining sections are recommended. A uniform rate of cooling will avoid cooling strains. Deep pockets, with sharp interior corners, produced by bodies of sand of insufficient area to conduct away heat of the surrounding metal should be avoided. Adequate fillets or the coring of heavy sections near adjoining members of a casting avoids concentration of cooling stresses. Abrupt changes in cross section invariably are to be avoided.

The linear shrinkage of steel as it cools causes an actual movement of the casting in the sand. Projections, flanges, bosses or brackets which prevent or impede this contraction may, by a proper use of cores, special sand mixtures and other precautions, be rendered harmless in their effect on the finished casting.

**FACTORS IN STEEL CASTING DESIGN** (Steel Founders' Society of America).—1. **Shrinkage Allowance for Patterns,** varies according to design of casting and temperature of metal poured, as well as the restraint of the mold caused by projecting lugs, cores, etc. Owing to this variation in shrinkage of steel when cooling in the mold, patterns may have different allowances for different dimensions. Shrinkage of open-hearth cast steel varies from $^1/_{16}$ in. to $^1/_4$ in. per lineal ft.; the average is taken as $^3/_{16}$ in. Shrinkage of both carbon and alloy electric cast steels, except high Mn, varies from $^1/_8$ in. to $^3/_8$ in. per lineal ft. High Mn steel shrinks from $^3/_{16}$ in. to $^7/_{16}$ in. per lineal ft.

2. **Uniformity of Section.**—Thickness of section should be as nearly uniform as possible throughout. When possible, changes in section should be made with a gradual taper, rather than abruptly; if this is impracticable, ribs should be used intelligently. Uneven shrinkage between heavy and light sections causes the metal to tear, results in shrinkage cavities, or gives rise to directional or dendritic crystals in the metal that do not promote high strength. A minimum of bosses, of the minimum thickness, should be used.

3. **Wall Thickness** as low as $^1/_4$ in. may be poured in the steel foundry. In special cases wall sections of $^1/_8$ in. and $^3/_{16}$ in. are possible, depending on design of the casting. The best practice is to use a wall thickness that can be poured without danger of misruns and give the necessary strength or weight. In general, sectional thickness should be proportional to the area or diameters to be run.

4. **Fillets** should be used on all inside corners where walls intersect. Unfilleted corners are weak.

5. **Avoidance of Sharp Corners.**—Fillets also obviate sharp corners in the mold, from which sand may be washed by molten steel. Size of fillets depends on shape and thickness of wall section and size of casting.

## Heat Treatment of Steel Castings

The following is abstracted from a paper by A. W. Lorenz, *Proc.* A.S.T.M., vol. xxxii, Pt. 2, p. 253, 1932:

**FULL ANNEALING** comprises heating iron-base alloys above the critical temperature range, holding above that range for a proper period, and slowly cooling through the range. Annealing temperature is usually about 100° F. above the upper limit of the critical temperature range; the time of holding is usually at least 1 hr. for each 1 in. of the heaviest section being treated. The treated castings ordinarily cool slowly in the furnace. They may, however, be removed and cooled in a medium which will prolong the cooling time as compared to unrestricted cooling in air. This treatment removes strains, improves machinability and ductility. In some cases it may be the only practical treatment, due to size of casting, maintenance of dimensions, or other limitations.

**NORMALIZING** is the heating of iron-base alloys to approximately 100° F. above the critical temperature range followed by cooling to below that range in still air at room temperature. Hypereutectoid steels rarely are normalized because of the coarsening of grain and the tendency to crystallize cementite at grain boundaries or in needles. However, it sometimes is necessary to normalize these steels by heating them above the $Ac_{cm}$ line of the Fe-C diagram. This raises yield point and tensile strength as compared with full annealing, particularly on alloy steels. In high-carbon and alloy castings, strains and brittleness usually will be induced, unless a tempering or drawing treatment follows. Castings both normalized and tempered have better ductility, higher yield point and tensile strength, and considerably better impact and fatigue resistance than full-annealed castings.

**TEMPERING** (also termed Drawing) is reheating iron-base alloys, after hardening, to some temperature below the critical temperature range, followed by any desired rate of cooling. Although tempering and drawing are practically synonymous as used commercially, tempering is preferred. This, *i.e.*, hardening and then reheating, is an illogical and confusing practice in the present (1937) state of heat treating and should be discouraged. Tempering increases ductility, removes strains, softens the metal, and improves machinability. It properly is applied only after one of the before-mentioned treatments. Used after liquid quenching, it is a means of adjusting hardness and other working properties within a wide range to suit individual requirements.

**LIQUID QUENCHING** confers maximum strength and hardness. It almost always should be followed by tempering to remove quenching strains and brittleness, and to obtain suitable modification of properties. When followed by such a treatment, most carbon and alloy-steel castings will develop the maximum combination of strength and ductility.

A.S.T.M. recommended practice for heat treatment of carbon steel castings covers full annealing and normalizing temperatures as follows: C up to 0.40% inclusive, 1650° F.; C over 0.40 to 0.60% inclusive, 1560° F.; C over 0.60%, 1525° F.

A.S.T.M. standard recommended practices recognize two types of cooling, *viz.*, furnace cooling or full annealing, and air cooling and normalizing. The higher properties obtainable by air cooling are so marked that in the field of small and medium-size castings the practice is widely followed. With large castings, the rate of cooling is not always a matter of choice. The tendency toward rapid cooling favors the use of liquid quenching. The time of holding at heat is variously recommended from $^1/_2$ to 1 hr. or more per 1 in. of cross-section. In some cases the time for large castings may exceed 2 hr. per 1 in. of section.

The liquid quenching medium may be either oil or water, water being used more generally. Temperature of quenching medium is held above 100° F. If the casting has had no previous treatment it is usual to quench from approximately ordinary normalizing temperatures or slightly lower, say 1550°-1650° F. After quenching, castings should be removed from the bath and transferred to the tempering furnace while still warm. The chief use of liquid quenching has been the treatment of alloy castings. Liquid quenching of alloy steel castings develops ultimate strengths and yield points at least 50% higher than those typical of the metal in the normalized condition, and without appreciable loss of ductility.

Ductility of cast-steel may be improved by tempering and spheroidizing treatments ranging from 750° to 1400° F. Tempering practically always is necessary after liquid quenching, after normalizing and sometimes even after full annealing. Reheating within or slightly below the critical range is common with alloy steels and is increasing in the treatment of plain carbon steels.

**SPECIFICATIONS FOR CARBON STEEL CASTINGS FOR MISCELLANEOUS INDUSTRIAL USES** (Designation A27-36T).—Specifications cover ten grades of castings, of which eight grades must be physically tested. The grades not required to be physically tested are N-1 (need not be heat treated, except when the carbon content exceeds 0.30%), and N-2 (which must be annealed). Maximum chemical composition requirements are: Manganese, 1.00%; phosphorus, 0.05%; sulphur, 0.06%.

Steel used for castings shall conform to the minimum requirements as to tensile properties for the eight grades as indicated in Table 17.

**SPECIFICATIONS FOR ALLOY STEEL CASTINGS FOR STRUCTURAL PURPOSES** (A.S.T.M. Designation A148-36).—Specifications call for three classes of castings with a total of eight grades, with minimum requirements as shown in Table 18. In relation to the chemical composition of these classes of steel castings, specifications state that the prescription of definite composition types of steel to meet the various physical requirements outlined in the specifications is not to be recommended, except by mutual agreement between the purchaser and manufacturer, since there are many types of alloy cast steel which can be made to conform to the tensile requirements in any class listed in the specifications, merely by selection of heat treatment.

### Table 17.—Specifications for Carbon Steel Castings for Industrial Uses

(A.S.T.M. Designation A27–36T)

| Grade | Tensile Strength, Minimum, lb. per sq. in. | Yield Point, Minimum, lb. per sq. in. | Elongation in 2 in., Minimum, percent | Reduction of Area, Minimum, percent |
|---|---|---|---|---|
| A–1, Unannealed.............. | 60,000 | 30,000 | 22 | 30 |
| A–2, Normalized*............. | 60,000 | 30,000 | 26 | 38 |
| A–3, Full annealed............ | 60,000 | 30,000 | 24 | 35 |
| B, Normalized*............. | 70,000 | 38,000 | 24 | 36 |
| B–1, Full annealed............ | 66,000 | 33,000 | 22 | 33 |
| B–2, Full annealed............ | 70,000 | 35,000 | 20 | 30 |
| H, Normalized*............. | 80,000 | 43,000 | 17 | 25 |
| H–1, Full annealed............ | 80,000 | 40,000 | 17 | 25 |

* Any casting that may be ordered to meet the tensile requirements above listed for normalized material of Grades A–2, B, and H may, at the option of the manufacturer, be given a full annealing treatment instead of a normalizing treatment, provided that the above listed tensile requirements for normalized material are met.

### Table 18.—Specifications for Alloy Steel Castings for Structural Purposes

(A.S.T.M. Designation A146–36)

| | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elongation in 2 in., percent | Reduction of Area, percent |
|---|---|---|---|---|
| Class A: | | | | |
| Grade 1..................... | 75,000 | 40,000 | 24 | 35 |
| Grade 2..................... | 85,000 | 53,000 | 22 | 35 |
| Class B: | | | | |
| Grade 1..................... | 85,000 | 55,000 | 22 | 40 |
| Grade 2..................... | 90,000 | 60,000 | 22 | 45 |
| Grade 3..................... | 100,000 | 65,000 | 18 | 30 |
| Class C: | | | | |
| Grade 1..................... | 90,000 | 65,000 | 20 | 45 |
| Grade 2..................... | 120,000 | 100,000 | 14 | 35 |
| Grade 3..................... | 150,000 | 125,000 | 10 | 25 |

### Table 19.—Specifications for Alloy Steel Castings for Valves, Flanges, and Fittings for Service at Temperatures from 750° to 1100° F.

(A.S.T.M. A157–36)

| Class....... | Ferritic Steels | | | | | | Austenitic Steels | |
|---|---|---|---|---|---|---|---|---|
| Identification Symbol | C1 | C2 | C4 | C5 | C6 | C11 | C9 * | C10 |
| Grade...... | Carbon-Molybdenum | Manganese-Molybdenum | Nickel-Chromium-Molybdenum | 4 to 6% Chromium | 13% Chromium | Nickel-Chromium | 18% Chromium 8% Nickel | 20% Nickel 8% Chromium |
| C, %...... | 0.45 max. | 0.35 max. | 0.45 max. | 0.15–0.35 | 0.15 max. | 0.45 max. | 0.15 max. | 0.35 max. |
| Mn, %..... | 1.00 max. | 1.00–1.60 | 1.00 max. | 1.00 max. | 0.75 max. | 1.00 max. | 1.00 max. | 1.50 max. |
| P, %...... | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. | 0.05 max. |
| S, %...... | 0.06 max. | 0.06 max. | 0.06 max. | 0.06 max. | 0.05 max. | 0.06 max. | 0.05 max. | 0.05 max. |
| Si, %...... | 0.20 min. | 0.20 min. | 0.20 min. | 0.20 min. | 1.00 max. | 0.20 min. | 2.00 max. | 2.00 max. |
| Ni, %...... | ....... | ....... | 0.75–1.50 | ....... | 0.80 max. | 1.00–2.25 | 8.00 min. | 19.00–22.00 |
| Cr, %...... | ....... | ....... | 0.50–1.00 | 4.00–6.50 | 11.50–13.50 | 0.50–1.00 | 18.00 min. | 8.00–10.00 |
| Mo, %..... | 0.40 min. | 0.20 min. | 0.30–0.60 | 0.40–0.65‡ | 0.50 max. | 0.20–0.40† | As agreed | ....... |
| W, %...... | ....... | ....... | ....... | 0.80–1.25‡ | ....... | 0.40–0.80† | ....... | ....... |

* For the more severe general corrosive conditions, and when so specified, the carbon content shall not exceed 0.07%. The addition of such elements as molybdenum, tungsten, titanium, columbium, and vanadium for purposes of stabilization shall be a matter of agreement between manufacturer and purchaser. When the purchaser requires free-machining, or better non-seizing, non-galling properties, the composition may contain suitable combinations of selenium and phosphorus or molybdenum and sulphur as follows:

| | | | |
|---|---|---|---|
| Selenium and phosphorus | Selenium, 0.20–0.35%<br>Phosphorus, max., 0.17%; | Molybdenum and sulphur | Molybdenum, 0.40–0.80%<br>Sulphur, 0.20–0.40%. |

† Either molybdenum or tungsten may be used, if desired.

‡ Either molybdenum or tungsten shall be used.

**STANDARD SPECIFICATIONS FOR ALLOY STEEL CASTINGS FOR VALVES, FLANGES AND FITTINGS FOR SERVICE AT TEMPERATURES FROM 750° to 1100° F.** (A.S.T.M. Designation A157–36) covers eight grades of materials, including six ferritic steels and two austenitic steels designated C1, C2, etc., which are classified in accordance with their chemical and physical properties and high temperature characteristics. Steels C1, C2, C4, C5 and C11 shall be made by open-hearth, crucible or electric furnace process. Steels C6, C9 and C10 shall be made by the electric furnace or crucible process.

**Table 20.—Specifications for Alloy Steel Castings**
(A.S.T.M. Designation A157–36)

| Grade | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elongation in 2 in., percent | Reduction of Area, percent | Grade | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elongation in 2 in., percent | Reduction of Area, percent |
|---|---|---|---|---|---|---|---|---|---|
| C1 | 80,000 | 50,000 | 22 | 45 | C6 | 85,000 | 55,000 | 20 | 40 |
| C2 | 90,000 | 60,000 | 22 | 45 | C11 | 100,000 | 65,000 | 18 | 30 |
| C4 | 100,000 | 65,000 | 18 | 30 | C9 | 70,000 | 30,000 | 35 | 40 |
| C5 | 100,000 | 65,000 | 18 | 30 | C10 | 65,000 | 30,000 | 30 | 35 |

All castings shall receive heat treatment suitable to their design and chemical composition. Heat treatment shall be performed before machining, except in instances when reheat treating is necessary. Unless method of heat treatment is specified by the purchaser, thus becoming a matter of agreement with the manufacturer, the type of heat treatment shall be at the option of the manufacturer.

The ferritic steels, C1, C2, C4, C5, C6 and C11 may be fully annealed, or fully annealed and drawn, or normalized and drawn, or annealed and normalized and drawn, or normalized and annealed. The austenitic steels C9 and C10 shall receive a stabilizing treatment.

Each alloy shall conform to the chemical requirements, given in Table 19, or to other chemical requirements as specified in the purchase order. The castings shall conform to the minimum requirements as to tensile properties as shown in Table 20.

**STANDARD SPECIFICATIONS FOR AUSTENITIC-MANGANESE STEEL CASTINGS** (A.S.T.M. A128–33).—**Chemical Requirements,** C, 1.00 to 1.40%; Mn not under 10.00%; P not over 0.10%; S not over 0.05%.

**Bend Test,** only when specified in the order; specimen, 12 × 3/4 × 1/2-in., to bend cold without breaking around pin 1 in. diam., to angle of 150 deg.

**Tension Test,** not required.

## 6. PERMANENT MOLD CASTINGS

Permanent mold castings are produced in molds which are used repeatedly. The term "permanent" is a misnomer, inasmuch as no molds can be used indefinitely. The types of permanent mold castings operations may be divided into gravity type, pressure die-cast and centrifugally cast.

**GRAVITY TYPE.**—Gravity-type castings are made in molds or dies constructed of various alloys of iron or steel. A machine known as the Holly permanent mold machine employs a multiple series of identical molds placed on a turntable which moves in a circular, path about a center, progressing slowly or intermittently. The open mold is coated with carbon in a sooty deposit from a torch, moves forward and is automatically closed. Metal is poured into the mold as it passes the pouring station. The casting is ejected or pulled from the mold which remains open as it approaches the point of beginning. This type of casting machine is used to make small parts for automotive service, fittings and a variety of work ranging from a few ounces to about 10 lb. It requires high-production long-run work to achieve economy. Gravity-type permanent mold castings are produced in gray iron and aluminum. Some experimental work has been done on other non-ferrous metals.

**DIE-CASTINGS** of the pressure type are obtained by forcing liquid metal into a die or mold and maintaining pressure until the casting is solid. This type of work is confined to comparatively low melting point, non-ferrous metals such as the tin, lead or zinc white metal alloys and aluminum. The most common type of pressure die-casting work is that employed in producing either linotype or monotype lines or characters in printing plants.

In the foundry industry the most common type of die casting is produced from zinc base alloys. Die-casting machines consist essentially of three parts: The melting pot where metal is melted, usually by gas flame; the die with appropriate gate channels; and the pressure equipment for forcing metal through the gate and into the mold. Hydraulic or air-operated cylinders, with mechanical toggle equipment, supply the necessary pressure. Cores may be used and many types of dies are self-coring.

Dies are constructed of special steel, and while some non-ferrous metals produce corrosive or growth destruction, machines are in operation which show an exceedingly long die life. High production is possible with automatic machines and castings weighing from a few ounces to 15 pounds are produced. On heavier castings, it is necessary to cool the dies by water-jacket or some equally effective device.

Aluminum die-castings, notably aluminum pistons and radiator grills, are produced in large quantities. Inherent soundness and uniformity are claimed for aluminum castings made by this process. For data on die-casting alloys see pp. 4–60 to 4–62.

**CENTRIFUGAL CASTINGS.**—The earliest application of spinning molds to obtain round or cylindrical castings by the application of centrifugal force was in the cast-iron pipe industry. A major portion of gray iron pipe produced in the U. S. today is made either in permanent molds or in hollow molds lined with a thin layer of sand. The processes differ only in the mold characteristic. The de Lavaud process of producing centrifugal pipe involves the use of a thin, forged steel mold mounted horizontally in a water-cooled housing. The bell of pressure pipe is formed by a single sand core fitting in the end of the permanent mold.

In operation, a predetermined quantity of molten iron at high temperature is poured into a ladle at the spigot end of the machine, connected by a long channel or trough running to the bell end of the pipe. The mold is rotated at slow speed as the metal is spilled into the mold along its entire length. The speed of rotation then is increased and centrifugal force drives the metal outward so that it coats the entire inner surface of the mold. Rotation is maintained until initial cooling or solidification is accomplished. The pipe then is drawn from the mold by gripping the bell end and removing the pipe longitudinally from the fixed gripper. The pipe is annealed, coated and lined. Pipes are made varying in length up to 30 ft. and in diameters from 3 to 30 in.

In the sand spun process the steps are similar, with the exception that the mold is lined with a layer of sand rammed about a central pattern. This sand is surface dried before the mold is placed in the casting machine.

Experimental work has been performed on automotive fly-wheels cast flat in a machine which rotates the mold in a horizontal plane. Another application of centrifugal molding is in the production of automobile brake drums. These machines employ a vertically-operated rotating mold which forms the drum against which the brake bands are applied in operation. This outer drum, in many cases, employs steel fittings which are cast in place as the mold is poured.

## 7. MOLDING SAND

Molding sand consists of quartz, feldspar and clay in varying quantities, depending on the location from which it is obtained. The qualities demanded of a good molding sand are: refractoriness, or ability to resist the heat of molten metal; strength, or ability to withstand the pressure of molten metal without distortion; permeability, permitting ready escape of gases from the mold. Refractoriness is furnished by the quartz, or silica, of the sand, which if pure is practically infusible. Strength is due to the clay which acts as a bond between the grains of quartz. Permeability depends on grain structure and on quantity of clay. The more regular the silica grains and the smaller the quantity of clay substance present, the greater the permeability. A coarse sand is more permeable than a fine one, and a sand with round grains more permeable than with angular grains. Other desirable qualities are the ease of destruction of the mold after pouring and a low percentage of "burned" sand resulting from contact with molten metal. These two latter qualities are promoted by the clay bond being highly refractory. Moldenke defines the ideal molding sand as one consisting of uniform-sized, rounded grains of silica, each grain coated with the thinnest necessary layer of the most refractory and fattest clay to be had. O. E. J. Abrahamson (*The Fdry.*, March–Sept., 1930) says that the production of smooth castings depends mainly on the amount of fine grains (passing 300 mesh or finer) in the sand, the nature and amount of bond, the moisture content, and the method of preparation of the sand for use. The number of coarse grains (those remaining on a 100-mesh sieve) is relatively unimportant.

The variable factors in sand influence castings production more often than do metallurgical or mechanical considerations. The selection, blending and use of sand formerly was largely a matter of personal judgment, but mechanical and chemical methods of testing sand have largely come into use, particularly in production foundries, for both molds and cores. A joint committee of technical societies, sponsored by the Am. Foundrymen's Assoc. has investigated and standardized these methods, which are given in detail in Testing and Grading Foundry Sand, *Trans.* A. F. A., 1931.

In continuous production foundries, sand may be used as often as 4 or 5 times an hour, as compared with its use once a day, the usual practice up to about 1920. This continuous use requires cooling, tempering, replenishment of bond, and the addition of other variables to those inherent in the sand. The absolute uniformity required has developed sampling, blending, testing of the natural sand, and a new technique in treatment. Synthetic molding sand, *i.e.*, sand made of washed silica with the admixture of various types of clay bond has been successful in production foundries. Natural sands, continuously used, require the addition of bond to replace that destroyed by mold heat. Correct moisture content

**Table 21.—Percentage Fineness of Molding Sands.** (Moldenke.)

Average fineness = {sum of products of (percentage through each mesh × mesh No.)} ÷ 100

| Locality | No. of Samples | Average Fineness | Over 20 Mesh | Between 20–40 Mesh | Between 40–60 Mesh | Between 60–80 Mesh | Between 80–100 Mesh | Over 100 Mesh |
|---|---|---|---|---|---|---|---|---|
| Albany: Max....... | 23 | 99.38 | 0.00 | 0.12 | 0.40 | 0.44 | 0.52 | 98.52 |
| Min....... | ........ | 38.31 | 8.72 | 55.00 | 10.40 | 5.60 | 2.28 | 18.00 |
| Aver....... | ........ | ........ | 1.46 | 11.91 | 14.47 | 11.79 | 9.89 | 50.79 |
| Georgia........... | 1 | 69.26 | 0.08 | 0.24 | 36.00 | 14.48 | 15.36 | 33.84 |
| Illinois............ | 1 | 42.78 | 2.56 | 39.60 | 34.16 | 5.52 | 1.36 | 16.80 |
| Kentucky: Max..... | 13 | 99.59 | 0.00 | 0.04 | 0.00 | 0.20 | 1.48 | 98.28 |
| Min..... | ........ | 53.19 | 0.20 | 21.64 | 34.92 | 17.60 | 6.52 | 19.12 |
| Aver.... | ........ | ........ | 0.43 | 5.22 | 10.30 | 8.54 | 5.79 | 72.04 |
| Missouri: Max...... | 10 | 93.09 | 0.00 | 0.08 | 8.24 | 3.72 | 2.08 | 85.88 |
| Min...... | ........ | 44.04 | 2.84 | 40.00 | 27.20 | 9.60 | 4.80 | 15.56 |
| Aver..... | ........ | ........ | 0.85 | 16.62 | 18.13 | 10.80 | 5.81 | 48.21 |
| New Jersey: Max... | 4 | 53.02 | 5.76 | 12.72 | 39.60 | 16.24 | 3.92 | 21.76 |
| Min... | ........ | 36.88 | 17.12 | 37.36 | 19.44 | 9.04 | 4.16 | 12.88 |
| Aver... | ........ | ........ | 8.64 | 27.14 | 26.02 | 15.40 | 7.61 | 15.19 |
| Ohio: Max......... | 14 | 95.01 | 0.00 | 0.04 | 4.28 | 4.52 | 2.92 | 88.24 |
| Min......... | ........ | 43.21 | 9.24 | 37.44 | 21.60 | 8.20 | 6.80 | 16.72 |
| Aver......... | ........ | ........ | 0.57 | 12.11 | 25.54 | 14.55 | 7.23 | 39.45 |
| Penna.: Max....... | 9 | 95.85 | 0.00 | 1.04 | 1.36 | 3.72 | 89.48 | 95.85 |
| Min........ | ........ | 50.14 | 0.24 | 9.92 | 62.00 | 8.00 | 6.40 | 13.44 |
| Aver....... | ........ | ........ | 0.17 | 4.43 | 24.49 | 12.86 | 8.63 | 49.47 |
| Tennessee......... | 1 | 43.73 | 7.20 | 25.76 | 38.96 | 10.48 | 4.48 | 13.12 |

influences maximum strength characteristics; permeability, as affected by even slight changes in moisture content, may be controlled by rapid and effective routine tests.

**COMPOSITION OF MOLDING SAND.**—Molding sands occur in many parts of the United States and vary in the above qualities and in analysis over a wide range. Moldenke gives the following comparison of the ideal American sand and the average of 76 sands from various localities reported in *Trans.* A. F. A., xxi, p. 85, 1913.

| | Quartz | Clay | Feldspar |
|---|---|---|---|
| Ideal.................... | 84.00 | 13.50 | 2.50 |
| Average.................. | 65.53 | 21.75 | 12.75 |

The size of casting and the metal must be considered in the selection of molding sands, according to C. P. Karr (*Trans.* A. F. A., xxiv, p. 155, 1915). For steel castings, the sand should contain 97% or more silica with a small amount of clay for bonding purposes. Sands for iron castings should be loamy with a texture varied with the size of the casting. Sands for bronze should be of fine grain, good bonding power and have a high permeability. Sands for brass and aluminum should be even grained, loamy and should also have high permeability.

Table 21 shows the percent of fineness of the molding sands tested by Moldenke. The finest sand possible that will afford adequate venting power, should be used in any given mold, due consideration being given to the size of the mold. The finer the sand the better will be the surface of the casting.

Table 22 shows the maximum, minimum and average quantities of quartz, clay substance and feldspar in the 76 sands.

**STRENGTH OF MOLDING SAND.**—Table 23 shows the results obtained in strength tests of the 76 sands tested by Moldenke. Crushing strength of sand is important, as on it depends the resistance of the mold to pressure of the molten iron. The strength of

**Table 22.—Summary of Analysis of 76 Molding Sands.** (Moldenke.)

| Locality | No. of Samples | Quartz | | | Clay Substance | | | Feldspar | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Max. | Min. | Aver. | Max. | Min. | Aver. | Max. | Min. | Aver. |
| Albany............. | 23 | 72.66 | 45.77 | 58.82 | 24.87 | 13.68 | 18.99 | 30.06 | 8.17 | 22.16 |
| Georgia............ | 1 | 77.37 | 77.37 | 77.37 | 17.94 | 17.94 | 17.94 | 4.69 | 4.69 | 4.69 |
| Illinois............ | 1 | 70.82 | 70.82 | 70.82 | 16.65 | 16.65 | 16.65 | 12.53 | 12.53 | 12.53 |
| Kentucky.......... | 13 | 70 99 | 45.55 | 64.53 | 30.52 | 19.68 | 24.77 | 23.93 | 3.31 | 10.69 |
| Missouri........... | 10 | 77.76 | 56.91 | 64.10 | 32.19 | 18.08 | 24.36 | 21.73 | 2.88 | 11.54 |
| New Jersey......... | 4 | 88.70 | 72.20 | 81.38 | 23.99 | 8.85 | 15.49 | 3.81 | 2.45 | 3.13 |
| Ohio............... | 14 | 78.99 | 55.00 | 71.02 | 41.20 | 17.37 | 23.79 | 9.90 | 2.50 | 5.17 |
| Pennsylvania....... | 9 | 78.52 | 59.79 | 67.21 | 26.82 | 18.16 | 21.99 | 20.56 | 2.34 | 10.79 |
| Tennessee.......... | 1 | 74.53 | 74.53 | 74.53 | 21.11 | 21.11 | 21.11 | 4.36 | 4.36 | 4.36 |

**Table 23.—Permeability and Strength of Molding Sands. (Moldenke.)**

| Region | No. of Samples | Dry Sand | | | 5% Moisture | | | 7 1/2% Moisture | | | 10% Moisture | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Permeability * | Transverse Strength † | Crushing Strength ‡ | Permeability * | Transverse Strength † | Crushing Strength ‡ | Permeability * | Transverse Strength † | Crushing Strength ‡ | Permeability * | Transverse Strength † | Crushing Strength ‡ |
| FINE SANDS—BETWEEN 90% AND 100% FINE | | | | | | | | | | | | | |
| Albany | 4 | 1.31 | 1.16 | 9.62 | 1.31 | 0.20 | 2.79 | 1.19 | 0.22 | 2.94 | 1.44 | 0.11 | 1.96 |
| Kentucky | 8 | 1.47 | 1.31 | 7.41 | 1.91 | 0.19 | 3.01 | 2.16 | 0.17 | 3.07 | 3.78 | 0.19 | 2.74 |
| Ohio | 2 | 0.88 | 1.98 | 6.94 | 2.50 | 0.15 | 4.66 | 2.50 | 0.04 | 2.34 | 2.88 | 0.08 | 2.02 |
| Pennsylvania | 2 | 0.88 | 1.20 | 9.59 | 0.88 | 0.16 | 2.14 | 0.88 | 0.09 | 3.47 | 1.25 | 0.06 | 1.97 |
| Missouri | 2 | 1.38 | 0.69 | 4.93 | 1.25 | 0.16 | 3.05 | 1.38 | 0.21 | 2.76 | 2.50 | 0.20 | 3.42 |
| Average of | 18 | 1.29 | 1.28 | 7.82 | 1.65 | 0.13 | 3.11 | 1.75 | 0.16 | 2.97 | 2.71 | 0.15 | 2.48 |
| MEDIUM SANDS—BETWEEN 80% AND 90% FINE | | | | | | | | | | | | | |
| Albany | 6 | 1.04 | 1.49 | 9.82 | 0.66 | 1.00 | 3.08 | 0.87 | 0.17 | 3.28 | 1.08 | 0.10 | 2.35 |
| Kentucky | 1 | 0.75 | 2.65 | 27.40 | 0.50 | 1.20 | 9.75 | 1.00 | 0.89 | 4.63 | 0.75 | 0.25 | 3.85 |
| Ohio | 1 | 0.75 | 2.08 | 18.15 | 0.75 | 0.13 | 5.84 | 0.75 | 0.00 | 2.58 | 1.00 | 0.00 | 1.30 |
| Pennsylvania | 2 | 0.75 | 1.55 | 16.24 | 0.75 | 0.17 | 3.09 | 0 75 | 0.14 | 2.96 | 0.88 | 0.14 | 2.26 |
| Average of | 10 | 0.93 | 1.68 | 13.70 | 0.68 | 0.76 | 4.03 | 0.85 | 0.22 | 3.28 | 1 10 | 0.11 | 2.38 |
| COARSE SAND—BELOW 80% FINENESS | | | | | | | | | | | | | |
| Albany | 13 | 1.27 | 1.57 | 12.92 | 0.77 | 0.29 | 3.92 | 0.89 | 0.15 | 2.95 | 1.18 | 0.12 | 2.37 |
| Kentucky | 4 | 1.06 | 1.18 | 18.69 | 0.94 | 0.58 | 8.50 | 1.06 | 0.57 | 6.56 | 1.38 | 0.38 | 8.06 |
| Ohio | 11 | 0.84 | 2.29 | 12.28 | 0.98 | 0.50 | 5.67 | 1.00 | 0.38 | 4.57 | 1.14 | 0.24 | 4.80 |
| New Jersey | 4 | 2.38 | 1.65 | 16.93 | 1.25 | 0.64 | 3.22 | 2.00 | 0.37 | 2.44 | 3.38 | 0.40 | 2.16 |
| Pennsylvania | 5 | 1.95 | 2.30 | 23.01 | 1.10 | 0.50 | 5.36 | 1.90 | 0.45 | 3.31 | 2.23 | 0.24 | 3.21 |
| Missouri | 8 | 1.19 | 2.00 | 10.60 | 1.03 | 0.34 | 3.52 | 1.31 | 0.28 | 2.83 | 1.88 | 0.20 | 3.24 |
| Illinois | 1 | 0.50 | 3.74 | 30.80 | 0.50 | 0.20 | 5.89 | 1.50 | 0.10 | 2.95 | 3.25 | 0.57 | 7.24 |
| Georgia | 1 | 2.00 | 1.39 | 4.32 | 1.00 | 0.20 | 2.80 | 3.25 | 0.40 | 2.70 | 3.25 | 0.50 | 1.58 |
| Tennessee | 1 | 2.75 | 0.87 | 5.22 | 3.00 | 0.56 | 7.92 | 4.00 | 0.30 | 5.03 | 5.00 | 0.15 | 0.93 |
| Average of | 48 | 0.97 | 1.88 | 14.29 | 0.89 | 0.42 | 4.82 | 1.34 | 0.31 | 3.43 | 1.79 | 0.24 | 3.68 |

* Minutes required to drive 1 gal. of air, in a 1/4-in. pipe through a cake of dried sand 1 in. thick under a pressure corresponding to a 7-in. head of molten metal, in excess of time required to pass out of apparatus freely.

† Weight, lb., required to rupture bars 1 in. sq. placed on supports 4 in. apart.

‡ Weight, lb., required to crush columns of sand 1 in. sq. and 2 in. high.

molding sand depends largely on its clay content. The larger the quantity and fatter the clay, the stronger will be the sand. High strength, however, usually means low permeability, since the clay fills the voids between the sand grains. Strength of molding sand is tested by slowly pushing bars of sand 1 in. square and 1 ft. long, over the edge of a glass plate and noting the length of the overhang as they break.

Another method is to break a 1 in. square bar supported on knife edges 4 in. apart by a load applied at the center. The strength of the sand is affected by the amount of water used for tempering, but tests usually are made with 5%, 7 1/2% and 10% of water.

## Sand Testing

Complete details of sand testing methods available in testing and grading foundry sands are given in Testing and Grading of Foundry Sand published by Am. Foundrymen's Assoc., 1931. A brief of the essential tests follows:

SAMPLING.—For plant control tests, an average representative selection of facing or heap sand should be made as tempered for use in mold or core. To determine properties under suitable foundry conditions, several water contents should be tested to develop that giving the maximum degree of property being investigated, starting with a water content that leaves the sand dry, and increasing by small increments. Open sand, bonded with colloidal clay, may require increments of 0.1%, while natural bonded sands may be tested satisfactorily with increments of 2%. In tempering, care should be taken to distribute water evenly through the sand, with no dry lumps remaining. The sand should temper in a closed receptacle for several hours before testing.

**THE MOISTURE CONTENT** of sand sample is determined as follows: Dry 100 gm. of tempered sand for 1 hr. at 105 to 110° C. (221–230° F.). Cool in a desiccator or covered dish. Reweigh. Loss of weight in grams is percentage moisture content.

**PERMEABILITY** is that physical property of sand that permits passage of gases, and on which depends the venting qualities of molds and cores. A high permeability sand has good venting quality due to its "openness." The natural characteristics of the sand and binders, the density to which they are packed, and the percentage of moisture used for tempering, are important in regulating permeability. Permeability is determined by the rate of flow of air at a given pressure through a standard specimen of sand. In the standard test a 2-in. (5.08 cm.) column of sand (area, 3.1416 sq. in. or 20.268 sq. cm.), rammed under uniform and predetermined conditions is placed in a cylinder equipped with a manometer. The time required to force 2000 cu. cm. of air through the sand is determined. Permeability, $P = (q \times h)/(a \times h \times t)$, where $q$ = cu. cm. of air forced through specimen; $h$ = height of specimen, cm.; $p$ = air pressure, grams per sq. cm.; $a$ = area of specimen, sq. cm.; $t$ = time, min. A number of variations in standard test procedure are optional for rapid shop control, green sand permeability and mold surface permeability requirements.

**THE STANDARD STRENGTH TEST METHOD** has been approved as standard to be used where comparisons are necessary. Compressive, tensile, shear or bar strength tests all may be applied as routine in foundry practice control, depending on properties which sand is intended to develop in use, but compression test is standard in case of disagreement. The compressions test may be made on the green and permeability specimen (see above) by applying a load uniformly to the two plane surfaces at the rate of 30 lb. per min. with a tolerance of 10 lb. per min.; the load to be applied along the axis of the cylindrical specimen. Compressive strength is the average of three tests, lb. per sq. in. of cross-sectional area. Various types of apparatus are provided for standard compression tests.

**FINENESS TEST** refers to the size distribution of the grains. The fineness of sand affects permeability, strength, moisture required for mixing and affects the surface of the casting. In fineness tests, U. S. Bureau of Standards sieves Nos. 6, 12, 20, 30, 40, 50, 70, 100, 140, 200, and 270 are used. These sieves, 8 in. diam., are known as half height. A pan at the bottom of the set collects material passing the lower, 270-mesh sieve. In testing, 50 gm. of dry sand are weighed into the uppermost of the series of sieves, previously placed in a shaker. The sieves are shaken for a definite time, depending on type of shaker. The amount of sand remaining on each sieve is weighed and expressed in percentage of the original sample weight. The portion passing the 270 sieve is called pan material. Sand containing clay or bonding substance is washed according to a predetermined procedure, the bonding material siphoned and the remaining sand dried and tested as above.

**MOLD SURFACE HARDNESS.**—The density to which molding sands are rammed affects two important physical properties of the mold: 1. Ability of the mold surface to withstand pressure of molten metals, 2. Relative permeability of the mold. An indication of these properties is obtained by a device to determine hardness of mold surface, which measures depth of penetration into the mold surface of a round-nose plunger 0.2 in. diam; the load is applied in a direction perpendicular to the mold surface. The instrument has a recording device that indicates to 0.001 in. the penetration of the plunger. For calibration, a pressure of 237 gm. on free end of plunger should indicate a movement of 0.1 in. on the dial. The instrument should be checked at frequent intervals.

**DYE ADSORPTION TEST** of molding sand is solely to ascertain the nature of the clay substance present. Different sands possess widely different adsorption capacities. The colloids in molding sands are mostly inorganic, comprising hydrated aluminum silicate, hydrated iron oxide, hydrated silica acid and other hydrated minerals. All of these are gelatinous and sticky and give the sand the property of bond. Strongly bonded molding sands commonly possess clay substance high in colloid content as measured by dye adsorption test. Weaker bond sands usually show a lower dye adsorption corresponding to the smaller quantity of colloids in the clay substance of those sands. Methods of making the dye adsorption test are given in the above-noted booklet.

**CORES AND CORE MIXTURES.**—Cores are made of sand and a suitable binder, which may be either dry or liquid. The sand should be high in silica to resist heat of the molten metal, and low in alumina. Old sand and gangway sand may be used in part for the sand mixture. The grains should be well rounded to afford venting power to the core. Linseed oil makes the best binder, but is expensive. Other binders in common use are flour, glue, molasses, resin and resin oil, as well as many proprietary compounds.

The usual proportions of the foregoing materials are as follows:

**Linseed Oil,** 1 part oil to 30 parts sand. Cores are strong, porous and easily removed from the casting.

**Flour,** 1 part flour to 18 to 30 parts sand. The lower the proportion of flour, the more porous but weaker will be the core. The flour should be made into a thin boiled paste which is used to dampen the sand.

**Glue,** 2 lb. of glue dissolved in 3 gal. of water to 100 lb. of sand. Cores are smooth, but friable. They give off less gas than other cores.

**Molasses,** 1 part, mixed with 20 parts of water, usually is used as a spray for core surfaces.

**Resin,** 1 part to 20 or 30 parts of sand.

**Resin Oil,** 1 part to 18 to 24 parts of sand.

Cores containing molasses, flour or glue must be kept dry as they soften when exposed to dampness. They should not be placed in the molds very long before pouring.

The results of a series of experiments made by A. M. Loudon are given in *Trans. A. F. A.*, xv, p. 34, 1906. The following core mixtures which gave satisfactory results, have been selected from those published.

1. Old flour sand, 1 part; new molding sand, 1 part; sharp or beach sand, 1 part; 1 part Syracuse dry core compound with 35 parts sand. Cores dry quickly; are clean and sharp and resist dampness and leave a good surface.

2. Old flour sand, 2 parts; new molding sand, 1 part; beach or sharp sand, 1 part; 1 part dextrine or British gum mixture to 150 parts of sand, tempered with water. Cores are strong, dry easily and have sharp edges. Useful for large cores.

3. Old sand, 1 part; new molding sand, 1 part; sharp sand, 1 part: 1 part dextrine to 100 parts of sand. Useful for small intricate cores.

4. Old sand, 1 part; new molding sand, 1 part; sharp sand, 1 part; 1 part Wago core compound to 30 parts of sand. This mixture makes a good core which gives off but little smoke.

5. New molding sand, 1 part; sharp sand, 1 part; 1 part Wago core compound to 35 parts of sand. Makes a strong, true and sharp core, but is not so easily removed from castings as No. 4. Resists dampness well.

6. Old sand, 1 part; sharp sand, 1 part; new molding sand, 1 part; 1 part Cleveland core compound to 30 parts of sand, tempered with water. Makes strong core for general use. Easily removed from the casting.

7. Old sand, 1 part; sharp sand, 1 part; new molding sand, 1 part; 1 part flour to 18 parts of sand. Makes a good core for general work.

8. Sharp sand, 30 parts; Holland linseed core mixture, 1 part. Makes a strong core for small and medium shapes. Requires venting. Resists moisture.

9. Sharp sand, 35 parts; Syracuse core oil mixture, 1 part, tempered with water. Makes good cores but requires venting. Resists dampness well.

10. New molding sand, 1 part; sharp sand, 1 part; old sand, 2 parts; 1 lb. of glue to 100 lb. of sand. Recommended for small cores. Used without venting.

11. Sharp sand, 1 part; new molding sand, 1 part; old sand, 1 part; 1 lb. glucose to 100 lb. sand. Makes a first-class core, easily removed from the casting and emitting no smoke.

An exhaustive research on core sands, core binders and core-room equipment was made by H. M. Lane (*Trans.* A.S.M.E., xxxiii, p. 735, 1911). Mr. Lane says that a core of the most exacting requirements, made of oil-sand mixtures, must have a tensile strength of at least 75 lb. per sq. in. The sand must contain no loam which will destroy the bonding value of the oil. The paint and varnish oils, including China wood, Soya bean, corn and cottonseed oils, are used in core oils. Blended oils also carry resin and neutral oil. The latter does not add appreciably to the binding power of the compound but carries the resin into the mixture and gives a waterproof oil of sufficient strength for ordinary work at lower cost than linseed oil. Greater strength may be afforded by using a bonded sand; *i.e.*, a sand containing clay, or by using flour, dextrine or other glutinous material. Strength of the dry core is governed by the amount of pressure it must resist as metal enters the mold. All cores tend to float, and if not rigid enough, may be broken or displaced.

**Facing Materials.**—Facing materials are those substances added to the molding sand rammed next to the pattern to give a smooth finish to the casting. Formerly it was the custom to add a finely ground gas coal to the sand, about an inch of this mixture being rammed against the pattern, the balance of the mold being formed of ordinary molding sand. Moldenke claims that the advantages of this so-called " sea coal " are much overrated and advocates the use of prepared facings, such as graphite, ground coke, charcoal or anthracite, soapstone, etc. Graphite, prepared with a binder to cause it to adhere to mold surfaces is widely used on green sand molds, being dusted on and smoothed with the trowel. On dry sand molds it should be wet with molasses water, clay wash or other substances of adhesive properties, and brushed on.

## 8. MOLDING MACHINES

Molding machines may be used either to save time in the ramming of the mold or in drawing the pattern. The latter saving is due not only to the more rapid work of the machine as compared to hand draft but also to the elimination of the labor of patching, which is frequently necessary with hand drawing. The density of the machine-rammed mold is more uniform than the hand-rammed one and the castings usually are free from scabs, blisters, etc.

**TYPES OF MOLDING MACHINE.**—Molding machines may be classified according to the method of ramming, as hand-rammed, squeezer, jar-ram machines or gravity machines.

**The Squeezer,** as the name implies, packs the sand by squeezing it between two platens. It is adapted to the average type of mold where the depth of sand is not excessive and where there are no deep pockets involving large bodies of hanging sand. The squeezing operation can be performed either by hand or power, compressed air usually being the medium for the latter.

**The Jar-ramming Machine** comprises a table upon which rest the pattern, flask and its contained sand. The table is lifted a short distance and allowed to drop suddenly, the inertia of the falling sand packing it around the pattern when it is brought to rest by striking the table. The lift of the table ranges from $^1/_4$ to 3 in., 2 $^1/_4$ in. being perhaps the average lift. With this machine the mold is densest immediately around the pattern and least dense at the outer surface. The number of blows to obtain the necessary hardness or density of the molding sand varies with: 1. Nature of the molding sand. 2. Kind of metal, brass, cast iron or steel. 3. Depth of flask. 4. Locally, *i.e.*, with the individual ideas of the foundrymen. The range is from 3 to 100 blows, at about 80 blows or jars per minute. Tests to determine the number of effective blows to give the greatest density of the molding sand showed that at 80 blows the sand had almost maximum density, a larger number of blows increasing the density but little, while the sand cracked in places.

Jar-ramming machines often are combined with the squeezer principle, the mold being jarred a few times to pack the sand in cavities and around corners and edges of the pattern and flask. The operation is finished by squeezing the mold as in the plain squeezer machine.

**The Shockless Jarring Machine,** built by Tabor Mfg. Co., Philadelphia, and described by Wilfred Lewis (*Trans.* A.S.M.E., xxxii, p. 789, 1910), has a table mounted on a plunger fitted into a cylinder base forming an anvil, and supported on springs which are compressed when the table is lifted by compressed air. When the air is released, the table drops and the springs force the anvil upward, each having approximately equal momentum at the instant of impact. The sand is rammed without any of the shock of impact being transmitted to the foundation or surrounding floor.

Molding machines also can be classified as: Split pattern machines, for molding patterns which can be divided on a plane and mounted on a plate; stripping-plate machines in which the pattern is drawn through a plate which supports the sand in the mold while the pattern is drawn; roll-over machines, in which the completed mold is rolled over by the machine and the pattern drawn in the rolled over position. Either hand or power may be used for any or all of the operations of these machines. The several fundamental types of machines are combined to form a great variety of machines to suit various foundry conditions. The following classification is given by J. J. Wilson and A. O. Backert (*Trans.* Am. Fdry. Assoc., xxiv, p. 269, 1916):

| | |
|---|---|
| Squeezers. | Hand squeezers. |
| | Power squeezers. |
| | Hand squeezer with stripping plate attachment. |
| | Squeezer of the adjustable drop plate type. |
| | Split pattern squeezers, hand and power type. |
| | Automatic machines. |
| Stripping Plate | Stripping plate machines, hand and power draw. |
| | Stripping plate machines, drop plate type. |
| Roll-over | Hand-rammed roll-over. |
| | Hand-rammed, power roll-over and power draw. |
| | Hand-rammed, hand roll-over, foot draw. |
| | Gravity machine, roll-over, pattern draw type. |
| Jar-ramming | Plain jar-ramming, air and electrically operated. |
| | Foot jolt machine with roll-over attachment. |
| Combination Machines Embodying Jar-ramming Principle. | Jar-ram, roll-over, pattern-drawing. |
| | Jar-ram and squeezer. |
| | Jar-ram, squeezer and pattern-drawing. |
| | Jar-ram, squeezer, roll-over, pattern-drawing. |
| | Jar-ram, pattern-drawing. |
| | Jar-ram, roll-over. |
| | Jar-ram, air-squeezer, stripping plate. |
| | Jar-ram, split pattern. |
| | Jar-ram, squeezer, split pattern. |
| | Jar-ramming and stripping plate. |

**SANDSLINGERS.**—A molding method differing from the conventional types which employ pressure squeeze or jolt ramming is defined as sand slinging. It involves throwing sand into the mold, relying on an inertia effect to compact the sand about the pattern. It is an outgrowth of the old-time molder's practice of throwing handfuls of sand against a pattern prior to adding additional sand which is hand rammed or jolted.

The mechanical ramming of sand by the sandslinger method is accomplished by feeding the sand into the path of a revolving cup which projects the sand outward against a fixed liner forming a wad. After the wad is formed, it is projected from the slinger head at a velocity of 4000 to 8000 ft. per min., which is ample to ram molds to varying degrees of density, a new wad being formed and discharged at each revolution of the ramming motor.

Mechanically, it is made so it can be passed over various size molds, from snap flasks to pit molding, producing castings that weigh ounces or many tons. In ramming molds, it can be passed over the mold by man or mechanical power. In production work of various kinds, it requires no operator. Sandslingers range in their capacity from five to thirty cubic feet of sand per minute, and they operate entirely by electric power.

**CORE BLOWING MACHINES.**—The pneumatic core blowing machine is used for producing small and medium size dry sand cores in commercial quantities, ranging in size from small cores such as are used for small fittings and lock work, up to cores weighing approximately 50 lb., such as automobile crank-case cores and similar work of a "chunky" type. Lighter cores of the rangy, lacy type such as automobile jacket work, are being blown successfully up to approximately 35 in. long.

The core blowing machine consists principally of a sand reservoir and blow plate mounted on a carriage which travels back and forth from the filling to the blowing positions. This carriage is suspended from the upper structure of the machine, which is in turn supported on a base carrying the table and vertical and horizontal clamping mechanisms. The core box is filled by projecting air at high pressure (approximately 110 to 120 lb. per sq. in.) onto the sand in the reservoir, thus forcing a stream of sand and air at high velocity through the holes in the blow plate into the cavity of the core box. Vents are provided in the core box to allow the air to pass on through. The vents resist the passage of the sand grains, and trap the sand in the core box cavity. The size and location of the vents are so arranged that the core box will fill properly.

The satisfactory operation of a core blowing machine requires first-class metal core boxes, with tight joints throughout, and sufficient land at the parting line and around the blow holes to insure a tight seal. Opposite outside surfaces must be machined parallel, so that the box will clamp squarely in the machine. Metal core dryers are necessary, except when the cores are of a nature that will permit standing them on a plate, following the usual practice.

The use of a core blowing machine should be confined to conditions where the expense for metal core box equipment is justified, and where the required sand mixture is free from heavy bond and sufficiently fluent to work freely in the machine.

**FIELD OF DIFFERENT TYPES OF MOLDING MACHINES.**—It is difficult to define the particular class of work to which each type of machine is best adapted. In general, the squeezer is used on comparatively shallow molds, or molds of the bench work class, while the jarring machine is adapted to deep molds, or molds where there are cavities in which it would be difficult to pack the sand by pressure alone. It may be said to be a machine for light or heavy floor work. The combination jar-ram squeezer is used for work a little deeper than that for which the plain squeezer is adapted, or where the sand requires hand tucking in pockets and corners, before the squeezing operation. Instead of hand tucking the mold is jarred a few times and then squeezed.

The split-pattern machine is used where the patterns are symmetrical or of the flat-back type. With symmetrical patterns, but one pattern plate is used, cope and drag being molded from a double set of half patterns, the mold in the right-hand side of the drag matching that in the left-hand side of the cope and *vice versa.* With flat-back work, all the drags are molded from the pattern, and then a blank plate is placed in the machine and all copes are made.

Stripping-plate machines are used where there is a long run of duplicate work. They are not economical for short runs due to the expense of fitting up the stripping plates. For short runs, where the sand would require support while drawing the pattern, a roll-over pattern-drawing type of machine is preferable. Stripping-plate machines are especially adapted to gears, sprockets and patterns of irregular outline such as binder and mower frames.

The roll-over machine is useful where the size of the mold is such that the labor of turning it over is considerable. While it may be used for light work, its true field begins with what is known as heavy bench or light floor work, and extends in the power machines to molds weighing several tons.

The roll-over machine may be either a plain hinged or a straight draft machine. In the former, the pattern plate revolves about the hinge-pin, while in the latter it is lifted vertically until the pattern is clear of the mold and is then rolled back to molding position. The former type may be used on shallow work with considerable draft, such as stove-plate, grates, etc. The straight draft machine is used for deeper work, as fly-wheels, pistons, cylinders, automobile crank- and transmission-cases, etc. In connection with jar-ramming the roll-over machine is used for railway trucks, planer and boring mill housings, milling machine columns, bases and tables, lathe beds, ship castings, etc.

For details of construction of molding machines, sizes, capacities, and the adaptability of a particular class of machine to a given type of work, manufacturers of this class of equipment should be consulted.

## 9. MISCELLANEOUS FOUNDRY EQUIPMENT

**LIFTING BEAMS** for foundry use are of various forms. For light loads, a light forged spreader, of triangular form with hook extensions at the two lower corners of the triangle for attaching the load, is serviceable. For large, heavy copes, a beam in the form of a cross is frequently used, to permit the cope to be suspended from 4 or more points. I-beams, suspended at two points from a lifting ring with the load slings placed outside the points of suspension, are the strongest and most reliable of all types. Table 24 gives safe loads for this type of beams.

**Table 24.—Safe Loads for Lifting Beams**

| Depth of I-Beam, in. | Distance Between Slings, in. | Weight per foot, lb. | Safe Load for Extreme Fiber Stress of 12,000 lb. per sq. in., lb. | Depth of I-Beam, in. | Distance Between Slings, in. | Weight per foot, lb. | Safe Load for Extreme Fiber Stress of 12,000 lb. per sq. in., lb. | Depth of I-Beam, in. | Distance Between Slings, in. | Weight per Foot, lb. | Safe Load for Extreme Fiber Stress of 12,000 lb. per sq. in., lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 8 | 16 | 4,800 | 10 | 12 | 33 | 10,750 | 20 | 16 | 80 | 36,200 |
| 6 | 10 | 16 | 3,800 | 12 | 10 | 40 | 18,750 | 20 | 18 | 80 | 32,200 |
| 8 | 8 | 22 | 9,000 | 12 | 12 | 40 | 15,600 | 24 | 18 | 80 | 38,150 |
| 8 | 10 | 22 | 7,200 | 15 | 14 | 80 | 29,950 | 24 | 20 | 80 | 34,300 |
| 10 | 10 | 33 | 12,900 | 15 | 16 | 80 | 26,200 | | | | |

**Table 25.—Dimensions of Foundry Ladles**
(Northern Engineering Works, Detroit)

| Capacity, lb. | Size of Bowl, in. | | | Allows for Lining, in. | | | Thickness of Steel, Gage or in. | | Weight of Crane Ladle, lb. |
|---|---|---|---|---|---|---|---|---|---|
| | Diam. Top | Depth | Diam. Bottom | Bottom | Sides | Top | Bottom | Sides | |
| HAND LADLES | | | | | | | | | |
| 30 | 7 | 7 | 6 | 1/2 | 1/2 | 3/4 | No. 14 | No. 16 | ........ |
| 50 | 8 | 8 | 7 | 3/4 | 1/2 | 3/4 | 14 | 16 | ........ |
| 100 | 10 | 10 | 8 3/8 | 3/4 | 1/2 | 1 1/2 | 14 | 16 | ........ |
| 150 | 11 | 11 | 9 3/4 | 1 | 1/2 | 1 1/2 | 14 | 16 | ........ |
| 200 | 12 | 12 | 10 1/2 | 1 | 1/2 | 1 1/2 | 14 | 16 | ........ |
| SHANK LADLES | | | | | | | | | |
| 250 | 13 | 13 | 11 1/4 | 1 | 1/2 | 1 1/2 | No. 14 | No. 16 | ........ |
| 300 | 14 | 14 | 12 | 1 | 5/8 | 1 1/2 | 3/16 in. | 1/8 in. | ........ |
| 400 | 15 1/2 | 15 1/2 | 13 1/2 | 1 1/4 | 3/4 | 1 3/4 | 3/16 | 1/8 | ........ |
| 500 | 16 1/2 | 16 1/2 | 14 1/2 | 1 1/4 | 3/4 | 1 3/4 | 3/16 | 3/16 | ........ |
| 600 | 17 1/2 | 17 1/2 | 15 | 1 1/4 | 3/4 | 1 3/4 | 3/16 | 3/16 | ........ |
| 800 | 19 | 19 | 16 1/2 | 1 1/4 | 3/4 | 1 3/4 | 1/4 | 3/16 | ........ |
| 1,000 | 20 1/2 | 20 1/2 | 18 | 1 1/2 | 1 | 2 | 1/4 | 3/16 | ........ |
| 1,200 | 21 3/4 | 21 3/4 | 19 | 1 1/2 | 1 | 2 | 1/4 | 3/16 | ........ |
| 1,500 | 23 | 23 | 20 | 1 1/2 | 1 | 2 | 1/4 | 3/16 | ........ |
| CRANE LADLES | | | | | | | | | |
| 2,000 | 26 | 26 | 23 | 1 1/2 | 1 | 3 | 1/4 | 1/4 | 650 |
| 3,000 | 29 | 29 | 26 | 1 1/2 | 1 | 3 | 1/4 | 1/4 | 700 |
| 4,000 | 32 | 32 | 28 | 1 1/2 | 1 | 3 | 1/4 | 1/4 | 900 |
| 5,000 | 34 | 34 | 30 | 2 | 1 1/2 | 3 | 5/16 | 1/4 | 1,100 |
| 6,000 | 36 1/2 | 36 1/2 | 32 | 2 | 1 1/2 | 3 | 5/16 | 5/16 | 1,370 |
| 8,000 | 40 1/2 | 40 1/2 | 35 1/2 | 2 1/2 | 1 3/4 | 3 | 5/16 | 5/16 | 1,600 |
| 10,000 | 43 1/2 | 43 1/2 | 38 | 2 1/2 | 1 3/4 | 4 | 3/8 | 5/16 | 1,900 |
| 12,000 | 48 | 42 | 42 | 2 1/2 | 1 3/4 | 4 | 3/8 | 5/16 | 2,200 |
| 14,000 | 50 1/2 | 44 1/2 | 44 1/2 | 2 1/2 | 2 1/4 | 4 | 3/8 | 3/8 | 2,600 |
| 16,000 | 53 | 47 | 44 | 2 1/2 | 2 1/4 | 4 | 7/16 | 3/8 | 3,100 |
| 16,000 | 57 | 50 | 50 | 5 | 4 1/2 | 4 | 7/16 | 3/8 | 3,700 |
| 20,000 | 57 | 50 | 50 | 2 1/2 | 2 1/4 | 4 | 7/16 | 3/8 | 3,800 |
| 20,000 | 60 | 54 | 54 | 5 | 4 1/2 | 4 | 7/16 | 3/8 | 4,200 |
| 24,000 | 60 | 54 | 54 | 2 1/2 | 2 1/4 | 4 | 7/16 | 3/8 | 4,350 |
| 24,000 | 65 | 57 | 57 | 5 | 4 1/2 | 4 | 7/16 | 3/8 | 4,800 |
| 30,000 | 65 | 57 | 57 | 2 1/2 | 2 1/4 | 4 | 7/16 | 3/8 | 5,000 |
| 30,000 | 69 | 61 | 61 | 5 | 4 1/2 | 4 | 1/2 | 7/16 | 5,900 |
| 40,000 | 75 | 66 1/2 | 66 1/2 | 5 | 4 1/2 | 4 | 3/4 | 7/16 | 7,000 |
| 50,000 | 80 | 70 | 70 | 5 | 4 1/2 | 4 | 3/4 | 7/16 | 8,100 |
| 60,000 | 84 | 75 | 75 | 5 | 4 1/2 | 4 | 3/4 | 1/2 | 10,200 |

**Table 26.—Free Air Discharge, Cu. Ft. per Min., and Power Required for Abrasive Blast Nozzles**

20% Added for Friction; Single Stage Compression at Sea Level

| Diam. of Nozzle, in. | | 25 | 30 | 35 | 40 | 45 | 50 | 60 | 70 | 80 | 90 | 100 | 125 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/8 | Cu. ft. | 8.6 | 10.0 | 11.2 | 12.2 | 13.4 | 14.5 | 16.8 | 19.0 | 21.2 | 23 | 25 | 31 |
| | Hp. | 0.7 | 1.0 | 1.2 | 1.5 | 1.7 | 2.0 | 2.7 | 3.3 | 4.0 | 4 | 5 | 7 |
| 3/16 | Cu. ft. | 19.4 | 22.4 | 25.0 | 27.4 | 30.4 | 32.8 | 37.5 | 42.9 | 47.5 | 52 | 58 | 70 |
| | Hp. | 1.7 | 2.2 | 2.8 | 3.4 | 4.0 | 4.6 | 6.0 | 7.5 | 9.0 | 10 | 12 | 17 |
| 1/4 | Cu. ft. | 34.5 | 40.0 | 44.7 | 49.0 | 53.8 | 58.2 | 67.0 | 76.0 | 85.0 | 94 | 103 | 125 |
| | Hp. | 3.0 | 4.0 | 5.0 | 6.0 | 7.1 | 8.3 | 10.7 | 13.4 | 16.2 | 19 | 22 | 31 |
| 5/16 | Cu. ft. | 54.0 | 63.0 | 70.0 | 76.7 | 84.0 | 91.0 | 105.0 | 119.0 | 132.5 | 147 | 161 | 196 |
| | Hp. | 4.8 | 6.4 | 7.9 | 9.5 | 11.2 | 13.0 | 16.9 | 21.0 | 25.3 | 30 | 35 | 49 |
| 3/8 | Cu. ft. | 77.0 | 90.0 | 100.0 | 110.5 | 121.0 | 130.0 | 151.0 | 171.0 | 191.0 | 211 | 231 | 282 |
| | Hp. | 6.9 | 9.1 | 11.3 | 13.6 | 16.2 | 18.6 | 24.3 | 30.0 | 36.5 | 43 | 50 | 71 |
| 1/2 | Cu. ft. | 138.0 | 161.0 | 179.0 | 196.4 | 215.0 | 232.0 | 268.0 | 304.0 | 340.0 | 376 | 412 | 502 |
| | Hp. | 12.2 | 16.3 | 20.1 | 24.2 | 28.7 | 33.1 | 43.1 | 54.0 | 65.0 | 77 | 90 | 127 |
| 5/8 | Cu. ft. | 216.0 | 252.0 | 280.0 | 307.0 | 336.0 | 364.0 | 420.0 | 476.0 | 532.0 | 587 | 645 | 785 |
| | Hp. | 19.2 | 25.6 | 31.7 | 37.9 | 44.9 | 52.0 | 68.1 | 84.0 | 101.7 | 120 | 140 | 198 |
| 3/4 | Cu. ft. | 310.0 | 362.0 | 400.0 | 441.8 | 482.0 | 522.0 | 604.0 | 685.0 | 765.0 | 843 | 925 | 1127 |
| | Hp. | 27.6 | 36.7 | 45.2 | 34.6 | 64.4 | 74.7 | 97.2 | 121.0 | 146.4 | 173 | 202 | 284 |
| 7/8 | Cu. ft. | 422.0 | 493.0 | 550.0 | 601.3 | 658.0 | 710.0 | 823.0 | 930.0 | 1004.0 | 1145 | 1260 | 1361 |
| | Hp. | 37.6 | 50.0 | 62.1 | 75.5 | 89.0 | 101.4 | 132.0 | 164.0 | 192.0 | 234 | 274 | 343 |
| 1 | Cu. ft. | 550.0 | 645.0 | 715.0 | 785.4 | 860.0 | 930.0 | 1070.0 | 1215.0 | 1360.0 | 1500 | 1648 | 2000 |
| | Hp. | 50.0 | 65.5 | 80.8 | 102.0 | 115.0 | 132.8 | 172.0 | 215.0 | 260.0 | 308 | 359 | 504 |

**Sandblasts** for cleaning castings are of the following types: gravity, suction and pressure. Good work may be done with a nozzle 1/4 or 3/8 in. diameter, with air pressure of 25 to 75 lb. per sq. in., depending on the material to be cleaned. Steel castings take higher pressures. The quantity of air used depends on air pressure and diameter of nozzle. No more air can be used than will be passed through the nozzle regardless of the diameter of air inlet. Since quantity of air used will increase rapidly as nozzle diameter and pressure are increased, the nozzle should be inspected and replaced at intervals. Quantity of air and power consumption are shown in Table 26.

The flow of sand should be so regulated as to deliver just enough to give a quick and thorough cleaning. This is effected by allowing each abrasive grain to strike the surface without interfering with its neighbors. When the sand can be seen issuing from the nozzle, the feed is about right. The nozzle should be held about 6 in. from the work and at an angle of 30° to 45°.

**AIRLESS ABRASIVE CLEANING.**—A type of machine employing the centrifugal method instead of pneumatic pressure for abrasive cleaning has been developed. This type of machine has a rotating head with cups or vanes, which takes small quantities of abrasive material, such as steel shot or grit. As each vane rotates, it delivers the abrasive material at high velocity and within a predetermined area on the surface of castings. The abrasive action of the material cleans the sand from casting surface. The same type of abrasive propelling element is employed in certain barrel or tumbling machines.

# MALLEABLE CASTINGS

By Enrique Touceda

The production of malleable castings dates from Réaumur's discovery that when a piece of hard and brittle white cast iron—that is iron, all of whose carbon is in combined form—was heated to a certain temperature and maintained at that temperature for a certain length of time, the iron then would be considerably less hard and brittle. Subsequent investigation showed that softness and ductility of the annealed product depended upon the composition of the white iron, and later researches that the most trustworthy product, measured by high ultimate strength and ductility, could be obtained only if the elements of the white iron were restricted to certain prescribed limits and the heat treatment conducted in accordance with certain rules, the violation of these rules resulting in an inferior product.

The manufacture of malleable castings thus comprises two steps: 1. The making of white iron castings of correct composition. 2. Heat treatment of the proper character. The melting is usually conducted in a reverberatory furnace (more commonly called an air-furnace). The open-hearth and electric furnaces are also used.

The customary mixture consists of: Pig iron, sprues from the previous heat and steel scrap for the purpose of carbon control. With the reverberatory or open-hearth furnace, the oxidizing action of the hearth atmosphere lowers the silicon and carbon of the charge —both during melting and also while molten metal is receiving sufficient superheat to enable the castings to be made without danger of misruns—to a point at which no carbon will be precipitated upon solidification. The castings then will show a white fracture. The product of the melting furnace, irrespective of the type used, is known as "hard iron" and should be, but not always is, completely white in fracture.

**COMPARISON OF MALLEABLE WITH GRAY IRON AND STEEL CASTINGS.**— Normal malleable cast iron as made in the U. S. is structurally unlike any other ferrous casting. The gray iron casting to which it is sometimes compared by the uninformed, usually consists of a ground-mass of varying proportions of ferrite and pearlite, through which are regularly distributed plates of graphite varying in thickness, length and curvature. If all the carbon is graphitic, which is rare, the ground-mass will consist entirely of ferrite. In limited cases where the combined carbon is high, say in the neighborhood of 0.90% or more, the ground-mass will consist of pearlite, or pearlite and free cementite. Gray iron castings sometimes are heat-treated, to facilitate machining, to increase the strength, hardness, or wear when pearlite predominates in the matrix, or to graphitize some of the pearlite in very low silicon cast irons. Hence the grain size in all sections in unannealed castings varies directly as the thickness. Since the percentage of combined carbon present in a gray iron casting depends appreciably upon the rate of cooling, the relative proportions of ferrite or cementite to the pearlite in the ground-mass will vary in the thick and thin sections of the same casting. The ordinary steel casting consists of a ground-mass of free ferrite and pearlite, the amount of pearlite increasing with the combined carbon content. So far as the structural constituents are concerned, the structure will be similar in the thick and thin sections, provided the sections are not abnormally disproportionate. The soft unannealed steel castings usually are sold in competition with malleable castings. In these the grain size will be coarse throughout the casting, and larger in the heavy sections than in the light sections.

The structure of malleable iron more closely resembles that of wrought iron, both having a ground mass of ferrite, the inert constituent in the malleable iron being temper carbon and in wrought iron, slag. The ferrite of the former, however, is much stronger because of the very large amount of silicon held in solid solution. The normal structure of malleable iron consists of but two constituents, both of which are soft. Its ground mass is practically 100% carbonless iron, or silico-ferrite, while uniformly distributed through it are small particles of carbon very different in shape and size from the plates of primary graphite that occur in gray iron. The white iron, which forms the basis of malleable castings, consists of cementite or carbide of iron, the hardest constituent that can exist in iron or steel, and cementite mechanically mixed with soft carbonless iron. Through heat treatment the carbide of iron is decomposed into its two soft constituents.

Inasmuch as malleable cast iron, due to the method of manufacture, is a product heat-treated slightly above the critical range, the grain size throughout the casting must be uniform and almost as fine as it is practical to obtain. This is of extreme importance in castings subject to dynamic stresses. Since the entire metallic part of a correctly annealed malleable casting consists of carbonless iron, such castings naturally must be soft and ductile, tough and free from internal stress.

**COMPOSITION OF MALLEABLE CASTINGS.**—The average reverberatory furnace mixture, to yield a tensile strength around 54,000 lb. per sq. in., accompanied by an elongation of 18.00% or over and the hard iron resulting therefrom, run about as follows:

| | Silicon | Max. Phosphorus | Max. Sulphur | Manganese | Carbon |
|---|---|---|---|---|---|
| Mixture........... | 1.25–1.35 | 0.20 | 0.10 | 0.60 | 2.65–2.80 |
| Hard Iron.......... | 0.95–1.10 | 0.20 | 0.10 | 0.22–0.30 | 2.00–2.40 |

Silicon in the mixture should be so adjusted that when the bath has received sufficient superheat to successfully run the castings, enough silicon has been eliminated from the bath to prevent graphitization upon solidification of the castings of heavy sections. If the silicon has not been lowered enough to accomplish this, it indicates too great an initial silicon in the mixture. This condition can be corrected by allowing the metal to remain in the furnace until both the silicon and carbon are still further lowered, which process, however, is wasteful of fuel. If the fracture of the test sprue, prior to complete superheating, shows white, the initial silicon of the mixture was too low and a white-iron composition will result that will yield, upon heat treatment, castings of abnormal steely fracture, lacking ductility. The addition of certain silicon-manganese-carbon combinations will correct this condition. Silicon can be replaced by the addition of ferro-silicon, and manganese by addition of ferro-manganese. The best practice is to use silico-spiegel which contains 6 to 12% of silicon and 17 to 22% manganese. As this alloy contains too little carbon, this element must be added separately. The usual procedure is to add a sufficient amount of petroleum coke for carbon replacement. Additions should be made immediately it is realized that any of the elements have been unduly eliminated. It is the common practice to make preliminary analyses as the heat progresses, and additions should be made immediately such analyses indicate that such are necessary.

**Phosphorus and Sulphur.**—Phosphorus in white iron should be under 0.20%; sulphur preferably under 0.10%.

**Manganese.**—The permissible amount of manganese is a function of the sulphur content as follows:

| | | | |
|---|---|---|---|
| Sulphur................ | 0.04 | 0.07 | 0.10 |
| Manganese............. | 0.20–0.24 | 0.23–0.27 | 0.27–0.34 |

**Carbon.**—If high tensile and ductile properties are desired, the carbon in the white iron should be about 2.25%. Such a carbon content will yield molten metal of sufficient fluidity to pour intricate castings. If less fluidity is permissible, the carbon can safely be as low as 2.00%.

If mixtures are made to accord with the above recommendations, hard iron of a composition suitable for sections up to 1 1/2 in. thick should be obtained without difficulty. Failure to obtain the proper primary mixture either through ignorance or lack of knowledge of the exact composition of the raw materials, may give "low-heat" or "high-heat" iron. Low-heat iron is a white iron so high in either silicon or carbon, or both, that primary graphite will be found in the heavy sections of the castings. No subsequent heat treatment can rectify this defect, and the castings will be undependable, as they will lack strength and ductility. This trouble is not likely to occur with any reasonable thickness of section if the carbon plus silicon is from 3.40 to 3.30%. High-heat iron, irrespective of the type of furnace producing it, is an iron so highly oxidized that too much silicon, carbon and manganese have been eliminated, or iron from a mixture in which the initial silicon was so low that white iron will have a composition giving an abnormal fracture when annealed, while it will be unsound and the surface covered with pin point holes.

**HEAT TREATMENT.**—The heat treatment of white iron castings, commonly, although improperly, called annealing, has for its object the decomposition of the free and pearlitic cementite that constitutes a large part of the structure of the white iron, by means of temperature, time and slow cooling. The very hard constituent, "cementite," was formed through a chemical union of iron and carbon, both of which are soft. The decomposition of the cementite brings about their separation. Consequently, the resultant product is a soft, strong, ductile and very easily machined metal, as no hard or brittle constituent is present. The author has frequently pointed out that the "anneal" in the case of hard iron castings should more properly be called a heat conversion rather than an annealing process.

In annealing, the castings are packed in rectangular or circular pots without bottom or cover, called rings. Rectangular rings usually are 24 × 18 × 15 in. high, and circular rings about 24 in. diameter by 15 in. high, the thickness of metal being 1 in. One of these rings is placed upon an iron "stool" of such horizontal dimensions that its edges extend about 3 in. beyond the sides of the ring. The stool serves both as a bottom for the ring that is placed upon it, and as a base for the entire assemblage of rings that are to be superimposed upon the lowest one. The stool is some 6 in. high and designed with feet so spaced that the forked lifter of the charging truck can slide under it to slightly raise the load and permit the truck to place it in the annealing oven. The detailed procedure is so to fill the ring that rests directly on the stool with castings and packing that each casting will be completely surrounded with the packing material. As soon as this ring has been packed solid, other rings in turn are superimposed upon it and a similar procedure followed in the case of each ring. The entire set, which may consist of 4 or 5 rings, is called a stand. The castings in the top ring should be kept not less than 4 in. from the top and packing to that depth filled in over them. A 2-in. luting of mixed clay is placed over the top of the stand, the joints between the rings being luted with the same material. After transference to an annealing oven, which is sealed, the castings are subjected to the following heat cycle: 1. Temperature of the oven is gradually raised to 1550° F. or higher in 40 hours. 2. The temperature is held at 1600° for about 60 hours and firing then discontinued. 3. The temperature is lowered at a rate that should not exceed 10° per hour, until it is certain that the temperature of the castings does not exceed 1200° F., after which the rate of cooling can be progressively accelerated. Heat treatment thus covers a period of 6 1/2 days. It is not recommended, nor is it practical in an ordinary type of oven, to attempt to shorten this period.

The tunnel type of oven is about 200 ft. long. This is constantly under fire and the annealing process is continuous, since the stands are introduced at one end of the oven on cars and withdrawn at the other end, the rate of travel through the oven being adjusted to accommodate the cycle adopted. By this method the total period of annealing can be shortened to not greatly exceed 100 hr. with a consequent saving of fuel.

A type of oven in which the anneal can be run through in a much shorter time is the car-type electric annealing oven, in which an anneal can be completed in three days or less. This method of annealing is advantageous because it makes possible more prompt delivery, and because the surface decarburization of the castings can be more closely controlled.

Another type of annealing apparatus is the muffle oven, in which castings that can be annealed without undue warping are stacked very closely together on the floor of the oven; other layers of castings are superimposed until the oven has been completely loaded. Frequently, plates of thin sheet steel are placed between each of several layers if the castings are not too large. As the products of combustion do not come in contact with the castings, a packing material is unnecessary, since if the muffles are tight no danger will exist of the castings being scaled. On the one hand, a product of the highest character can be produced in such an oven. This type is very flexible, due to the fact that very large castings cannot be heat-treated in any other type of annealing oven.

Small car-type annealing ovens, fired with pulverized coal, are in use, by means of which two anneals per week are possible.

The latest type (1937) of heat-treating unit is the radiant tube, batch or continuous oven with controlled atmospheres, which do not oxidize or decarburize. In these, because of the accurate temperature control, much shorter heat treatments are possible than in any other type. The castings are not packed in pots, but are laid on trays.

**PHYSICAL CHARACTERISTICS OF MALLEABLE CASTINGS.**—In connection with tensile properties, it is the practice of the Malleable Iron Research Institute (formerly the Am. Malleable Castings Assoc.) to submit to their consulting engineer for test, a standard test bar from each day's run. Based on the average results for the years 1928-1932, the approximate physical properties were: Ultimate strength, 54,000 lb. per sq. in.; yield point, 36,000 lb. per sq. in.; elongation, 19.00%. In 1931, the A.S.T.M. and the Am. Foundrymen's Assoc. published jointly a symposium on malleable iron castings, from which the average properties in Table 1 are quoted.

**The Electrical Properties** of malleable iron lie about midway between those of cast iron and cast steel. The permeability curve lies below that of cast steel and above that of cast iron. The electrical resistance of malleable iron is about 0.000044 ohm per cu. cm. compared with the resistance of cast iron of 0.000009 ohm per cu. cm. Malleable iron possesses high induction, and both a low hysteresis loss and coercive force.

**EFFECT OF SKIN ON STRENGTH OF MALLEABLE CASTINGS.**—A popular but erroneous belief is that the removal of the so-called skin of malleable iron so reduces its strength that the machined castings are more or less worthless. These conclusions must have been based upon experiments on bars that either were unsound, due to the presence of a central shrink, or were more or less worthless even with the skin on. Results of experiments made by the author and by others, prove conclusively that the removal of the skin does not affect the strength of malleable castings of good quality.

The following tests were made by Frazer & Jones, Syracuse, N. Y. A set of 4 bars each was cast from 4 consecutive heats. Each bar had the dimensions of the A.S.T.M. standard tensile test bar, excepting that 3 had diameters in the gage length of $^3/_4$, $^7/_8$ and 1 in., respectively, instead of the $^5/_8$-in. standard diameter. These bars were machined to a gage length diameter of $^5/_8$ in. Table 2 shows results of physical tests made on these bars by Halcomb Steel Co., Syracuse, N. Y. Only the gage length of the finished bars was machined and, therefore, it is practically certain that the machined surface of the test bar ends fitting the grips of the testing machine was not parallel with the machined gage length. Consequently, the pull on the machined bars was properly more or less eccentric and results are lower than if the bars had been properly prepared for tests.

Table 3 is a record of tests made by Robt. W. Hunt & Co., Chicago, on another set of bars cast by Frazer & Jones.

**EFFECT OF THICKNESS OF SECTION ON STRENGTH.**—Based on tests made on round sections only, it can be stated that as the diameter increases, the ultimate strength, yield point and elongation all decrease slightly. With rolled steel or forgings the same is true, except that the elongation is not lowered and may increase slightly. However, malleable iron has an advantage over the other two products, in that it is of equal strength in every direction, while with rolled or forged steel the mechanical properties are appreciably lower in samples cut crosswise of the principal direction of the flow of metal. No tests have been made on flats because, as in the case of steel also, the results are greatly influenced by each change in the ratio of thickness to width.

**SHOCK RESISTANCE OF MALLEABLE CASTINGS.**—Malleable cast iron has high shock-resisting properties, due to the grain size in every part of the casting being

**Table 1.—Average Properties of Malleable Iron**

From symposium of A.S.T.M. and A.F.A., 1931

**Chemical Composition**

C, 1.00 to 2.00; Si, 0.60 to 1.10; Mn, under 0.30; P, under 0.20; S, 0.06 to 0.15%.

**Physical Constants**

Specific gravity, 7.15 to 7.45.

Coefficient of thermal expansion, varies with temperature; average value, 0.000012 per deg. C. (0.0000066 per deg F.).

Specific heat, varies with temperature; average value from 20 to 100 deg. C., 0.122 calorie per gram per deg. C.

**Mechanical Properties**

| | |
|---|---|
| Tensile strength | 54,000 lb. per sq. in. |
| Yield point in tension | 36,000 lb. per sq. in. |
| Elongation in 2 in. | 18% |
| Modulus of elasticity in tension | 25,000,000 |
| Ultimate shearing strength | 48,000 lb. per sq. in. |
| Yield point in shear | 23,000 lb. per sq. in. |
| Modulus of elasticity in shear | 12,500,000 |
| Modulus of rupture in torsion | 58,000 |
| Yield point in torsion | 24,000 lb. per sq. in. |
| Brinell hardness number | 115; range 100 to 140 |

**Table 2.—Results of Tensile Tests on Machined Malleable Test Bars**

(Halcomb Steel Co., Syracuse, N. Y.)

| Bar Number | Heat Number | Gage Length, Diam. as Cast, in. | Gage Length, Diam. After Machining, in. | Amt. Stock Removed During Machining, percent | Ultimate Strength, lb. per sq. in. | Percent of Elongation in 2 in. | Bar Number | Heat Number | Gage Length, Diam. as Cast, in. | Gage Length, Diam. After Machining, in. | Amt. Stock Removed During Machining, percent | Ultimate Strength, lb. per sq. in. | Percent of Elongation in 2 in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 5–12–1 | $^5/_8$ | ...... | ...... | 54,000 | 24.00 | 1 | 5–13–1 | $^5/_8$ | ...... | ...... | 53,300 | 26.00 |
| 2 | 5–12–1 | $^3/_4$ | $^5/_8$ | 30.50 | 52,800 | 19.00 | 2 | 5–13–1 | $^3/_4$ | $^5/_8$ | 30.50 | 50,700 | 24.00 |
| 3 | 5–12–1 | $^7/_8$ | $^5/_8$ | 48.90 | 50,100 | 13.00 | 3 | 5–13–1 | $^7/_8$ | $^5/_8$ | 48.90 | 50,400 | 20.00 |
| 4 | 5–12–1 | 1 | $^5/_8$ | 60.90 | 50,400 | 13.50 | 4 | 5–13–1 | 1 | $^5/_8$ | 60.90 | 50,100 | 12.00 |
| 1 | 5–12–2 | $^5/_8$ | ...... | ...... | 54,100 | 25.00 | 1 | 5–13–2 | $^5/_8$ | ...... | ...... | 52,000 | 19.00 |
| 2 | 5–12–2 | $^3/_4$ | $^5/_8$ | 30.50 | 49,900 | 14.00 | 2 | 5–13–2 | $^3/_4$ | $^5/_8$ | 30.50 | 49,100 | 15.50 |
| 3 | 5–12–2 | $^7/_8$ | $^5/_8$ | 48.90 | 48,000 | 13.50 | 3 | 5–13–2 | $^7/_8$ | $^5/_8$ | 48.90 | 47,600 | 15.50 |
| 4 | 5–12–2 | 1 | $^5/_8$ | 60.90 | 46,750 | 13.00 | 4 | 5–13–2 | 1 | $^5/_8$ | 60.90 | 49,900 | 17.00 |

very small irrespective of the size of section, although slightly larger in the heavy than in the light sections. The shock-resistance is measured by the impact test used by the Malleable Iron Research Assoc. which was suggested by Benjamin Walker of the Erie Malleable Iron Co. The test can be quickly conducted and has the advantage of using cheap test specimens consisting of a wedge 6 in. long × 1 in. wide, tapering from $^{1}/_{2}$ in. at the base to $^{1}/_{16}$ in. at the thin end. The wedges are held upright at their base on the anvil of the testing machine and a weight of 21 lb. is dropped 3.33 ft. on the thin end of the specimen, causing it to curl; the procedure is continued until fracture occurs. The blows required to cause fracture are recorded and the length of the butt is measured. The length furnishes a comparative idea of the stiffness of the metal, *i.e.*, assuming that it required, in each case, 25 blows to cause two wedges to fail, the one having the longest butt would be the stiffer. This test is of extreme severity in that the blows must be resisted by sections that are very small, the extreme base of the wedge being but $^{1}/_{2}$ sq. in., while the first blow is resisted by a section only slightly larger than $^{1}/_{16}$ sq. in. If the wedge does not fail under 30 blows, the testing is stopped on the ground that a wedge withstanding 30 blows is A-1 material, and to prolong the test would serve no useful purpose.

The foundries make no use of the notched bar test, first because there is no agreement among mechanical engineers as to what actually is shown by this test, and second, because malleable iron castings are not notched. Cast specimens for the notched bar test are difficult so to gate that they will be absolutely sound. If even incipient unsoundness is present, the results will not be trustworthy. On the other hand, the wedges used for the wedge test can be cast sound; they are not notched, and the test is a fair one as indicating the toughness of the metal.

**Machining Properties of Malleable Castings.**—The fact that malleable iron can be machined at depths of cut, feeds, and speeds not possible in the case of any other ferrous metal of equal mechanical properties is freely acknowledged. While a few manufacturers of malleable iron have advanced the claim that a high tensile product cannot be machined quite as rapidly as one of lower strength, the author has attempted to determine by service tests during the years 1928 to 1932, if there was any justification for this statement. A hundred castings of low, medium and high carbon were machined at an automobile plant. Although the machine tool was run exactly in accordance with the usual practice in machining this particular casting, the machinist was unable to differentiate between the three grades. According to L. J. Kelly (*Trans.* A.F.A., 1930, p. 242), the following were the machining operations on 2 $^{1}/_{4}$-in. grease plugs cast from a hard iron of about

**Table 3.—Results of Tensile Tests on Machined Malleable Test Bars**

(R. W. Hunt & Co., Chicago, 1921)

| Date, 1920 | Bar Number | Heat Number | Diameter of Bar as Tested, in. | Diameter of Bar as Cast, in. | Elastic Limit, lb. per sq. in. | Ultimate Strength, lb. per sq. in. | Percent of Elongation in 2 in. | Reduction of Area, percent |
|---|---|---|---|---|---|---|---|---|
| Nov. 16 | 1 | 1 | 0.625 | $^5/_8$ | 35,410 | 55,720 | 29.0 | 30.8 |
| | 2 | 1 | .625 | $^3/_4$ | 35,350 | 55,380 | 22.5 | 22.6 |
| | 3 | 1 | .625 | $^7/_8$ | 34,950 | 54,980 | 21.5 | 21.7 |
| | 4 | 1 | .623 | 1 | 35,800 | 55,140 | 19.5 | 16.9 |
| | 1 | 2 | .623 | $^5/_8$ | 35,040 | 55,880 | 25.0 | 26.2 |
| | 2 | 2 | .625 | $^3/_4$ | 35,310 | 54,840 | 20.5 | 19.7 |
| | 3 | 2 | .623 | $^7/_8$ | 34,980 | 53,740 | 15.5 | 14.2 |
| | 4 | 2 | .625 | 1 | 34,500 | 53,600 | 14.5 | 13.0 |
| Nov. 17 | 1 | 1 | .625 | $^5/_8$ | 34,600 | 56,580 | 26.0 | 30.8 |
| | 2 | 1 | .623 | $^3/_4$ | 35,760 | 55,580 | 24.0 | 24.9 |
| | 3 | 1 | .623 | $^7/_8$ | 36,320 | 55,460 | 24.0 | 20.5 |
| | 4 | 1 | .623 | 1 | 35,370 | 54,820 | 24.0 | 22.1 |
| | 1 | 2 | .625 | $^5/_8$ | 35,220 | 55,440 | 26.0 | 29.4 |
| | 2 | 2 | .623 | $^3/_4$ | 35,960 | 55,260 | 23.0 | 24.3 |
| | 3 | 2 | .625 | $^7/_8$ | 35,310 | 54,420 | 20.0 | 21.1 |
| | 4 | 2 | .623 | 1 | 35,140 | 54,540 | 23.0 | 22.6 |
| Nov. 18 | 1 | 1 | .625 | $^5/_8$ | 35,310 | 55,200 | 26.5 | 20.8 |
| | 2 | 1 | .623 | $^3/_4$ | 35,340 | 55,380 | 22.5 | 23.5 |
| | 3 | 1 | .623 | $^7/_8$ | 35,340 | 55,020 | 21.5 | 19.2 |
| | 4 | 1 | .623 | 1 | 35,370 | 53,840 | 24.5 | 22.9 |
| | 1 | 2 | .630 | $^5/_8$ | 33,720 | 55,120 | 26.0 | 29.2 |
| | 2 | 2 | .624 | $^3/_4$ | 35,120 | 54,840 | 26.0 | 25.1 |
| | 3 | 2 | .624 | $^7/_8$ | 34,310 | 55,240 | 23.0 | 23.2 |
| | 4 | 2 | .623 | 1 | 34,650 | 54,920 | 24.0 | 22.6 |

2% carbon: Two cuts to remove $1/4$ in. of stock; an undercut both front and back in one operation, after which the plug was threaded for a distance of $5/8$ in. As to tool life, 500 were machined every 22 hours (day and night) with but one grinding of tools; 20,000 were threaded with but three grinds of the chasers, the grinding necessary being very slight.

The author has been unable to find, during a 5-year investigation, one instance where the piece rate, set by rate-setters, has had to be altered, when the source of supply was changed from a plant making low-strength castings, to one making a high-strength product. If there is any difference, it apparently cannot be ascertained under service conditions, and no worthwhile proof has been furnished that it has been ascertained in any other manner.

Poor machinability of malleable castings results from: 1. A deep decarburized skin; such a skin mineralogically is very soft but is hard to machine, being similar in character to pure copper and pure aluminum. 2. A skin consisting of pearlite or of a layer of pearlite and ferrite, or of pearlite and free cementite, will be hard to machine, because of mineralogical hardness. The metal under the skin may be easy or difficult to machine, being easy in the majority of cases. If, however, during the anneal, the product has been heated to too low a temperature or to too great temperature for too short a time, or both, the structure will contain considerable free cementite and, therefore, will be hard to machine, depending on how seriously the correct laws of annealing have been violated. If the annealing oven has been incorrectly designed, the temperature not being uniform throughout, castings that are easy and others that are difficult to machine may be obtained from the same oven. Further, certain hard iron compositions cannot be annealed in such a manner as to yield a structure that will machine with ease. Also, annealing carried out at too high temperature will make the product difficult to machine, due to the formation of pearlite, under certain conditions, in the structure.

**HARDNESS TESTS OF MALLEABLE CASTINGS.**—Whether or not malleable iron, when machined, will test efficiently under the impact or tensile test, depends entirely on whether or not the test specimen is sound. Due to improvement in molding operations, the danger of unsound castings has largely been eliminated. Whether or not a product will machine easily or with difficulty is of first importance. Only in exceptional cases can mineralogical or cutting hardness be satisfactorily determined by the Brinell or Shore instruments. The reason for this is that during the anneal, the structure of the surface of the casting to a depth of from $1/32$ to $1/12$ in. or more will be dissimilar to the structure of the body of the metal underlying the so-called skin, which, in the majority of cases, consists of decarbonized iron, providing the pot atmosphere is oxidizing. The skin, however, may vary in structure, depending on the pot atmosphere. Under certain conditions, a skin structure consisting of 2 layers may be obtained. The outer one of these consists largely of pearlite, with perhaps some free cementite, and will be extremely hard. Between this layer and the body of the casting will be a ring of carbonless iron. This effect usually is produced through the action of an oxidizing pot atmosphere which decarbonizes the surface to a certain depth, followed by a change in the pot atmosphere from oxidizing to reducing. Under these conditions part or all of the decarbonized layer will be recarbonized, the depth depending upon the length of time the castings are exposed to this action.

In other cases, the skin will consist of an outer ring of decarbonized iron and an inner ring of pearlite and at other times of a skin consisting wholly of pearlite occasionally accompanied by considerable free cementite. The last two cases produce a fracture that is known as " picture frame " iron. The fracture shows a more or less brilliant steely frame surrounding the section and separated from it by a distinct line of demarcation. It is this character of iron that probably originated the term "black heart malleable iron" since in the early days most of the malleable iron had this type of fracture. Some writers on the subject of malleable cast iron, both in the United States and England, have presented fractures of " picture-frame iron " as representing a superior product, notwithstanding the fact that castings of such fracture would be worthless when subjected to bending or impact, the frame being hard and brittle and actually inviting fracture. It is therefore obvious that an attempt to ascertain hardness by means of either the Brinell testing machine or the Shore scleroscope would be futile. In the case of Brinell test, the ball would crush through the skin, irrespective of its character, leaving its impression partly in the skin and partly in the metal beneath. In the case of the scleroscope, the proper preparation of the specimen would involve the removal of most, if not all, of the skin.

**SHRINKAGE.**—Shrinks in castings are harmful not only because of their influence in decreasing the strength of the casting, but also due to the formation of hard spots wherever the shrink exists, which will cause trouble in machining. With few exceptions, castings can be made perfectly sound by means of large risers and heads properly located to feed the casting. With intricate patterns, it may be impossible to avoid the use of an occasional chill, due to the inability to attach a riser or a sink head in such a manner as to insure the proper feeding. The remedy lies in a change of design to meet this condition. Detection of shrinks is a most important function of the inspector, aside from confirming the strict trueness to pattern of the castings. Heavy shrinks, and even those which are less pronounced, can be detected easily by the breaking up of hard iron castings and a careful inspection of fractures at sections where shrinkage is liable to occur; that is, at fillets and at junctions of light and heavy sections. Very light shrinkage cannot be detected by this method, as such become visible only when the casting has been annealed. If castings are purchased from one of the many concerns which has paid equal attention to its furnace and annealing oven design and operation, a guarantee probably can be obtained with the product that it is free from shrink, even in cases of most complicated design. If, on the other hand, the foundry has not informed itself on the best modern practice, the product is liable to vary in composition, physical properties and machinability, while shrinks will be of rather common occurrence.

**INSPECTION.**—Irrespective of where castings are purchased, inspectors should first examine the test lugs placed upon them for that purpose. If castings are too small to warrant placing of test lugs, some castings should be broken for fracture examination as well as to test their toughness and ductility. When malleable iron fails in tension, the crystalline grains at the fracture stretch into spines, and if the product is good, the appearance of the fracture will be uniform throughout, except that there will be a slight contrast between the decarbonized surface and core. The color will depend largely upon how ductile the crystalline grains may be as well as upon composition. If the grains are very ductile and the carbon is low, the fracture will be mouse-colored. The higher the carbon content the darker will be the fracture. In correctly annealed malleable iron, if the silicon is in the vicinity of 1.10% or more, and manganese much in excess of 0.30%, or if both conditions exist, the fracture, while normal in appearance, will be more or less sparkling. This characteristic, however, is much more evident in fractures resulting from impact or when a bar is broken transversely than when fracture takes place from tension.

In the breaking of a test lug, each lug should receive as nearly as possible the same treatment. If the lug is repeatedly hit in such a manner that it will be sheared off at its base, the fracture will have a different appearance than if the blows had gradually bent the lug over and broken it in a manner similar to a transverse test. In the latter case, the fracture on the compression side will be bright to a certain depth from the surface, due to the grains having been smashed down to destruction on that side, while the side that broke in tension will show a fracture similar to what would be expected in a tension test. High carbon or high manganese or both will make the bright part on the compression side deeper. In certain cases there may be a slight shrinkage at the fillet of the lug which may affect the appearance of the fracture. If the test lugs are broken hot, that is, just as the castings are dumped from the pot, an abnormal close-grained dove-colored fracture will result that is not truly representative of the metal. Before he can gather all the information that is possible from the test lug inspection, the inspector must have had considerable experience.

A casting may be sound and the fracture abnormal, that is, the fracture may be of the picture-frame variety. It may be of steely appearance or have other faulty fracture characteristics. In such cases, the tensile test will indicate that the material is not first class. The cause of this condition usually can be ascertained by metallographic means, although sometimes a chemical analysis is also necessary. "Picture-frame" iron usually results from too low a silicon, carbon and manganese content, or from an unbalanced sulphur-manganese ratio. Steely iron can result from a low silicon or manganese content, combined, or when either is extremely low. If castings have been cooled from the annealing temperature at too fast a rate, an all-white or whitish fracture will result, the appearance depending upon the rate of cooling. The same holds true if the time of annealing has been too short or if it has been conducted at too low temperature. The cause of the trouble usually is indicated by the appearance of the fracture, or if the castings have not been held long enough close to, but under, the critical range.

**LOCATION OF TEST COUPON.**—Test pieces cut from different parts of castings of any metal are liable not to measure up to the standards of standard test bars cast from the same heat. Much, therefore, depends on the location on the casting from which the test coupon is cut. But few investigations have been conducted by the author in this connection. Such tests are hardly fair, as the specimens obviously must be cut from parts that favor solidity and from which shrink is usually absent, and also on account of the fact that the test specimens ordinarily must be of different shape than the test bar with which they are compared. In two cases, a $^1/_2$-in. round specimen was secured from a web and showed an ultimate strength about 2000 lb. lower than the standard test piece, the elongation in one case being a little in excess and in the other a little lower than the test piece.

In another case, two samples of rectangular section $^3/_8 \times {}^3/_4$ in. at gage length, machined on all sides, the ultimate strength and elongation were both slightly higher than the standard test piece. One of the principal manufacturers of railway castings reports as follows: "In order to convince ourselves that sections cut from the casting would meet specifications, two column-castings were selected, and from the cope side, coupons 12 in. long were cut with the following results":

| Sample Mark | Dimensions, in. | Area, sq. in. | Ultimate Strength, lb. per sq. in. | Elongation in 2 in., % |
|---|---|---|---|---|
| No. 1 | 0.306 × 0.825 | 0.252 | 47,500 | 11.00 |
| No. 2 | 0.340 × 1.03 | 0.350 | 48,200 | 11.00 |

**SPECIFICATIONS FOR MALLEABLE CASTINGS.**—The following is an abstract of A.S.T.M. Standard Specifications (Designation A-47-33) covering two grades of malleable iron castings:

Castings shall be produced by either the air furnace, open-hearth or electric furnace process, and be free from primary graphite. Tension specimens shall be of the form and dimensions shown in Fig. 1, and shall conform to the following *minimum* requirements as to tensile properties:

| | Grade No. 32510 | Grade No. 35018 |
|---|---|---|
| Tensile strength, lb. per sq. in. | 50,000 | 53,000 |
| Yield point, lb. per sq. in. | 32,500 | 35,000 |
| Elongation in 2 in., percent | 10.0 | 18.0 |

The yield point is defined as that load under which the specimen has an elongation in 2 in. of 0.01 in., and may be determined by the drop of the beam of the testing machine or by the divider method.

A set of three tension test specimens shall be cast from each melt, suitably marked for identification with the melt. The specimens shall be placed in some one oven containing castings to be annealed. Three of these specimens shall be selected by the inspector as representing the castings in the oven from which they were taken. The castings are accepted if the first specimen conforms to requirements, or in the event that it fails both the second and third specimen shall meet requirements. If castings are rejected, due to insufficient anneal, a reanneal is permitted. The castings shall conform to patterns or drawings (a variation of $1/_8$ in. per ft. will be permitted). Injurious defects must be absent.

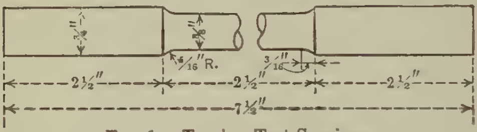

Fig. 1. Tension Test Specimen

**CORROSION RESISTANCE.**—Messrs. Wolf and Meisse (Symposium, A.S.T.M. and A.F.A., 1930) state:

1. Under the action of locomotive smoke, malleable iron shows a decided superiority over wrought-iron, basic open-hearth steel, and commercial pure iron. 2. Copper additions in malleable iron up to 2% have been found to materially increase its resistance to corrosion in locomotive smoke. 3. Copper additions to malleable iron decrease its resistance to corrosion in acid mine water. 4. Malleable iron without copper, protected by a hot-dipped galvanized coating, shows a decided superiority over steel similarly protected, after a service of 17 years in an atmosphere contaminated by smoke. 5. A hot-dipped galvanized malleable iron, with and without copper, shows a heavier iron-zinc alloy layer than either wrought-iron, basic open-hearth steel, or commercial pure iron, when galvanized under identical conditions.

Inasmuch as accelerated laboratory tests have been discredited as reflecting what might be anticipated under service conditions, coupled with the fact that field tests require years in order to secure results, the Malleable Iron Research Institute has endeavored to obtain sufficient satisfactory evidence, of a character that could not well be refuted, through examination of existing built-up parts that have been exposed for many years under various conditions. To date (1937) numerous examples have been photographed, among them heavy fences built of steel pipe with malleable iron fittings, that show, after 20 years of exposure to the atmosphere, the steel rusted through in many places and the fittings remained unaffected. Numerous other examples could be cited of steel pipe laid underground, the steel being corroded through while the malleable fittings show a surface corrosion only.

# NON-FERROUS FOUNDRY PRACTICE

By N. K. B. Patch

The production of bronze or brass, or other non-ferrous castings under modern (1937) requirements for their satisfactory service demands much higher specialization in this branch of the foundry industry than was the case only a few years ago. Metallurgical research and the development of metals of high purity both have influenced improvement in quality and uniformity of this type of castings. Skill of an increasingly higher order is demanded in order to insure the right kind of product. The old-fashioned brass foundry, commonly a corner in the iron foundry, casually attended by a practical man, who, by his own admission, was " skilled in the arts of metal " is passing. Instead, castings are purchased from foundries organized to produce uniform castings from the highest quality of modern metals.

**GENERAL.**—Non-ferrous castings being produced of materials of relatively high cost, foundry practice should be much more precise than where the metal is cheap and where much of it may be wasted without undue loss in machining. Hence, patterns for non-ferrous castings generally are made with the minimum allowance for machining. Molders should be trained to rap patterns only just enough to remove them from the sand, and also should be trained to make the molds as accurate to the pattern as a high degree of skill will permit. Uniformity of wall thickness, accuracy of casting to pattern and similar characteristics all are desirable in non-ferrous castings, not only to save the high-priced metal, but also to standardize all the different elements connected with the production of castings to a point that insures uniform quality and a material reduction in defective castings. These elements include pouring temperature, melting practice, cores and core mixtures, and the time of shaking the castings out of the mold.

## 1. MELTING EQUIPMENT

In the brass foundry, melting equipment is of first importance. Improper melting is the cause of many defective castings that heretofore have been attributed to bad molding or other causes.

**TYPES OF FURNACES.**—Three general types of furnaces regularly are used:

1. Crucible furnaces in which the metal is contained in crucibles during melting and pouring.
2. Hearth furnaces, burning solid fuel, oil, or gas, and in which the metal is melted on a hearth.
3. Hearth furnaces, which use the heat radiated from an electric arc to melt the metal.

The crucible furnace is the more common, since it entails a much smaller original investment that the other types and requires less experience and study for proper handling. The source of heat in crucible furnaces may be oil, gas, coal or coke. Furnace manufacturers should be consulted concerning the proper type of equipment for a given set of melting conditions. In small foundries producing castings weighing up to 10 lb., crucibles holding 100 to 150 lb. per charge are the most common. These require only a medium-size furnace and are readily carried from furnace to mold. For large castings, larger charges are melted and larger crucibles are necessary. Overhead equipment then must be provided to move the molten metal from furnace to mold.

Fig. 1 shows the usual arrangement of a battery of pit furnaces for crucibles. Dimensions are given in Table 1. The thickness of the refractory lining used will vary, but generally is 4 1/2 in. Many foundrymen prefer to line the iron shell with an insulating material before inserting the refractory lining. Flue sizes must be increased above those given in Table 1, if the distance from the furnace to the main flue tends to retard easy flow of gases. Where crucibles smaller than No. 20 are used, it frequently is more convenient to use a furnace that will accommodate two or three crucibles, or more. Above No. 20 size, only one crucible is inserted in the furnace, and is surrounded with the fuel. Table 2 gives dimensions of standard crucibles.

Gas or oil commonly is used as fuel, although coke or coal sometimes is used. If coal or coke is to be used for the production of heat, a well designed flue from the furnace to the chimney must be provided with adequate draft to insure complete combustion of the fuel. Such draft may be developed by a chimney of adequate height; or if furnaces are designed to accommodate it, forced draft may be used to good advantage.

Crucible furnaces of the tilting type are used in many foundries, with oil or gas as fuel. See Fig. 3. The fuel-air mixture is introduced into the furnace through a hole in

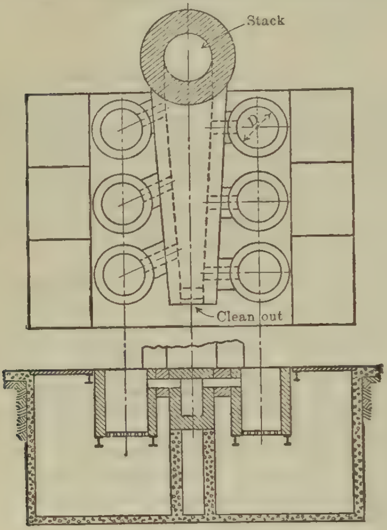

Fig 1. Arrangement of Pit Furnace for Crucibles

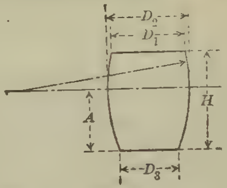

Fig. 2. Standard Crucible. See Table 2

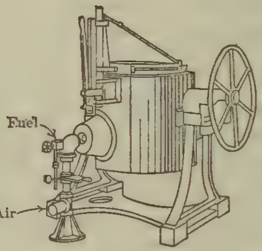

Fig. 3. Crucible Tilting Furnace

**Table 1. Pit Furnace Sizes, Flues and Crucibles.** (See Fig. 1)

| | | | | |
|---|---|---|---|---|
| Diameter of furnace *D*, in | 24 | 32 | 36 | 40 |
| Approximate flue size, in | 1×4 | 4×4 | 4×6 | 5×7 |
| Crucible generally used, No | 10–25 | 25–80 | 60–125 | 100–400 |

**Table 2.—American Standard Plumbago Crucibles for Non-tilting Furnaces**
A.S.A. Tentative Standard H 13–1925. All dimensions in inches. See Fig. 2

| Size Number | *H* | *A* | $D_1$ | $D_2$ | $D_3$ | Size Number | *H* | *A* | $D_1$ | $D_2$ | $D_3$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 8 1/16 | 5 1/4 | 6 1/16 | 6 9/16 | 4 13/16 | 70 | 15 1/16 | 9 13/16 | 11 1/4 | 12 3/16 | 8 15/16 |
| 12 | 8 1/2 | 5 1/2 | 6 3/8 | 6 7/8 | 5 1/16 | 80 | 15 5/8 | 10 1/8 | 11 11/16 | 12 11/16 | 9 1/4 |
| 14 | 8 7/8 | 5 3/4 | 6 11/16 | 7 3/16 | 5 1/4 | 90 | 16 3/16 | 10 1/2 | 12 1/8 | 13 1/8 | 9 9/16 |
| 16 | 9 1/4 | 6 | 6 15/16 | 7 1/2 | 5 1/2 | 100 | 16 11/16 | 10 7/8 | 12 1/2 | 13 1/2 | 9 7/8 |
| 18 | 9 13/16 | 6 3/8 | 7 5/16 | 7 15/16 | 5 13/16 | 125 | 17 3/8 | 11 5/16 | 13 | 14 1/16 | 10 5/16 |
| 20 | 10 5/16 | 6 11/16 | 7 11/16 | 8 3/8 | 6 1/8 | 150 | 18 3/8 | 11 15/16 | 13 3/4 | 14 7/8 | 10 7/8 |
| 25 | 10 15/16 | 7 1/8 | 8 3/16 | 8 7/8 | 6 1/2 | 175 | 19 1/4 | 12 1/2 | 14 3/8 | 15 9/16 | 11 3/8 |
| 30 | 11 1/2 | 7 1/2 | 8 5/8 | 9 5/16 | 6 13/16 | 200 | 20 | 13 | 15 | 16 1/4 | 11 7/8 |
| 35 | 12 | 7 13/16 | 9 | 9 3/4 | 7 1/8 | 225 | 20 3/4 | 13 1/2 | 15 1/2 | 16 13/16 | 12 5/16 |
| 40 | 12 1/2 | 8 1/8 | 9 3/8 | 10 1/8 | 7 7/16 | 250 | 21 3/8 | 13 7/8 | 16 | 17 5/16 | 12 11/16 |
| 45 | 13 3/16 | 8 9/16 | 9 7/8 | 10 11/16 | 7 13/16 | 275 | 22 | 14 5/16 | 16 7/16 | 17 13/16 | 13 |
| 50 | 13 3/4 | 8 15/16 | 10 1/4 | 11 1/8 | 8 1/8 | 300 | 22 1/2 | 14 5/8 | 16 7/8 | 18 1/4 | 13 3/8 |
| 60 | 14 7/16 | 9 3/8 | 10 13/16 | 11 11/16 | 8 9/16 | 400 | 24 5/16 | 15 13/16 | 18 3/16 | 19 11/16 | 14 7/16 |

**Table 3.—Dimensions and Capacity of Tilting Crucible Melting Furnaces**
Air pressure, 16 oz. Oil pressure, 15 lb.
(Monarch Engineering & Manufacturing Co., Baltimore, Md.)

| Crucible No. | Capacity, lb. | Oil, gal. per hr. | Gas, cu. ft. per hr. | Average Time for Melt, min. | Air, cu. ft. per hr. | | Pipe Size, in. | | | Floor Space, in. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Forced Draft | By Induction | Oil | Gas | Air | Long | Wide | High |
| 125 | 375 | 10 | 3000 | 30 | 12,000 | 5,500 | 3/8 | 1 | 2 | 60 | 67 | 66 |
| 150 | 450 | 12 | 3600 | 30 | 14,400 | 7,000 | 3/8 | 1 | 2 | 64 | 72 | 68 |
| 275 | 825 | 16 | 4650 | 45 | 19,200 | 9,300 | 3/8 | 1 | 3 | 70 | 70 | 70 |
| 400 | 1200 | 18 | 5300 | 60 | 21,600 | 10,500 | 1/2 | 1 | 3 | 76 | 72 | 74 |

the side, from a stationary burner, and combustion is completed within the furnace. When the charge is melted the entire furnace is tilted to pour the metal. Dimensions and capacity of one type are given in Table 3.

Fig. 4 shows one form of double-chamber hearth furnace of the tilting type. Capacities of these furnaces range about as follows:

| | Single Chamber | | | | | | | Double Chamber | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Furnace No. | 92 | 93 | 94 | 95 | 96 | 97 | 98 | 1 | 2 | 3 | 4 |
| Capacity, lb. | 500 | 1000 | 2000 | 4000 | 6000 | 12,000 | 24,000 | 250* | 500* | 1000* | 2000* |

* Each chamber.

**LININGS.**—The refractory linings used in those furnaces in which the metal actually makes contact with the lining should be of such material as will reduce to an absolute minimum the chemical reaction between metals, metal oxides and refractory at elevated temperatures. An acid refractory will react with many of the metal oxides to form a slag, and thus will be of relatively short life. Hence, a basic or neutral refractory is to be preferred, and will, if carefully chosen, give maximum life and consequent maximum economy in cost per pound of metal melted. It should be remembered that the different types of fuel affect the amount of reaction of this kind that occurs, since the percentage of reducing or oxidizing gases in the furnace materially affects the rapidity with which the chemical action takes place.

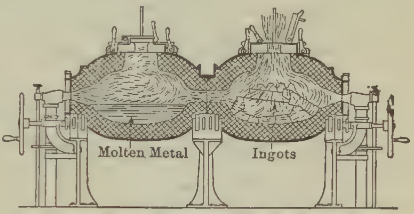

Fig. 4. Double Chamber Tilting Furnace

**MELTING TIME.**—Good practice dictates that melting must take place as rapidly as economy and good combustion will permit. This reduces the danger of gas inclusions in the metal, with resultant defective castings, and results in economy of fuel consumption. A crucible containing 150 lb. should be heated to proper pouring temperature in not over one hour, and preferably in about 45 minutes.

**CHARGING.**—The crucible should be charged first with a few gates or risers or small pieces of the metal to be melted. The larger pieces are charged on top of them. The small pieces melt first and form a puddle in the bottom of the crucible which rapidly dissolves the balance of the charge, thereby increasing the speed of melting.

**MELTING OF LIGHT ALLOYS.**—Light alloys, as aluminum- or magnesium-base alloys are melted in a cast-iron or steel bowl. These bowls are so located that the hot gases produced by the combustion of the fuel flows around them. The metal is dipped from the bowl with small ladles to be transported to the molds. The surface of the metal is kept covered with the proper type of flux, and the temperature (see Table 3) of the metal is controlled carefully by pyrometers, so located as to indicate at all times the exact temperature of the bath of metal. The furnace operator then may control the fire to insure the desired uniformity of temperature. These alloys dissolve iron from the bowl, and should not be permitted to remain therein long enough to absorb more than the maximum permissible Fe content.

To insure rapid changing of the contents of the bath of metal, the proper size of bowl must be chosen with relation to the amount of metal demanded for the kind of work produced. Where large castings are made on a mass production basis, larger bowls are used, but the metal is dipped out so fast that it is being constantly changed by the replacement of fresh metal. Consequently the metal is in contact with the iron only for a very short time. Where smaller castings are produced, and particularly where fewer molds are being poured, the demand for metal is materially less, charges are smaller, and smaller bowls are used. The metal thus is changed practically as frequently as in the first instance.

**ELECTRIC FURNACE PRACTICE.**—In the production of bronze or brass castings by means of electric furnaces, the correct size furnace for the requirements of the product is important. If castings weighing 150 lb. or more are to be regularly made, a 1-ton furnace is necessary. For castings of smaller sizes, a 250-lb. furnace will be found the most desirable. Where a variety of alloys is to be produced, several furnaces are absolutely necessary, so that a different hearth may be available for the different types of alloys. An alloy from which zinc is to be excluded should not be melted on a hearth that has previously melted an alloy containing much zinc. Aluminum-bronze, for example, should not be melted on a hearth where alloys containing lead, tin, or zinc have been melted previously; nor should phosphor-bronzes be melted on hearths previously used for aluminum-bronzes. All contamination must be avoided. This is one reason for the wide use of crucible furnaces, since they permit a crucible to be used exclusively for a given alloy. Where a change of alloy is desired, it is, therefore, only necessary to change the crucible, reserving each crucible for its own specific alloy.

Manufacturers of electric furnaces should be consulted concerning the selection of the proper type or size for melting non-ferrous alloys. The kind of work influences these details, as well as the kind of alloys to be melted and the weight of melted metal required per hour. The cost of electricity also is an important factor, since in some localities the cost of current is so high that the operating cost of electric furnaces far exceeds that of fuel-fired furnaces using oil or gas. On the other hand, in some localities electric current costs are so low that the electric furnace is the ideal unit. Natural gas, when available at very low figures is the preferred fuel, if fuel cost is the primary requisite. If, however, the quality of the metal resulting from the melting process is the controlling factor, a careful survey of the availability of the different types of fuel is advisable. The cost of each laid down, available for use in the furnace, the cost of electric current, including the installation of the necessary transformers, etc., and the kind of alloys to be melted all should have careful consideration before the final choice of melting unit is made.

## 2. MELTING TEMPERATURES

The wide variety of alloys demanded from the non-ferrous foundry prevents precise definition of proper pouring temperatures. A variation in the proportions of the different elements in an alloy changes its fluidity at a given temperature, *i.e.*, changes the proper pouring temperatures.

**INFLUENCE OF COMPOSITION ON MELTING TEMPERATURE.**— There are a number of types of alloys commonly used in the modern brass foundry. One of these, known as the 85–5–5–5 type, is specified as 85% Cu, 5% Sn, 5% Pb, 5% Zn. As produced in different plants, it often varies from these proportions. Copper may range from 86% or 87% to 82% or 83%. This change in copper content, the element of highest melting point in the alloy, would necessarily change the temperature at which the metal is sufficiently fluid to pour a given class of work. The presence of oxides entangled in the bath of molten metal also influences fluidity.

Another alloy is the 80–10–10 type, specified as 80% Cu, 10% Sn, 10% Pb. This alloy is supposed to have no zinc. However, in the regular product of many plants some zinc not only is allowed, but, in some instances, is demanded. Copper is allowed as high as 83% or 84%, or as low, in some instances, as 75%. Tin is maintained relatively close to 10%, so far as maximum is concerned, but its minimum limit is frequently as low as 7%. Since tin is the high-priced ingredient, the maximum generally is preserved and the minimum is allowed to creep rather low, in order to make an imaginary saving. Zinc frequently is permitted to make up for a reduction in tin, although it does not accomplish the same end, particularly where a hard bearing crystal is demanded. The pouring temperature of this alloy, therefore, will vary within a considerable range because of the permissible changes in the composition as produced in different plants.

Manganese-bronze, which is an alloy of copper and zinc, with some iron, aluminum, tin and manganese, is allowed to change within rather marked limits. Therefore, the pouring temperature also may vary for this type of alloy. Since this alloy contains about 40% Zn, and consequently much less copper than the two types of alloys previously discussed, the pouring temperature is much lower than the other two.

Aluminum-bronze is an alloy of copper and aluminum, in which copper predominates, but in which the aluminum content may range from 5% or 6% to as high as 11%. Iron may range from as high as 4% or 5% to approximately 1%, or none. The presence of iron tends to raise the melting point. The higher the aluminum content, the lower will be the melting point. Consequently, the pouring temperature again will vary within fairly wide limits for this type of alloy. In the aluminum-base alloys there are so many combinations of elements with aluminum that are regularly poured, that to prescribe a precise pouring temperature is impossible. The same is true of magnesium-base alloys.

It is evident that the proper pouring temperature for each non-ferrous alloy depends on the metallurgical control exercised in maintaining uniformity of alloy. Foundries producing, for example, 85–5–5–5 alloy that is relatively accurate to the specification, can, with very little experimenting, determine the preferred pouring temperature for the alloy necessary for the type of work they are required to produce. Hence, good foundry practice demands metallurgical experience and study to handle the different alloys and to pour them at their proper temperature in order to thereby secure the most desirable characteristics and the minimum number of defective castings.

It is recommended that standardization of the type of alloy to be used in a given foundry be carried as far as possible. Where special alloys are required from a foundry regularly producing a standard alloy, the foundry frequently can purchase advantageously

**Table 4.—Pouring Temperature Ranges for Various Types of Alloys**

| Type of Alloy | Range of Pouring Temperature |
|---|---|
| 85 Cu, 5 Sn, 5 Pb, 5 Zn | 1950 to 2300° F. (See Note 1) |
| Leaded bronze without zinc, 80 Cu, 10 Sn, 10 Pb type | 1500 to 2050° F. (See Note 2) |
| Manganese-bronze type | 1830 to 2100° F. (See Note 3) |
| Aluminum-bronze type | 1950 to 2250° F. (See Note 4) |

Note 1. Where this composition is departed from materially, the pouring temperature necessarily changes, but a little experience will soon indicate the desired temperature for the work in hand.

Note 2. With this type of alloy, varying as it does from 30% Pb down to 10% Pb, or even less, and sometimes containing a little zinc, the pouring temperature necessarily must range within the wide limits indicated. When it contains the upper limit of lead, the pouring temperature is lower than where the lead content is less or where some zinc is present.

Note 3. The pouring temperature of this alloy varies within the limits indicated because there are wide variations of mixtures known as manganese-bronze. See pp. 4–18 to 4–23. Some of these contain 2% Fe and some only 1% Fe. Some contain upwards of 3% Al, some only a fraction of 1% Al. The temperature for the range indicated, however, will cover satisfactorily pouring temperatures for the range of alloys in this class.

Note. 4. The aluminum-bronze standard formulas call for a range of alloys containing varying amounts of iron and aluminum (see pp. 4–18 to 4–23), and as in the case of the manganese-bronzes, the variation in the content of these materials changes the melting point and consequently the desired pouring temperature. The range given, however, will be found to be reasonably satisfactory for the average class of work.

such castings from another foundry in which these special alloys are standard and which, therefore, is better equipped to produce the highest quality of special-alloy casting.

In general, the pouring temperatures of Table 4 are for average classes of work. For castings with very thin sections, say up to 1/8 in. thick, of course, the higher pouring temperature is necessary. Where this condition exists, pouring temperatures may be reduced by providing a number of gates. In any event, pouring temperature should be no higher than is absolutely necessary. Likewise, with very heavy sections or large castings, say over 25 lb., with consequently greater mass, pouring temperatures may in some instances be even lower than those in Table 4.

## 3. MOLDING

Molds for non-ferrous castings are made by much the same process as for gray iron castings. The sand should be somewhat finer. Standard No. 00 Albany molding sand is almost universally used for the production of medium- or light-weight sand castings of brass or aluminum-bronze. However, molds for non-ferrous castings should not be rammed as hard as those for gray iron. (See p. 20–33.) Venting should be very carefully done to insure the free exit of all gases from the mold.

**CORES** used for non-ferrous castings are quite different from those used for gray iron. The higher temperature at which gray iron is poured tends to burn bonds that are only charred by the lower temperature of the non-ferrous alloys. Further, the shrinkage of many non-ferrous alloys is much greater than that of the usual gray iron alloys. Consequently, to avoid stressing castings to the cracking point during solidification and cooling, cores must permit the castings to shrink. The core binders used, therefore, should be those which will hold the sand together satisfactorily, and yet will break down sufficiently under available temperature to allow the cores to give way as shrinkage stresses develop. Cores for gray iron castings are very hard, but the binder is well burnt by the elevated temperature of the iron. Similar cores for bronze or brass castings, and even more so for aluminum castings, would be almost impossible of removal from the castings. Also they probably would not be sufficiently affected by the temperature of the metal to destroy the bond and permit the casting to shrink.

Cores for non-ferrous castings, particularly of bronze and brass, generally are made from sands mixed with a very small portion of oil binder or molasses binder, or similar type of binder. See Core Mixtures, p. 20–33. If the cores are well baked, and the absolute minimum amount of binder having been used, a minimum amount of heat will affect the binder and reduce the core to loose sand after having been subjected to the temperature of the molten metal. Where heavy walls of non-ferrous metal surround the cores, the quantity of heat available obviously is such that combustion of the core binder is much more vigorous. Consequently, the core should be somewhat harder. Where heat is available to attack the core sufficiently to cause it to scab or give trouble, the cores frequently are coated with a graphite wash, as in gray iron practice. The wash should not be thicker than is absolutely necessary. It preferably should not cause flowing by reason of having been thick, or so closely bonded, as to preclude the passage of gases through it to the core vent.

**SHRINKAGE.**—Many non-ferrous alloys are extremely brittle at a red temperature, or at the temperature at which they are just set, but before they have reached their full hardness. Shrinkage begins at this temperature and the castings, consequently, will crack if the cores do not permit shrinkage. Therefore, where very thin wall castings are made of brittle alloys, the correct choice of core binder is important if shrinkage cracks are to be avoided. In some instances, particularly aluminum-base alloys, the core binder is a thin solution of glue. This softens readily and burns quickly, and thus relieves casting strains. This is an example of the extreme that may be necessary to provide for shrinkage in extraordinary cases.

## 4. PERMANENT MOLD CASTINGS

The production of non-ferrous castings in metal or permanent molds is a specialty that has become most highly skilled and very carefully organized. The use of permanent molds for certain types of castings, particularly, aluminum-base castings, requires a high degree of experience and skill. Consequently, they should be used only by those who are competent to handle them. Similarly, die-castings are non-ferrous castings produced in dies, made by those who understand their production, and cast under pressures controlled by special machines designed to meet special conditions.

At the present state of development (1937), the die-casting process and the metal mold process are differentiated by the fact that in die-casting, the metal is forced into the mold under pressure, whereas it enters the mold by gravity in the metal mold process. The molds are similar, but obviously must be different in design, since in the die-casting process, the pressure on the metal compensates to some extent for the shrinkage of the alloy during cooling. In the metal mold process, the shrinkage must be compensated by risers, shrink balls, or similar means.

The production of centrifugal castings likewise is a specialty which is highly developed. It calls for knowledge and skill that includes a knowledge of the kind of alloys cast and of the proper design of molds, machines and the forces connected with the centrifugal casting of metals.

## 5. CLEANING CASTINGS

Cleaning or snagging in the non-ferrous foundry is much the same process as in iron and steel foundries, except that sand blasting or shot blasting is done at lower pressures to avoid injury to the surface of the casting. Risers usually are removed by band saws or hack saws. Sprue cutters also are used to advantage to remove gates or risers from castings with thin gates. Water tumbling may be used to clean some brass or bronze alloy castings, where the finish thereby derived short-cuts the final finishing process. If polishing is desirable, leather chips, or in some cases wooden chips, are placed in the tumbling barrel with the castings. Cleaning and polishing then is accomplished in a single operation.

In foundries producing non-ferrous castings exclusively, grindings from the grinding wheels contain considerable brass and bronze. These grindings should be saved and sold for their metal content. If non-ferrous castings are a very small proportion of the total production of an iron foundry, and the non-ferrous grindings are mixed with iron grindings, no credit will be given for the non-ferrous grindings. It, therefore, is desirable to so organize the foundry that non-ferrous grindings can be separated, and the value of the copper content realized.

## 6. DESIGN OF NON-FERROUS CASTINGS

In the design of non-ferrous castings engineers will do well to bear in mind that many alloys shrink rather more than gray iron castings. All of them shrink somewhat during solidification. Therefore, where thick sections join thin walls, proper fillets of ample size should be provided. In other words, transition from a thin wall to a thick wall should be as gradual as possible. Heavy bosses or lumps on castings should be avoided if soundness is to be maintained, particularly if such heavy sections are at points where proper feeding to take care of shrinkage is difficult. If possible, engineers should submit preliminary designs to competent non-ferrous foundrymen for criticism and for suggestions from the foundry standpoint before a final design is adopted. This almost invariably will avoid many pitfalls that otherwise would cause much trouble and probably large expense. It also is desirable to discuss pattern design with the foundry. Plans then may be formulated to produce the most desirable patterns from the standpoint of foundry cost and insurance against defective castings.

# Section 21

# THE MACHINE SHOP

## LATHE WORK

## SINGLE-POINT METAL-CUTTING TOOLS

By Robert C. Deale

## PLANER WORK

By Forrest E. Cardullo

## MILLING MACHINE PRACTICE

By Hanst Ernst and Mario Martelloti

## DRILLS AND DRILLING

## PRESSURE WORKING OF METALS

By H. B. Morse

## ABRASIVE PROCESSES

## BROACHING

## METAL SAWING

## MACHINE TOOL MOTOR DRIVES

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# THE MACHINE SHOP

Progress in the development of machine tools is so rapid that it is impossible to give details concerning the construction and operation of the many standard and special forms of lathes, milling machines, planers, shapers, boring mills, drilling machines, and other tools. The reader is referred to the catalogs of manufacturers for the latest information on this subject. The following pages cover mainly the fundamental practices underlying the operation of the various standard machine tools.

# LATHE WORK

## 1. GENERAL

**SPEED OF CUTTING TOOLS.**—The following formulas give the relations of the diameter of a rotating tool, or piece of work, number of revolutions, and cutting speeds. Let $d$ = diameter of rotating piece, in.; $n$ = revolutions per minute; $S$ = speed of circumference, ft. per min. Then

$$S = (\pi d n/12) = 0.2618\, d n \quad [1]$$

$$n = (S/0.2618\, d) = 3.82\, S/d \quad [2]$$

$$d = 3.82\, S/n \quad [3]$$

Table 1 gives the revolutions per minute for various diameters of work or tool over the range of cutting speeds commonly used in the machine shop.

**LATHE CHANGE GEARS** for screw cutting may be selected as follows:

**Simple-geared Lathes.**—Let $G_s$ = number of teeth on stud gear; $G_d$ = number of teeth on lead-screw gear; $T_s$ = number of threads per inch on lead-screw; $T_c$ = number of threads per inch to be cut. Select any gear for $G_s$. Then

$$G_d = (G_s \times T_s)/T_c \quad [4]$$

If the result is a fractional number, select another gear for $G_s$ and repeat the calculation. This equation assumes that the stud rotates at the same speed as the spindle. If these speeds are different, divide by the ratio of stud speed to spindle speed.

**Compound-geared Lathes.**—Let $G_{1r}, G_{2r}, \ldots G_{nr}$ = number of teeth on driving gears; $G_{1d}, G_{2d} \ldots G_{nd}$ = number of teeth on driven gear; $T_s$ = number of threads per inch on lead-screw; $T_c$ = number of threads per inch to be cut. Select at random all of the driving gears and also all of the driven gears except the last one. Then

$$G_n = (T_c \times G_{1r} \times G_{2r} \times \ldots G_{nr})/(T_s \times G_{1d} \times G_{2d} \times \ldots G_{(n-1)d}) \quad [5]$$

**Metric Screw Threads** may be cut on lathes with inch-divided lead-screws, by using change gears with 50 and 127 teeth, since 127 cm. = 50 in., and 127 × 0.3937 = 49.9999 in.

**THE SPEED OF COUNTERSHAFT** of a lathe is determined by assuming a slow speed with the back gear, say 6 ft. per min., on the largest diameter that the lathe will swing.

Example.—A 30-in. lathe will swing 30 in. = say, 90 in. circumference = 7 ft. 6 in.; the lowest triple gear should give a speed of 6 ft. per minute = 7.5/6 = 1.25 r.p.m.

**TURNING TAPERS IN A LATHE.**—Tapers can be turned in a lathe either by means of a taper attachment, which is supplied by many lathe manufacturers, or by setting the tailstock center a certain distance out of line with the headstock center. No rule for setting over the center can be given that will be exact, because the centers enter the work an indefinite distance. An approximate rule is: Divide the difference in diameters of the large and small ends of the taper by 2. Multiply the quotient by the ratio of the total length of the shaft to the length of the taper portion.

**SHAPE OF LATHE TOOLS.**—See Single-point Cutting Tools, pp. 21–07 to 21–27. For shapes of tools for aluminum and other non-ferrous metals see discussion of the various metals in other pages of this book.

**LUBRICANTS FOR LATHE CENTERS.**—*Machinery* recommends as lubricants for lathe centers to prevent cutting or abrasion: 1. Dry or powdered red lead mixed with a good mineral oil to the consistency of cream. 2. White lead mixed with sperm oil, together with enough graphite to give the mixture a dark red color. 3. One part graphite, four parts tallow, thoroughly mixed.

**Table 1.—Cutting Speeds of Turned Work, Milling Cutters, and Drills**

| Diam., in. | Feet per minute | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 20 | 30 | 40 | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 225 | 250 | 300 |
| | Revolutions per minute | | | | | | | | | | | | |
| 1/4 | 305.6 | 458.4 | 611.2 | 764.0 | 1145.9 | 1527.9 | 1910.0 | 2291.8 | 2674.0 | 3055.8 | 3438.0 | 3819.7 | 4583.7 |
| 3/8 | 203.7 | 305.6 | 407.4 | 509.3 | 763.7 | 1018.6 | 1273.7 | 1527.5 | 1783.3 | 2036.7 | 2292.8 | 2545.8 | 3055.0 |
| 1/2 | 152.8 | 229.2 | 305.6 | 382.0 | 572.9 | 763.9 | 955.0 | 1145.9 | 1337.0 | 1527.9 | 1719.0 | 1909.9 | 2291.8 |
| 5/8 | 122.2 | 183.4 | 244.5 | 305.6 | 458.4 | 611.2 | 763.8 | 916.7 | 1069.3 | 1222.3 | 1374.8 | 1527.9 | 1833.5 |
| 3/4 | 101.8 | 152.8 | 203.7 | 254.6 | 382.0 | 509.3 | 636.3 | 763.9 | 890.8 | 1018.6 | 1145.3 | 1273.2 | 1527.9 |
| 7/8 | 87.3 | 130.9 | 174.6 | 218.3 | 327.4 | 436.6 | 545.6 | 654.9 | 763.9 | 873.3 | 982.1 | 1091.5 | 1309.8 |
| 1 | 76.4 | 114.6 | 152.8 | 191.0 | 286.5 | 382.0 | 477.5 | 573.0 | 668.5 | 763.9 | 859.5 | 954.9 | 1145.9 |
| 1 1/8 | 67.9 | 101.8 | 135.8 | 169.7 | 254.4 | 339.5 | 424.5 | 508.8 | 594.3 | 678.4 | 764.1 | 848.0 | 1017.6 |
| 1 1/4 | 61.1 | 91.7 | 122.2 | 152.8 | 229.2 | 305.6 | 382.0 | 458.4 | 534.8 | 611.2 | 687.6 | 763.9 | 916.7 |
| 1 3/8 | 55.6 | 83.3 | 111.1 | 138.9 | 208.3 | 277.7 | 347.3 | 416.5 | 486.2 | 555.4 | 625.1 | 694.2 | 833.1 |
| 1 1/2 | 50.9 | 76.4 | 101.8 | 127.2 | 190.8 | 254.4 | 318.4 | 381.6 | 445.7 | 508.8 | 573.1 | 636.0 | 763.2 |
| 1 3/4 | 43.7 | 65.5 | 87.3 | 109.2 | 163.6 | 218.1 | 272.9 | 327.2 | 382.0 | 436.2 | 491.2 | 545.3 | 654.3 |
| 2 | 38.2 | 57.3 | 76.4 | 95.5 | 143.2 | 191.0 | 238.8 | 286.5 | 334.3 | 382.0 | 429.8 | 477.5 | 573.0 |
| 2 1/4 | 34.0 | 50.9 | 67.9 | 84.9 | 127.2 | 169.6 | 212.3 | 254.4 | 297.2 | 339.2 | 382.1 | 424.0 | 508.8 |
| 2 1/2 | 30.6 | 45.8 | 61.1 | 76.4 | 114.6 | 152.8 | 191.0 | 229.2 | 267.4 | 305.6 | 343.8 | 382.0 | 458.4 |
| 2 3/4 | 27.8 | 41.7 | 55.6 | 69.5 | 104.0 | 138.7 | 173.6 | 208.3 | 243.1 | 277.3 | 312.5 | 346.6 | 416.0 |
| 3 | 25.5 | 38.2 | 50.9 | 63.7 | 95.4 | 127.2 | 159.1 | 190.8 | 222.8 | 254.4 | 286.4 | 318.0 | 381.6 |
| 3 1/2 | 21.8 | 32.7 | 43.7 | 54.6 | 81.6 | 108.9 | 136.4 | 163.3 | 191.0 | 217.7 | 245.5 | 272.2 | 326.6 |
| 4 | 19.1 | 28.7 | 38.2 | 47.8 | 71.6 | 95.5 | 119.4 | 143.2 | 167.1 | 191.0 | 214.9 | 238.7 | 286.5 |
| 4 1/2 | 17.0 | 25.5 | 34.0 | 42.5 | 63.6 | 84.8 | 106.1 | 127.2 | 148.6 | 169.6 | 191.0 | 212.0 | 254.4 |
| 5 | 15.3 | 22.9 | 30.6 | 38.1 | 57.3 | 76.4 | 95.5 | 114.6 | 133.7 | 152.8 | 171.9 | 191.0 | 229.2 |
| 5 1/2 | 13.9 | 20.8 | 27.8 | 34.7 | 52.1 | 69.4 | 86.9 | 104.2 | 121.6 | 138.9 | 156.4 | 173.6 | 208.3 |
| 6 | 12.7 | 19.1 | 25.5 | 31.8 | 47.6 | 63.4 | 79.6 | 95.1 | 111.5 | 126.8 | 143.3 | 158.5 | 190.2 |
| 7 | 10.9 | 16.4 | 21.8 | 27.3 | 41.0 | 54.6 | 68.2 | 81.9 | 95.6 | 109.2 | 122.9 | 136.6 | 163.9 |
| 8 | 9.6 | 14.3 | 19.1 | 23.9 | 35.8 | 47.7 | 59.8 | 71.6 | 83.7 | 95.5 | 107.6 | 119.4 | 143.2 |
| 9 | 8.5 | 12.7 | 17.0 | 21.2 | 31.8 | 42.4 | 53.0 | 63.6 | 74.2 | 84.8 | 95.4 | 106.0 | 127.2 |
| 10 | 7.6 | 11.5 | 15.3 | 19.1 | 28.6 | 38.2 | 47.8 | 57.3 | 66.9 | 76.4 | 86.0 | 95.5 | 114.6 |
| 11 | 6.9 | 10.4 | 13.9 | 17.4 | 26.0 | 34.7 | 43.4 | 52.1 | 60.7 | 69.4 | 78.1 | 86.8 | 104.2 |
| 12 | 6.4 | 9.5 | 12.7 | 15.9 | 23.8 | 31.7 | 39.8 | 47.6 | 55.7 | 63.4 | 71.6 | 79.3 | 95.1 |
| 13 | 5.9 | 8.8 | 11.8 | 14.7 | 22.1 | 29.4 | 36.8 | 44.1 | 51.5 | 58.8 | 66.2 | 73.5 | 88.2 |
| 14 | 5.5 | 8.2 | 10.9 | 13.6 | 20.5 | 27.3 | 34.1 | 40.9 | 47.8 | 54.6 | 61.4 | 68.3 | 81.9 |
| 15 | 5.1 | 7.6 | 10.2 | 12.7 | 19.1 | 25.4 | 31.8 | 38.2 | 44.6 | 50.9 | 57.3 | 63.6 | 76.3 |
| 16 | 4.8 | 7.2 | 9.5 | 11.9 | 17.9 | 23.9 | 29.8 | 35.8 | 41.8 | 47.8 | 53.7 | 59.7 | 71.6 |
| 18 | 4.2 | 6.4 | 8.5 | 10.6 | 15.9 | 21.2 | 26.5 | 31.8 | 37.1 | 42.4 | 47.7 | 53.0 | 63.6 |
| 20 | 3.8 | 5.7 | 7.6 | 9.6 | 14.3 | 19.1 | 23.9 | 28.6 | 33.4 | 38.2 | 43.0 | 47.8 | 57.3 |
| 22 | 3.5 | 5.2 | 6.9 | 8.7 | 12.9 | 17.2 | 21.8 | 25.8 | 30.5 | 34.4 | 39.2 | 43.0 | 51.6 |
| 24 | 3.2 | 4.8 | 6.4 | 8.0 | 11.9 | 15.9 | 19.9 | 23.8 | 27.8 | 31.7 | 35.8 | 40.1 | 47.6 |
| 26 | 2.9 | 4.4 | 5.9 | 7.3 | 10.9 | 14.5 | 18.4 | 21.8 | 25.7 | 29.0 | 33.1 | 36.3 | 43.5 |
| 28 | 2.7 | 4.1 | 5.5 | 6.8 | 10.3 | 13.7 | 17.0 | 20.5 | 23.8 | 27.3 | 30.6 | 34.2 | 41.0 |
| 30 | 2.5 | 3.8 | 5.1 | 6.4 | 9.5 | 12.7 | 15.9 | 19.1 | 22.3 | 25.4 | 28.6 | 31.8 | 38.2 |
| 32 | 2.4 | 3.6 | 4.8 | 6.0 | 9.0 | 11.9 | 14.9 | 17.9 | 20.9 | 23.9 | 26.8 | 29.9 | 35.8 |
| 34 | 2.2 | 3.4 | 4.5 | 5.6 | 8.4 | 11.2 | 14.0 | 16.9 | 19.6 | 22.4 | 25.2 | 28.1 | 33.7 |
| 36 | 2.1 | 3.2 | 4.2 | 5.3 | 7.9 | 10.6 | 13.3 | 15.9 | 18.6 | 21.2 | 23.9 | 26.5 | 31.8 |
| 38 | 2.0 | 3.0 | 4.0 | 5.1 | 7.6 | 10.1 | 12.6 | 15.2 | 17.7 | 20.0 | 22.7 | 25.3 | 30.0 |
| 40 | 1.9 | 2.9 | 3.8 | 4.8 | 7.2 | 9.6 | 11.9 | 14.3 | 16.7 | 19.1 | 21.5 | 23.9 | 28.7 |
| 42 | 1.8 | 2.7 | 3.6 | 4.5 | 6.8 | 9.1 | 11.4 | 13.6 | 15.9 | 18.2 | 20.5 | 22.7 | 27.3 |
| 44 | 1.7 | 2.6 | 3.5 | 4.3 | 6.5 | 8.7 | 10.9 | 13.0 | 15.2 | 17.4 | 19.5 | 21.7 | 26.0 |
| 46 | 1.7 | 2.5 | 3.3 | 4.2 | 6.2 | 8.3 | 10.4 | 12.5 | 14.5 | 16.6 | 18.7 | 20.8 | 24.9 |
| 48 | 1.6 | 2.4 | 3.2 | 4.0 | 6.0 | 8.0 | 10.0 | 11.9 | 13.9 | 15.9 | 17.9 | 19.9 | 23.9 |
| 54 | 1.4 | 2.1 | 2.8 | 3.5 | 5.3 | 7.1 | 8.8 | 10.6 | 12.4 | 14.1 | 15.9 | 17.7 | 21.2 |
| 60 | 1.3 | 1.9 | 2.5 | 3.2 | 4.8 | 6.4 | 8.0 | 9.5 | 11.1 | 12.7 | 14.3 | 15.9 | 19.1 |

## 2. DRIVES, FEEDS AND SPEEDS

**DRIVES.**—The methods in common use for driving lathes and other machine tools are: Cone pulleys, usually with one or more sets of back gears; geared heads, driven by a single-speed pulley or electric motor, giving from 12 to 16 spindle speeds; individual electric motor, with a variable speed drive interposed between it and the machine. On belt-driven machines, the number of speeds available can be increased by the use of a two-speed countershaft. Speeds usually are in geometric progression. For the details of construction of the drives of a wide variety of machine tools, see O. W. Boston, Engineering Shop Practice (John Wiley & Sons).

**PROPORTIONING MACHINE TOOL GEARS IN GEOMETRIC PROGRESSION.** (C. C. Stutz, *Am. Mach.*, Jan. 15, 1920.)—**Notation.**—$a$ = slowest speed; $b$ = highest speed; $n$ = total number of speeds; $n - 1$ = total steps or intervals between total number of speeds; $f$ = ratio between successive speeds, or factor for obtaining next higher speed; $d$ = number of steps in cone pulley.

The speeds form the algebraic series, $a$, $af$, $af^2$, $af^3$ . . . $af^{n-2}$, $af^{n-1}$.

Since $af^{n-1} = b$, $f^{n-1} = b/a$, and $f = \sqrt[n-1]{b/a}$ . . . . . . . . [6]

If the desired number of speeds cannot be obtained by a cone pulley alone, back gearing, triple gearing or quadruple gearing must be used. For example, a triple-geared lathe, with a four-step cone ($d = 4$), would have the following series of speeds.

| Cone | Back Gear | Triple Gear |
|---|---|---|
| $af^{11}$, $af^{10}$, $af^9$, $af^8$ | $af^7$, $af^6$, $af^5$, $af^4$ | $af^3$, $af^2$, $af$, $a$. |

These three series bear the following relation to each other:

1. The second series can be obtained by dividing each member of the first series by $f^4$, and $f^4$ is the ratio of the back gears.
2. The third series can be obtained by dividing each member of the first series by $f^4f^4$, and $f^8$ = ratio of the triple gears.

These relations may be summarized by:

log of ratio of back gears $= d \log f$. . . . . . . . . [7]
log of ratio of triple gears $= 2d \log f$. . . . . . . . [8]
log of ratio of quadruple gears $= 3d \log f$, etc. . . . . . . [9]

When two countershaft speeds are available, the number of speeds obtainable at the spindle $= 2n$, and in this case $f = \sqrt[2n-1]{b/a}$.

We may consider that one-half of the obtainable speeds are due to the first countershaft speed, and one-half to the second countershaft speed, giving two series as follows:

$$a,\ af^2,\ af^4\ \ldots\ af^{2n-4},\ af^{2n-2}.$$
$$af,\ af^3,\ af^5\ \ldots\ af^{2n-3},\ af^{2n-1}.$$

Both series are geometrical progressions, with the same factor $f^2$. The second series can be obtained by multiplying the first series by factor $f$, which indicates that the ratio of countershaft speeds also must be $f$. The gear ratios of the machine are found by considering either one of the two series, and

log of ratio of back gears $= df^2$. . . . . . . . . [10]
log of ratio of triple gears $= 2df^2$. . . . . . . . . [11]
log of ratio of quadruple gears $= 3df^2$. . . . . . . . [12]

EXAMPLE 1.—Required the speeds and gear ratios of a 6-step cone, triple-geared; slowest speed 0.75 r.p.m.; highest speed, 117 r.p.m.

*Solution.* $a = 18$; $b = 117$; $n = 18$; $n - 1 = 17$; $d = 6$.

From [6]. $f = \sqrt[17]{117/0.75} = \sqrt[17]{156}$; $\log 156 = 2.1931246$.

$\log f = 1/17 \times \log 156 = 0.129007329$, and $f = 1.3456$.

The speeds are most conveniently found by the continuous addition of log $f$ as follows:

| TRIPLE-GEAR SPEEDS. | | BACK-GEAR SPEEDS. | | CONE SPEEDS. | |
|---|---|---|---|---|---|
| log 0.75 = 0.875061300 − 1 | = log 0.75 | 0.520097945 | | 1.294141919 | |
| log $f$ = 0.129007329 | | 0.129007329 | | 0.129007329 | |
| 0.004068629 | = log 1.009 | 0.649105274 | = log 4.457 | 1.423149248 | = log 26.294 |
| 0.129007329 | | 0.129007329 | | 0.129007329 | |
| 0.133075958 | = log 1.358 | 0.778112603 | = log 5.999 | 1.552156577 | = log 35.658 |
| 0.129007329 | | 0.129007329 | | 0.129007329 | |
| 0.262083287 | = log 1.828 | 0.907119932 | = log 8.074 | 1.681163906 | = log 47.991 |
| 0.129007329 | | 0.129007329 | | 0.129007329 | |
| 0.391090616 | = log 2.461 | 1.036127261 | = log 10.867 | 1.810171235 | = log 64.591 |
| 0.129007329 | | 0.129007329 | | 0.129007329 | |
| 0.520097945 | = log 3.312 | 1.165134590 | = log 14.626 | 1.939178564 | = log 86.931 |
| | | 0.129007329 | | 0.129007329 | |
| | | 1.294141919 | = log 19.685 | 2.068185893 | = log 117 |

The gear ratios are as follows:
Since $d = 6$, we have from [7],

log (back gear ratio) $= 6 \log f = 6 \times 0.129007329 = 0.774043974$,
whence back gear ratio $= 5.9435$,
From [8], log (triple gear ratio) $= 12 \log f = 12 \times 0.129007329 = 1.548087948$,
whence triple gear ratio $= 35.325$.

Example 2.—Required the speeds of a 4-step, back-geared cone, with two countershaft speeds; slowest speed, 25 r.p.m.; highest speed, 500 r.p.m.

$$a = 25; b = 500; 2n = 2(4 \times 2) = 16; 2n - 1 = 15; d = 4.$$

$$f = \sqrt[15]{500/25} = \sqrt[15]{20};\ \log 20 = 1.301030.$$

$$\log f = 1/_{15} \times \log 20 = 0.08673533, \text{ and } f = 1.2210.$$

Then by continuous addition of log $f$ to log $a$, as in Example 1, we obtain the series of 16 speeds as follows:

| Speed | r.p.m. | Speed | r.p.m. | Speed | r.p.m. | Speed | r.p.m. |
|---|---|---|---|---|---|---|---|
| 1 | 25 | 5 | 55.58 | 9 | 123.54 | 13 | 274.64 |
| 2 | 30.53 | 6 | 67.86 | 10 | 150.85 | 14 | 335.35 |
| 3 | 37.28 | 7 | 82.86 | 11 | 184.20 | 15 | 409.48 |
| 4 | 45.51 | 8 | 101.18 | 12 | 224.92 | 16 | 500. |

These speeds are divided into two series, each one countershaft speed.

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| 1st Countershaft Speed | 1 | 3 | 5 | 7 | 9 | 11 | 13 | 15 |
| r.p.m. ................ | 25 | 37.28 | 55.58 | 82.86 | 123.54 | 184.20 | 274.64 | 409.48 |
| 2d Countershaft Speed | 2 | 4 | 6 | 8 | 10 | 12 | 14 | 16 |
| r.p.m. ................ | 30.53 | 45.51 | 67.86 | 101.18 | 150.85 | 224.92 | 335.35 | 500 |

Gear ratios are: from [10], back gear ratio = $d f^2 = \log^{-1} 4 \times (2 \log f) = \log^{-1} 8 \times 0.08673533 =$ 4.9418

Countershaft ratio = $f$ = 1.2210.

Carl G. Barth (*Am. Mach.*, Jan. 11, 1912) suggests a ratio of $f = \sqrt[4]{2} = 1.189$. With this ratio the revolutions per minute of the spindle are doubled every fourth speed. An editorial (*Am. Mach.*, Dec. 3, 1914) discussing the advantage of adopting this ratio for a speed series shows that it will fulfill all the ordinary requirements of machine tool work, and that practically any desired speed in either lathe or drill press can be obtained when the machine is speeded according to a geometric progression based on the ratio 1.189.

A geometric progression of the feeds available on machine tools also is desirable, and Mr. Barth has recommended the same ratio for the feed series as for the speed series, that is, $\sqrt[4]{2} = 1.189$.

**HYDRAULIC FEEDS.**—Hydraulic feeding mechanism for machine tools has had wide application since about 1925, particularly to grinding, milling and drilling machines. The inherent advantages of hydraulic feed as compared to geared feed are stated to be as follows: The feed automatically adjusts itself to variations in depth of cut, decreasing as the cut becomes heavier, and returning to normal when the depth of cut becomes normal. This prolongs tool life and eliminates danger of tool breakage. The feed can be set for the maximum rate that the tool and machine can withstand, and can be varied by the operator from maximum down to zero without stopping the machine. Feed pressure is under control of the operator at all times. Increase of feed pressure indicates dulling of tools, and this is shown by the pressure reading and control mechanisms.

Two systems of hydraulic feed are in general use: 1. The closed circuit or variable volume system; 2. The open circuit or constant volume system. In both systems, the operating fluid either acts against a piston in a cylinder to give straight line motion to the feed mechanism, or operates some form of hydraulic motor to give rotary motion.

In the *closed circuit system*, a variable-delivery pump draws oil from a reservoir and delivers it to a cylinder to drive a piston connected to the machine table or other mechanism that is to be moved. Suitable valves, actuated by the table, control the admission and release of oil in the cylinder. Oil flows from the cylinder back to the reservoir. For rotary motion, the oil is delivered to a hydraulic motor instead of a cylinder. The pump is set to deliver only the amount of oil necessary to maintain the given rate of feed, plus unavoidable leakage. The rate of feed is proportional to the discharge of the pump, less the leakage past the pump pistons and the feed piston.

In the *open circuit system* a gear pump delivers a constant volume of oil. Such portion as is needed to maintain the desired constant pressure in the feed cylinder or motor, is delivered thereto. The balance is automatically by-passed to the reservoir. This system is particularly applicable to grinding machine tables, where the load is constant and is principally an inertia load. Applied to a drilling machine, it has the disadvantage of feeding a large drill more slowly than a small one, and dull drills more slowly than sharp ones. Further, the rate of feed can be determined only by trial, and drills jump ahead when breaking through.

For many data on hydraulic feeds, see Symposium on Hydraulic Feeds for Machine Tools, *Trans.* A.S.M.E., MSP–50–4, 1928. For data on the construction of hydraulic feed equipment, see literature issued by the Oilgear Corp., Milwaukee, Wis., and by the Cincinnati Milling Machine Co., Cincinnati.

# SINGLE-POINT METAL-CUTTING TOOLS

By Robert C. Deale

## 1. DEFINITIONS

The following definitions have been proposed (1937) to the American Standards Association. They relate to the terminology to be used in describing the construction and use of single-point metal-cutting tools for lathes, planers, shapers, boring mills, etc. For other forms of single-point tools, used largely in planers, see p. 21–30.

COMPONENTS.—**The Shank** is that part of the tool on one end of which the point is formed or the tip or bit supported.

**The Size of a Tool** of square or rectangular section is expressed by giving in the order named the width of shank, the height of shank and the total tool length in inches, *i.e.*, $3/4 \times 1\ 1/2 \times 12$ in.

**The Point or Nose** is all that part of a tool which is shaped to produce the cutting edges and face. See Figs. 1 and 2.

**The Neck** is an extension of the shank but of reduced sectional area. See Fig. 3.

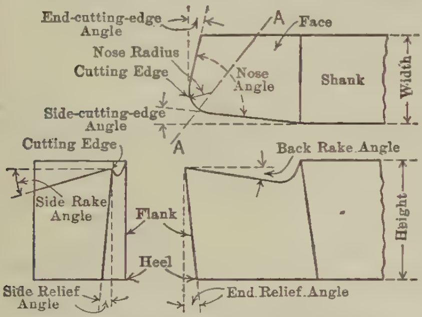


FIG. 1. Right-cut, Straight-shank Single-point Ground Tool

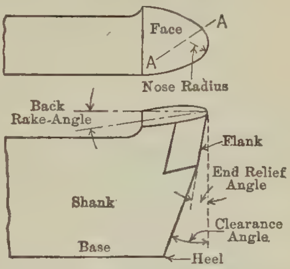


FIG. 2. Right-cut, Straight-shank, Single-point Raised Face Forged Tool

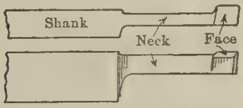
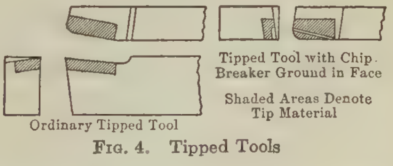


FIG. 3. Necked Tool, Forged



Tipped Tool with Chip Breaker Ground in Face
Shaded Areas Denote Tip Material
Ordinary Tipped Tool

FIG. 4. Tipped Tools

**The Base** is that side of the shank which bears against the support taking the tangential pressure of the cut.

**The Face** is that surface on which the chip impinges as it is cut from the work.

**The Heel** is the end of the base immediately below and supporting the face.

**The Cutting Edge,** or lip, is that portion of the face edge along which the chip is separated from the work. It consists usually of the side-cutting-edge, the nose radius and the end-cutting-edge. See Fig. 1.

**The Nose** is the curve formed by joining the side- and end-cutting edges.

**The Profile** is a plan view when looking at the face from a point at right angles to the base. It is formed by joining the side- and end-cutting-edges with the nose.

**The Flat** is a straight portion of the cutting-edge intended to produce a smooth machined surface.

**The Flank** of the tool is the surface below the cutting-edge.

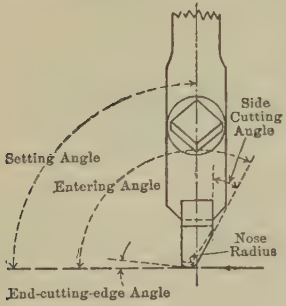
TYPES OF TOOLS.—Single-point tools may be made up as follows:

**A Ground Tool** is one in which a point is formed on the end of a bar (shank) of tool steel by grinding. See Fig. 1.

**A Forged Tool** is one in which the point is forged roughly to shape on the end of a bar (shank) of tool steel and subsequently hardened and ground. See Fig. 2.

**A Tipped Tool** is one with a tip of cutting material permanently attached to a shank of non-cutting metal. See Fig. 4.

**A Bit Tool** is one in which a small part of tool material of square, rectangular, or other section, or forged to a special shape, is held in the end of a holder or shank. See Fig. 5.



**A Tool Bit** is a relatively small piece of tool material clamped in a tool holder in such a way that it can be readily removed and replaced.

**A Right-cut Single-point Tool** is one in which the cutting-edge is on the right side when looking at the point end of the tool with the face upward and the shank pointing away.

**A Left-cut Single-point Tool** is one in which the cutting-edge is on the left side when looking at the point end with the face upward and the shank pointing away.

**An End Cut Tool** is one having its principal cutting-edge on the end.

**A Straight Tool** is one in which the point is on the forward edge of a straight shank. See Fig. 1.

**A Bent Tool** is one in which the point is bent to the right or the left. A right bent tool has the point bent to the right when looking at the tool from the point end with the face upward and the shank pointing away.

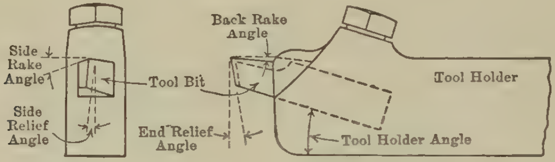


FIG. 5. Tool Bit and Holder

**The Shank Angle** is the angle by which the point of a bent tool deviates from the straight portion of the shank. See Fig. 6.

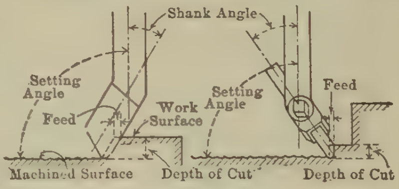


FIG. 6. Bent-shank, Single-point Tool

**An Offset Tool** is one in which the point is at either side of, but parallel to, the shank. In a right offset tool, the point is offset to the right of the shank when looking at the tool from the point end with the face upward and with the shank pointing away.

**TOOL ANGLES.**—The angles defined below are angles formed between various portions or surfaces of the tool. They are angles formed in forging or grinding the tool, and should not be confused with the working angles which mainly are angles formed between some surface of the tool and of the work.

**Back-rake Angle** is the angle between the face of a tool and the shank base, measured in a plane at right angles to the base and parallel to the center line of the nose. The angle is positive if the face slopes downward from the nose toward the shank, and negative if the face slopes upward toward the shank.

**Side-rake Angle** is the angle between the face of the tool and the shank base, measured in a plane at right angles to the base and at right angles to the center line of the nose.

**Relief Angle** is the angle between a plane perpendicular to the base of a tool or tool holder and that portion of the flank immediately adjacent to the cutting edge.

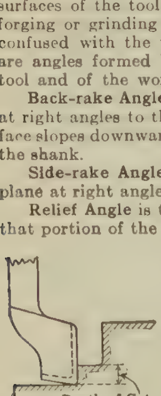


FIG. 7. Right-cut, Right-offset, Single-point Tool

**Side Relief Angle** is measured in the plane of the side-rake angle to the center line of the tool nose.

**The End Relief Angle** is measured in the plane of the back-rake angle.

**The Clearance Angle** is an angle between a perpendicular to the base of a tool and that portion of the flank immediately adjacent to the base. The side clearance is measured in the plane of the side-rake angle. The end clearance is measured in the plane of the back-rake angle.

**The Side-cutting-edge Angle** is the angle between the straight cutting edge and the side of the tool shank.

**The End-cutting-edge Angle** is the angle between the cutting edge on the end of the tool and a line at right angles to the side edge of the tool shank.

**Nose Angle** is the angle included between the side-cutting-edge and the end-cutting-edge. See Fig. 1.

**WORKING ANGLES** are those angles between the tool and work which depend not only on the shape of the tool but also on its position with respect to the work.

**Setting Angle** is the angle made by the shank of a tool with a line at right angles to the finished surface of the work. See Figs. 5 and 6.

**The Entering Angle** is the angle which the side-cutting-edge of the tool makes with the finished surface of the work. See Fig. 5.

**True Rake Angle** (or top rake) is the actual slope of the tool face from the active cutting edge in the direction of chip flow. It is a combination of front and side rake angles and varies with the setting of the tool and with the feed and depth of cut.

**Lip Angle** is the included angle of the tool material between the face and the ground flank, measured in a plane at right angles to the cutting edge. When measured in a plane perpendicular to the cutting edge at the end of the tool it is called the **end lip angle.** When measured in the direction of chip flow, it is called **the true lip angle.** See Fig. 8.

**The Working Relief Angle** is the angle between the ground flank of the tool and a line, tangent to the cut surface, passing through the active cutting edge.

**Cutting Angle** is equal to the lip angle plus the working relief angle at the point of action.

**The Working End-cutting-edge Angle** is the angle between the straight end-cutting-edge and a plane tangent to the machined surface at the point of cutting.

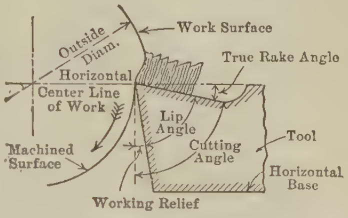


FIG. 8. Section through Plane of Chip Flow, Indicated as *AA* in Figs. 1 and 2

**CUTTING SPEED** is the peripheral or surface speed of the work with respect to the tool. In turning, it usually is measured on the uncut or work surface of the work ahead of the tool.

**DEPTH OF CUT** is the distance between the bottom of the cut and the uncut surface, measured in a direction at right angles to the finished surface of the work.

**FEED** is the relative amount of motion of the tool into the work for each revolution, stroke or unit of time.

**MACHINED SURFACE** is the surface left by the cutting tool. See Figs. 6 and 8.

**WORK SURFACE** is the surface that is to be machined. See Figs. 6 and 8.

## 2. TOOL CONTOUR AND SIZE *

**CHIP PROPORTIONS.**—The proportions of a chip, as affected by the nose contour of a tool or by the relation between depth of cut and feed, have great effect on the cutting speed which may be used. A change in chip proportions caused by a change in nose contour of the tool has the same effect on cutting speed as a similar change caused by a change in depth of cut and feed. When removing a chip of given cross-sectional area, changes in the tool contour may cause variations in the cutting speed of as much as 60%, while changes of as much as 400% may be caused by changes in the relation between depth of cut and feed.

**TOOL SIZES.**—The number of sizes of tools used in a given shop should be as small as possible. The following sizes should meet all ordinary shop requirements: *Rectangular Tools.*—$^3/_8 \times {}^3/_4$ in.; $^1/_2 \times 1$ in.; $^5/_8 \times 1\,^1/_4$ in.; $^3/_4 \times 1\,^1/_2$ in.; $1 \times 2$ in.; $1\,^1/_2 \times 3$ in. *Square Tool Bits.*—$^3/_{16}$, $^1/_4$, $^3/_8$, $^1/_2$, $^5/_8$, $^3/_4$, 1, $1\,^1/_4$, and $1\,^1/_2$ in. *Rectangular Tool Bits.*—$^3/_{32} \times {}^1/_2$ in.; $^1/_8 \times {}^3/_4$ in.; $^3/_{16} \times 1\,^1/_8$ in.; $^1/_4 \times 1\,^3/_8$ in. A relatively small number of the foregoing sizes should be ample for all except the largest machine shops.

Tool bits are available which are forged to the desired form. Such bits must be used in special holders, usually made by the company furnishing the bits. Such bits are at times very convenient for shop use, but are much more expensive per pound than bar stock. When adopting such tools and holders, both the stiffness of support to the cutting edge, and the percentage of the new tool which may be used before it is discarded should be considered.

**Effect of Tool Size.**—The size of a tool has no effect other than to give strength to the tool, to reduce deflection, or to increase the tool width available for a forming cut. A properly supported small tool may be run at the same cutting speed as a large one having the same shape, for the same depth of cut and feed.

**TOOL FORM.**—When heavy roughing cuts are to be taken, and it is desired to use heavy forged tools, there are no better shapes of tool than those developed by Carl Barth. These are shown in Fig. 9. Dimensions are given in Table 1.

Tools of the form shown in Fig. 10 are used for parting cuts. They usually are from $^1/_2$ to $^3/_8$ in. wide at the nose, except when the diameter of work to be parted is large, when they may be wider. To obtain the necessary stiffness, the cutting section of such a tool usually is forged down from a shank of comparatively large section. Note that the sides of the neck of this parting tool converge at a slight angle, usually from $^1/_2$ deg. to 2 deg., to avoid interference with the side of the cut. The side relief angle is small, usually not over 2 deg. At times a blade having a height of from 5 to 8 times its width, held in a special holder, is used as a parting tool.

* For complete data on the use of cutting tools, including feed and speed data for many varieties of alloy steel, see "Manual of Metal-Cutting," published by Am. Soc. M. E., 1939.

**Table 1.—Dimensions of Round-nose Forged Single-point Tools**

Left-cut straight tool with raised face. Barth standardization. See Fig. 9

| Tool size, in., *W* | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 |
|---|---|---|---|---|---|---|---|---|---|
| Blunt Tool for Hard Steel and Cast Iron | | | | | | | | | |
| Nose radius, in., *R* | 0.101 | 0.169 | 0.236 | 0.303 | 0.371 | 0.506 | 0.641 | 0.776 | 0.911 |
| Flank radius, in., *F* | 0.820 | 0.888 | 0.955 | 1.022 | 1.090 | 1.225 | 1.360 | 1.495 | 1.630 |
| Sharp Tool for Medium and Soft Steel | | | | | | | | | |
| Nose radius, in., *R* | 0.072 | 0.126 | 0.180 | 0.235 | 0.289 | 0.397 | 0.506 | 0.614 | 0.723 |
| Flank radius, in., *F* | 0.791 | 0.845 | 0.899 | 0.954 | 1.008 | 1.116 | 1.225 | 1.333 | 1.442 |

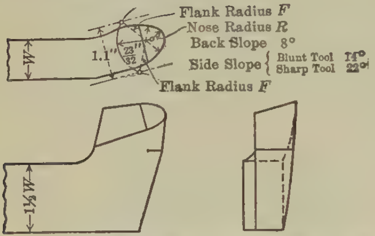


FIG. 9. Left-cut, Straight-shank, Round-nose Tool with Raised Face

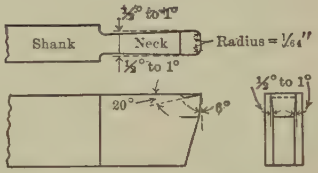


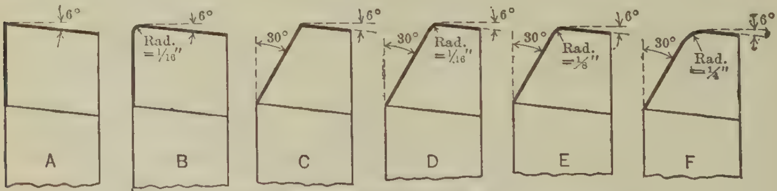
FIG. 10. Recommended Shape of Parting Tool

The contours shown in Fig. 11 are suitable and recommended for right-cut tool bits, or small solid tools ground directly from the bar. A similar set of tools, with the cutting edge on the right-hand side of the tool, would be necessary for cutting from left to right.

For facing operations the active cutting edge is on the end of the tool, preferably with a back rake equal to the side rake used on ordinary roughing tools, and with a side slope of approximately 8 deg. The right-cut contours shown in Fig. 12 should be ample for all ordinary operations. A similar set, with the active cutting edge on the opposite side of the tool, would be required for left-cut tools.

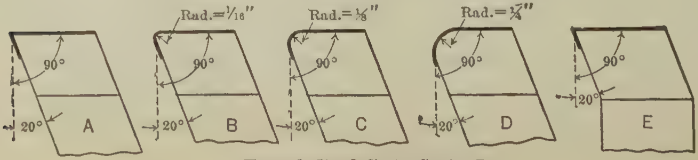
Tools for finishing cuts should have an appreciable flat on the end, substantially wider than the feed per revolution or stroke. See Fig. 13.

**RAKE ANGLE.**—There seems to be little necessity for making any considerable variation in rake and relief. While cutting pressures are decreased by an increase in rake, the cutting speed is not greatly affected by such changes. While the chip is removed with less expenditure of energy by the tool having the greater rake, the reduction in cutting angle tends to weaken the tool, and so causes failure sooner than would be the case if a smaller rake were used.



Heavy Outline Indicates Cutting Edge

FIG. 11. Recommended Contour, Right-cut Roughing Tool

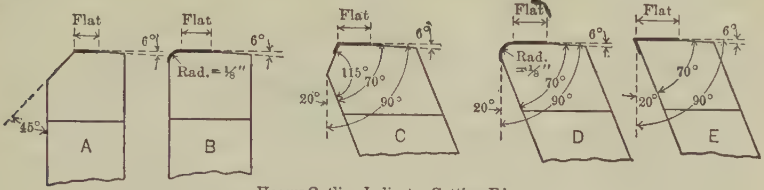


Heavy Outline Indicates Cutting Edge

FIG. 12. Recommended Contour, Right-cut Facing Tool

The values of rake given in Table 2 are based on the practice of a leading manufacturing company, for use with high-speed steel tools.

Table 3 shows the practice recommended by Fred H. Colvin and Frank A. Stanley (Turning and Boring Practice; McGraw-Hill Book Co.) for use with cemented carbide tools.



Heavy Outline Indicates Cutting Edge

FIG. 13. Recommended Contour, Right-cut Finishing Tool

## 3. CUTTING SPEEDS AND FEEDS

**ECONOMIC TOOL LIFE.**—Under a given set of conditions, it may be shown that there is a particular tool life which gives lowest overall costs. This tool life, and the accompanying cutting speed may be calculated, if all data are available. The values given in Table 5, however, may be used without great loss in efficiency.

While sufficient data are not available (1937) for exact figures, a tool life of from three to four times as great as is desirable for use with high-speed steel should be used in determining operating conditions for cemented carbide tools.

**Table 2.—Rake and Relief Angles for High-speed Steel Tools Cutting Various Metals**

| Metal Cut | Operation | Back Rake, deg. | Side Rake, deg. | Side Relief, deg. | End Relief, deg. |
|---|---|---|---|---|---|
| Steel, soft | Roughing | 6–10 | 14–22 | 5–9 | 5–9 |
| | Finishing | 14–22 | 0 | 0 | 5–9 |
| | Parting | 14–22 | 0 | 1/2–2 | 5–9 |
| | Forming | 14–22 | 0 | 0 | 3–5 |
| Steel, hard | Roughing | 4–8 | 10–14 | 5–9 | 5–9 |
| | Finishing | 8–14 | 0 | 0 | 5–9 |
| | Parting | 8–14 | 0 | 1/2–2 | 5–9 |
| | Forming | 4–8 | 0 | 0 | 3–5 |
| Steel, very hard | Roughing | 3–7 | 5–10 | 5–9 | 5–9 |
| | Finishing | 5–10 | 0 | 0 | 5–9 |
| | Parting | 5–10 | 0 | 1/2–2 | 5–9 |
| | Forming | 0–5 | 0 | 0 | 3–5 |
| Cast iron | Roughing | 8 | 14 | 5–9 | 5–9 |
| | Finishing | 6–10 | 0 | 0 | 5–9 |
| | Parting | 4–8 | 0 | 1/2–2 | 5–9 |
| | Forming | 4–8 | 0 | 0 | 3–5 |
| Brass and Bronze | Roughing | 6–8 | 4–10 | 5–9 | 5–9 |
| | Finishing | 14–22 | 0 | 0 | 5–9 |
| | Parting | 6–12 | 0 | 1/2–2 | 5–9 |
| | Forming | 12–20 | 0 | 0 | 3–5 |
| Copper | Roughing | 8–12 | 16–28 | 5–9 | 5–9 |
| | Finishing | 8 | 16–28 | 0 | 5–9 |
| | Parting | 6–12 | 0 | 1/2–2 | 5–9 |
| | Forming | 12–20 | 0 | 0 | 3–5 |
| Monel Metal | Roughing | 4–8 | 10–14 | 5–9 | 5–9 |
| | Finishing | 14–22 | 0 | 0 | 5–9 |
| | Parting | 4–8 | 0 | 1/2–2 | 5–9 |
| | Forming | 4–8 | 0 | 0 | 3–5 |
| Aluminum | Roughing | 8 | 16–22 | 5–9 | 5–9 |
| | Finishing | 8 | 16–22 | 0 | 5–9 |
| | Parting | 6–12 | 0 | 1/2–2 | 5–9 |
| | Forming | 12–20 | 0 | 0 | 3–5 |
| Magnesium alloys | Roughing | 5–8 | 3–5 | 6–10 | 6–10 |
| | Finishing | 10–15 | 0 | 0 | 6–10 |
| | Parting | 10–15 | 0 | 6–10 | 6–10 |
| | Forming | 5–10 | 0 | 0 | 6–10 |

## Table 3.—Rake and Relief Angles for Cemented Carbide Tools Cutting Various Materials

| Material Cut | Tensile Strength, lb. per sq. in. | Tool Angles, deg. | | |
|---|---|---|---|---|
| | | Relief | Side Rake | Back Rake |
| Cast iron, 150–170 Brinell hardness number | 18,000–26,000 | 4 | 10–12 | 0–4 |
| “ “ 170–195 “ “ “, 1.5% Ni | 20,000–28,000 | 4 | 8–10 | 0–4 |
| “ “ 210 “ “ “ 1% Cr, 3.5% Ni | 30,000–36,000 | 4 | 4–6 | 0 |
| Cast iron, chilled rolls | ........... | 3 | 0 | 0–3 |
| Malleable cast iron | ........... | 5 | 10–12 | 6 |
| Steel, S.A.E. 1020, soft forging | 63,000–80,000 | 4–6 | 10–14 | 5–7 |
| “ S.A.E. 1035 | 75,000–90,000 | 4–6 | 10–12 | 5–6 |
| “ S.A.E. 1050 | 80,000–100,000 | 4–6 | 10–12 | 5–6 |
| “ S.A.E. 1112, Bessemer screw stock | 70,000–90,000 | 5–8 | 12–20 | 6–10 |
| “ S.A.E. 1120, open-hearth screw stock | 70,000–85,000 | 5–8 | 10–16 | 5–8 |
| “ S.A.E. 2315 gear blanks | 80,000–115,000 | 4–6 | 10–12 | 5–6 |
| “ S.A.E. 3120 | 80,000–110,000 | 4–6 | 10–12 | 5–6 |
| “ S.A.E. 5200 | 100,000–125,000 | 4–6 | 10–12 | 5–6 |
| “ S.A.E. 6150 | 125,000–150,000 | 4–6 | 8–10 | 4–5 |
| “ structural, 20–30% steel scrap, 2% Si, 170–195 Brinell hardness number | 30,000–36,000 | 4 | 6–8 | 0–2 |
| Stainless steel, 18% Cr, 8% Ni | 85,000–110,000 | 4–6 | 10–14 | 5–7 |
| Aluminum, pure cast, No. 43 | 19,000 | 8–10 | 12–16 | 25–40 |
| Duralumin, No. 17 | 58,000 | 8 | 0–3 | 6 |
| Copper, rolled | 30,000 | 8–12 | 18–25 | 4 |
| Brass, cast yellow | 25,000 | 6–8 | ...... | ...... |
| Bronze, S.A.E. 62, hard, cast | 30,000 | 6 | 6–10 | 0 |
| “ phosphor-, S.A.E. 64 | 25,000 | 6 | 4–8 | 0 |
| Clay and porcelain, unfired | ........... | 15–30 | ...... | 10–15 |
| Mica and copper commutators | ........... | 8–10 | 16–20 | 10–15 |
| Bakelite | ........... | 8–10 | 8–12 | 4–6 |
| Casein products | ........... | 8–12 | 4–6 | 0–3 |
| Rubber, hard | ........... | 8–10 | ...... | ...... |

## Table 4.—Economic Tool Life Under Cut

High-speed steel tools, cutting steel and cast iron

| Machine | Tool Setting | Economic Tool Life in Minutes |
|---|---|---|
| Engine Lathe or Planer | Ordinary | 40 |
| | Accurate | 120 |
| Drill Press | Ordinary | 30 |
| Hand Screw Machine | Ordinary | 120 |
| Automatic Screw Machine | Ordinary | 400 |
| Automatic Milling Machine, simple cutters | ................ | 600 |
| Milling Machine, ordinary cutters | ................ | 250 |
| “ “ large forming cutters | ................ | 2000 |

## Table 5.—Effect on Tool Life of Variations in Cutting Speed

| Tool Life, min. | Carbon Steel Tools | | | High-speed Steel Tools | | | | | | Stellite Tools | | | Cemented Carbide Tools | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Metal Cut | | | | | | | | | | | | | | |
| | Steel | | Cast Iron | Steel | | Cast Iron | Bronze | | Brass | Steel | | Cast Iron | Steel | | Cast Iron |
| | Rough | Finish | Rough, Finish | Rough, | Finish | Rough, Finish | Rough | Finish | Rough, Finish | Rough | Finish | Rough, Finish | Rough | Finish | Rough, Finish |
| | Tool Life Factor *F* | | | | | | | | | | | | | | |
| 15 | 1.32 | 1.11 | 1.11 | 1.19 | 1.15 | 1.15 | 1.34 | 1.26 | 1.26 | 1.29 | 1.22 | 1.22 | 1.25 | 1.20 | 1 20 |
| 30 | 1.15 | 1.05 | 1.05 | 1.09 | 1.07 | 1.07 | 1.16 | 1.12 | 1.12 | 1.13 | 1.11 | 1.11 | 1.12 | 1.09 | 1.09 |
| 45 | 1.06 | 1.02 | 1.02 | 1.04 | 1.05 | 1.03 | 1.06 | 1.05 | 1.05 | 1.05 | 1.04 | 1.04 | 1.05 | 1.04 | 1.04 |
| 60 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 |
| 90 | 0.92 | 0.98 | 0.98 | 0.95 | 0.96 | 0.96 | 0.94 | 0.92 | 0.92 | 0.93 | 0.94 | 0.94 | 0.94 | 0.95 | 0.95 |
| 120 | .87 | .95 | .95 | .92 | .93 | .93 | .86 | .89 | .89 | .88 | .91 | .91 | .90 | .91 | .91 |
| 180 | .80 | .92 | .92 | .87 | .89 | .89 | .79 | .83 | .83 | .82 | .86 | .86 | .84 | .87 | .87 |
| 240 | .76 | .90 | .90 | .84 | .87 | .87 | .75 | .79 | .79 | .78 | .82 | .82 | .80 | .84 | 84 |
| 300 | .72 | .89 | .89 | .82 | .85 | .85 | .71 | .77 | .77 | .75 | .79 | .79 | .77 | .81 | .81 |
| 360 | .70 | .87 | .87 | .80 | .84 | .84 | .69 | .74 | .74 | .72 | .77 | .77 | .75 | .79 | .79 |
| 400 | .68 | .86 | .86 | .78 | .82 | .82 | .66 | .72 | .72 | .70 | .75 | .75 | .73 | .78 | .78 |
| 480 | .66 | .86 | .86 | .77 | .81 | .81 | .65 | .71 | .71 | .69 | .74 | .74 | .72 | .76 | .76 |

Table 5 shows the effect of variations in cutting speed on tool life. This table is based on a value of unity for a tool life of 60 minutes. If the proper cutting speed is known for a cut $A$ of given duration, the speed for a cut $B$ of different duration may be obtained by the formula

$$S_b = S_a \times (F_b/F_a)$$

where $S_b$, $S_a$ = respectively cutting speeds of $A$ and $B$, and $F_a$, $F_b$ = respectively tool life factors for $A$ and $B$ from Table 5.

Example.—A cutting speed of 95 ft. per min. in rough turning steel with high-speed steel tools gives a tool life of 45 min. Required the cutting speed to give a tool life of 120 min. *Solution.*—$A = 45$; $B = 120$; $S_a = 95$. From Table 5 tool life factor $F_a = 1.04$, and $F_b = 0.92$. Then $S_b = 95 \times (0.92/1.04) = 84$ ft. per min.

**ECONOMIC PRODUCTION.**—To keep costs at a minimum in rough machining operations, the maximum volume of chips per minute must be removed. This involves: 1. The maximum depth of cut that can be taken without distortion of the part machined, up to the point where all surplus stock is removed in a single roughing cut. 2. A relatively large feed, which gives thick chips, preferably above 0.030 in. 3. The use of a tool having a contour which gives the smallest possible chip thickness for a given feed without setting up injurious chatter. Increase in the nose radius or the side-cutting-edge angle decreases chip thickness under constant conditions of feed and depth of cut. 4. A cutting speed such that the life of the tool approximates the economic life.

**MATERIALS CUT.**—Tables 6, 7 and 9 give the approximate relative machinability, based on the speed with which a given cut may be taken, of a large number of common materials. These tables may be used to determine the correct cutting speed and horsepower requirements for the various materials under a variety of working conditions, by use of the multipliers for speed and horsepower in connection with the basic data given in Tables 10 to 18. Table 6 is based on S.A.E. 1030 hot-rolled steel (see Tables 11–13). Table 7 is based on cast iron of 177 Brinell hardness and tensile strength of 30,000 lb. per sq. in. (see Tables 14 to 17).

Example.—Required the correct cutting speed and horsepower for turning S.A.E. 2330 steel, normalized, using a round-nose tool set at an angle of 30 deg. to the work, with a $^1/_8$ in. depth of cut and a $^1/_{32}$ in. feed. *Solution.*—From Table 12, the desirable cutting speed for S.A.E. 1030 hot-rolled steel, under the conditions given, is 80 ft. per min. The horsepower required at the nose of the tool is 2.5. From Table 6, the multiplier for speed for S.A.E. 2330 normalized steel is 0.68. The multiplier for horsepower is 0.65. Cutting speed for S.A.E. 2330 normalized steel = 80 × 0.68 = 54.4 ft. per min. Horsepower required at nose of tool = 2.5 × 0.65 = 1.63.

**TOOL PRESSURE.**—F. W. Taylor (*Trans.* A.S.M.E., xxviii, 1908) found that no definite relation exists between the cutting speed of tools and the pressure, tangential to the finished surface of the work, exerted on the tool by the chip.

The value of this tangential component, which is of interest in the design of machine tools is

$$P = K_1L^{1.1}T^{0.78}, \text{ for cutting steel, and}$$

$$P = K_2LT^{0.65}, \text{ for cutting cast iron,}$$

where $K_1$, $K_2$ = constants, given in Tables 6 and 7; $L$ = total width of chip, in., which for the average tool is equal to the total length of cutting edge of the tool in contact with the work (see Fig. 14); $T$ = average thickness of the chip, in. This usually equals (Feed × Depth)/$L$.

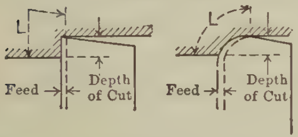


Fig. 14. Determination of Factor $L$ in Formula for Tool Pressure

**METAL CUTTING TESTS.**—Metal cutting tests, to be on a thoroughly scientific basis, require the consumption of much more time and material than usually is available in the average shop. For detailed discussion of the subject see Test Code for High-speed Steel for Turning Tools, *Mech. Engg.*, March, 1931, and Standardization of Testing Conditions for Metal Cutting Tools, *Mech. Engg.*, Sept., 1932. Also, A Metal Cutting Test Metal for Machine Shop Use, *Iron Age*, May 13, 1937.

**CUTTING SPEED TABLES.**—Tables 11, 12 and 13 are based on the cutting of S.A.E. 1030 hot-rolled machinery steel with an 18–4–1 high-speed steel tool. The horsepower given is that developed at the nose of the tool, and does not include any losses in the machine. Tables 15, 16 and 17 are based on the cutting of gray cast iron, Brinell hardness No. 177. The tools used were 18–4–1 high-speed steel tool. See Table 22.

**SPEEDS AND FEEDS FOR MONEL METAL.**—Table 8, giving speeds for Monel metal, is based on experiments by the International Nickel Co., and by Melvin Matson at Montana State College. See Melvin Matson, Lathe Cutting of Monel Metal, *Iron Age*, Sept. 10, 1936.

**TOOLS FOR TURNING MONEL METAL** should be of much the same shape as those used for turning medium steel. Because of the toughness of the chip it is particularly desirable to use a chip breaker.

## Table 6.—Machinability Constants of Steel

For analysis of the various steels listed, see p. 2–42.

| Steel, S.A.E. No. | Condition* | Heat Treatment | | | Physical Properties | | | | | Chip Pressure Constant $K_1$ | Multipliers for S.A.E. 1030 HR Values. See Tables 10 to 13 | | Relative Machinability |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Heat to, deg. F. | Quench in | Draw at, deg. F. | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elong. in 2 in., percent | Reduction of Area, percent | Brinell Hardness No. | | Speed | Hp. | |
| 1010 | HR | .... | ........ | ...... | 55,000 | 35,000 | 35 | 55 | 110 | 114,000 | 1.82 | 1.59 | 8.5 |
| 1010 | CD | .... | ........ | ...... | 66,000 | 60,000 | 23 | 51 | 141 | 125,000 | 1.19 | 1.16 | 5.6 |
| 1015 | HR | .... | ........ | ...... | 57,000 | 37,000 | 33 | 55 | 143 | 116,000 | 1.71 | 1.53 | 8.0 |
| 1015 | CD | .... | ........ | ...... | 73,000 | 62,000 | 22 | 51 | 150 | 127,000 | 1.00 | 0.99 | 4.7 |
| 1020 | HR | .... | ........ | ...... | 66,000 | 32,000 | 35 | 52 | 127 | 122,000 | 1.28 | 1.21 | 6.0 |
| 1020 | CD | .... | ........ | ...... | 78,000 | 65,000 | 21 | 50 | 160 | 131,000 | 0.91 | 0.92 | 4.2 |
| X1020 | HR | .... | ........ | ...... | 61,000 | 43,000 | 31 | 61 | 126 | 119,000 | 1.51 | 1.39 | 6.8 |
| X1020 | CD | .... | ........ | ...... | 86,000 | 80,000 | 6 | 57 | 156 | 138,000 | 1.33 | 1.42 | 6.2 |
| 1025 | HR | .... | ........ | ...... | 66,000 | 39,000 | 29 | 54 | 160 | 122,000 | 1.25 | 1.21 | 5.8 |
| 1025 | CD | .... | ........ | ...... | 82,000 | 67,000 | 20 | 50 | 168 | 135,000 | 0.85 | 0.88 | 3.9 |
| 1025 | HT | 1600 | Water | 900 | 80,000 | 56,000 | 28 | 59 | 168 | 133,000 | 0.94 | 0.95 | 4.3 |
| 1030 | HR | .... | ........ | ...... | 75,000 | 47,000 | 26 | 50 | 168 | 129,000 | 1.00 | 1.00 | 4.7 |
| 1030 | CD | .... | ........ | ...... | 85,000 | 70,000 | 19 | 50 | 180 | 137,000 | 0.79 | 0.83 | 3.7 |
| 1030 | HT | 1600 | Water | 900 | 89,000 | 63,000 | 26 | 56 | 185 | 140,000 | 0.81 | 0.87 | 3.7 |
| 1035 | HR | .... | ........ | ...... | 82,000 | 49,000 | 25 | 51 | 174 | 135,000 | 0.88 | 0.93 | 4.1 |
| 1035 | HT | 1550 | Water | 1300 | 85,000 | 50,000 | 34 | 68 | 174 | 137,000 | 0.87 | 0.92 | 4.1 |
| 1035 | HT | 1550 | Water | 860 | 105,000 | 75,000 | 23 | 52 | 212 | 152,000 | 0.62 | 0.72 | 2.9 |
| 1040 | HR | .... | ........ | ...... | 90,000 | 53,000 | 24 | 41 | 187 | 141,000 | 0.78 | 0.84 | 3.6 |
| 1040 | CD | .... | ........ | ...... | 92,000 | 75,000 | 18 | 46 | 195 | 142,000 | 0.72 | 0.78 | 3.3 |
| 1040 | HT | 1550 | Water | 1300 | 92,000 | 55,000 | 31 | 64 | 186 | 142,000 | 0.76 | 0.83 | 3.5 |
| 1040 | HT | 1550 | Water | 800 | 113,000 | 83,000 | 19 | 49 | 230 | 158,000 | 0.55 | 0.66 | 2.5 |
| 1045 | HR | .... | ........ | ...... | 93,000 | 56,000 | 26 | 55 | 187 | 143,000 | 0.77 | 0.83 | 3.5 |
| 1045 | N | .... | ........ | ...... | 93,000 | ....... | ...... | ...... | 185 | 143,000 | 0.78 | 0.86 | 3.6 |
| 1045 | HT | 1500 | Water | 1300 | 98,000 | 60,000 | 28 | 56 | 197 | 147,000 | 0.79 | 0.89 | 3.7 |
| 1045 | HT | 1500 | Water | 800 | 121,000 | 91,000 | 15 | 46 | 248 | 165,000 | 0.49 | 0.61 | 2.2 |
| 1050 | HR | .... | ........ | ...... | 103,000 | 60,000 | 18 | 41 | 201 | 150,000 | 0.63 | 0.72 | 2.9 |
| 1050 | HT | 1500 | Water | 1300 | 104,000 | 65,000 | 25 | ...... | 208 | 151,000 | 0.59 | 0.69 | 2.7 |
| 1050 | HT | 1500 | Water | 800 | 129,000 | 99,000 | 11 | ...... | 266 | 171,000 | 0.42 | 0.55 | 1.9 |
| 1060 | HR | .... | ........ | ...... | 117,000 | 71,000 | 16 | 26 | 217 | 162,000 | 0.54 | 0.66 | 2.5 |
| 1060 | N | .... | ........ | ...... | 110,000 | ....... | ...... | ...... | 225 | 156,000 | 0.59 | 0.71 | 2.7 |
| 1060 | HT | 1500 | Water | 1300 | 116,000 | 74,000 | 19 | ...... | 230 | 161,000 | 0.49 | 0.60 | 2.2 |
| 1060 | HT | 1500 | Water | 800 | 144,000 | 114,000 | ...... | ...... | 315 | 182,000 | 0.28 | 0.39 | 1.3 |
| 1095 | HR | .... | ........ | ...... | 147,000 | 74,000 | 11 | 18 | 280 | 184,000 | 0.38 | 0.54 | 1.8 |
| 1095 | N | 1500 | Furnace | cooled | 93,000 | 45,000 | 6 | 7 | 207 | 143,000 | 0.53 | 0.58 | 2.4 |
| 1112 | CD | .... | ........ | ...... | 88,000 | 75,000 | 20 | 45 | 167 | 84,000 | 1.76 | 1.13 | 8.2 |
| X1112 | CD | .... | ........ | ...... | 88,000 | 75,000 | 20 | 45 | 183 | 84,000 | 2.38 | 1.54 | 11.1 |
| 1115 | HR | .... | ........ | ...... | 75,000 | 50,000 | 30 | 53 | ...... | 75,000 | 2.40 | 1.39 | 10.0 |
| 1120 | HR | .... | ........ | ...... | 68,000 | 39,000 | 30 | 50 | 120 | 70,000 | 1.82 | 0.98 | 8.5 |
| 1120 | CD | .... | ........ | ...... | 78,000 | 68,000 | 20 | 50 | 152 | 97,000 | 1.19 | 0.89 | 5.6 |
| X1314 | HR | .... | ........ | ...... | 80,000 | 68,000 | 23 | 52 | 167 | 79,000 | 1.77 | 1.07 | 8.2 |
| X1315 | HR | .... | ........ | ...... | 67,000 | 43,000 | 42 | 64 | 120 | 70,000 | 3.10 | 1.67 | 14.5 |
| X1315 | CD | .... | ........ | ...... | 90,000 | 75,000 | 20 | 45 | 161 | 85,000 | 1.81 | 1.18 | 8.6 |
| X1330 | HR | .... | ........ | ...... | 83,000 | 42,000 | 20 | 40 | 165 | 81,000 | 1.77 | 1.09 | 8.2 |
| X1330 | CD | .... | ........ | ...... | 97,000 | 73,000 | 14 | 32 | 200 | 91,000 | 1.26 | 0.88 | 5.9 |
| X1335 | HR | .... | ........ | ...... | 92,000 | 57,000 | 35 | 41 | 197 | 87,000 | 1.31 | 0.88 | 6.1 |
| X1335 | CD | .... | ........ | ...... | 90,000 | 75,000 | 20 | 45 | 197 | 86,000 | 1.31 | 0.87 | 6.1 |
| X1335 | HT | 1525 | Oil | 1000 | 114,000 | 88,000 | 21 | 57 | 229 | 101,000 | 1.00 | 0.77 | 4.6 |
| T1330 | HR | .... | ........ | ...... | 111,000 | 73,000 | 22 | 57 | 212 | 157,000 | 0.91 | 1.09 | 4.2 |
| T1330 | HT | 1550 | Oil | 1000 | 139,000 | 121,000 | 20 | 53 | 277 | 178,000 | 0.60 | 0.81 | 2.7 |
| T1335 | HT | 1550 | Oil | 1000 | 129,000 | 103,000 | 17 | 52 | 262 | 171,000 | 0.64 | 0.84 | 3.0 |
| T1340 | HR | .... | ........ | ...... | 115,000 | 75,000 | 20 | 54 | 217 | 160,000 | 0.87 | 1.08 | 4.1 |
| T1340 | HT | 1525 | Oil | 1000 | 135,000 | 106,000 | 15 | 51 | 277 | 176,000 | 0.60 | 0.80 | 2.7 |
| T1345 | HT | 1500 | Oil | 1000 | 141,000 | 108,000 | 13 | 50 | 241 | 180,000 | 0.54 | 0.75 | 2.4 |
| T1350 | HT | 1500 | Oil | 1000 | 147,000 | 112,000 | 11 | 49 | 305 | 184,000 | 0.49 | 0.70 | 2.3 |
| 2015 | HR | .... | ........ | ...... | 73,000 | 53,000 | 34 | 67 | 130 | 120,000 | 1.09 | 1.01 | 5.1 |
| 2015 | N | .... | ........ | ...... | 60,000 | 37,000 | 41 | 70 | ...... | 118,000 | 1.35 | 1.23 | 6.3 |
| 2015 | CD | .... | ........ | ...... | 84,000 | 80,000 | 17 | 60 | 168 | 122,000 | 0.78 | 0.74 | 3.6 |
| 2115 | HR | .... | ........ | ...... | 70,000 | 47,000 | 30 | 65 | ...... | 120,000 | 1.14 | 1.06 | 5.3 |
| 2115 | N | .... | ........ | ...... | 64,000 | 40,000 | 39 | 71 | ...... | 119,000 | 0.88 | 0.81 | 4.1 |
| 2115 | CD | .... | ........ | ...... | 94,000 | 85,000 | 17 | 60 | 175 | 124,000 | 0.74 | 0.70 | 3.4 |

* A = annealed; CD = cold drawn; HR = hot rolled; HT = heat treated; N = normalized.

Table 6.—Machinability Constants of Steel—*Continued*

| Steel, S.A.E. No. | Condition* | Heat Treatment | | | Physical Properties | | | | | Chip Pressure Constant $K_1$ | Multipliers for S.A.E. 1030 HR Values. See Tables 10 to 13 | | Relative Machinability |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Heat to, deg. F. | Quench in | Draw at, deg. F. | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elong. in 2 in., percent | Reduction of Area, percent | Brinell Hardness No. | | Speed | Hp. | |
| 2315 | HR | .... | ........ | ...... | 76,000 | 55,000 | 31 | 65 | 153 | 121,000 | 1.13 | 1.06 | 5.3 |
| 2315 | N | .... | ........ | ...... | 75,000 | 52,000 | 34 | 67 | ...... | 121,000 | 1.05 | 0.98 | 4.9 |
| 2315 | CD | .... | ........ | ...... | 87,000 | 78,000 | 17 | 61 | 192 | 123,000 | 0.74 | 0.70 | 3.4 |
| 2320 | HR | .... | ........ | ...... | 86,000 | 63,000 | 27 | 65 | 183 | 123,000 | 0.85 | 0.80 | 3.9 |
| 2320 | N | .... | ........ | ...... | 84,000 | 58,000 | 31 | 61 | ...... | 122,000 | 0.87 | 0.82 | 4.0 |
| 2320 | CD | .... | ........ | ...... | 104,000 | 100,000 | 15 | 58 | 223 | 126,000 | 0.60 | 0.57 | 2.7 |
| 2330 | HR | .... | ........ | ...... | 105,000 | 68,000 | 21 | 49 | 207 | 126,000 | 0.64 | 0.63 | 3.0 |
| 2330 | N | .... | ........ | ...... | 102,000 | 67,000 | 24 | 50 | ...... | 124,000 | 0.68 | 0.65 | 3.2 |
| 2330 | CD | .... | ........ | ...... | 124,000 | 119,000 | 12 | 43 | 223 | 136,000 | 0.46 | 0.48 | 2.1 |
| 2330 | HT | 1450 | Oil | 1000 | 129,000 | 106,000 | 13 | 36 | 271 | 139,000 | 0.51 | 0.55 | 2.4 |
| 2330 | HT | 1450 | Oil | 800 | 160,000 | 128,000 | 18 | 48 | 324 | 172,000 | 0.42 | 0.55 | 1.9 |
| 2335 | HR | .... | ........ | ...... | 108,000 | 63,000 | 23 | 56 | 195 | 127,000 | 0.72 | 0.70 | 3.3 |
| 2335 | N | .... | ........ | ...... | 115,000 | 77,000 | 23 | 47 | ...... | 129,000 | 0.60 | 0.59 | 2.8 |
| 2335 | HT | 1450 | Oil | 1000 | 132,000 | 108,000 | 22 | 60 | 275 | 142,000 | 0.53 | 0.58 | 2.4 |
| 2335 | HT | 1450 | Oil | 800 | 170,000 | 147,000 | 17 | 54 | 335 | 186,000 | 0.40 | 0.57 | 1.8 |
| 2340 | HR | .... | ........ | ...... | 114,000 | 78,000 | 19 | 42 | 240 | 129,000 | 0.58 | 0.59 | 2.7 |
| 2340 | N | .... | ........ | ...... | 106,000 | 74,000 | 22 | 47 | 223 | 134,000 | 0.64 | 0.66 | 3.0 |
| 2340 | HT | 1425 | Oil | 1000 | 136,000 | 120,000 | 21 | 60 | 300 | 147,000 | 0.49 | 0.55 | 2.3 |
| 2340 | HT | 1425 | Oil | 800 | 180,000 | 165,000 | 17 | 53 | 376 | 200,000 | 0.38 | 0.59 | 1.8 |
| 2345 | HR | .... | ........ | ...... | 129,000 | 94,000 | 16 | 35 | 260 | 140,000 | 0.54 | 0.58 | 2.5 |
| 2345 | N | .... | ........ | ...... | 126,000 | 82,000 | 20 | 39 | ...... | 137,000 | 0.53 | 0.55 | 2.4 |
| 2345 | HT | 1425 | Oil | 1100 | 125,000 | 113,000 | 22 | 62 | 273 | 137,000 | 0.55 | 0.58 | 2.5 |
| 2345 | HT | 1425 | Oil | 900 | 160,000 | 150,000 | 18 | 55 | 340 | 172,000 | 0.40 | 0.53 | 1.8 |
| 2350 | HR | .... | ........ | ...... | 128,000 | 85,000 | 23 | 26 | 269 | 138,000 | 0.53 | 0.55 | 2.4 |
| 2350 | N | .... | ........ | ...... | 134,000 | 87,000 | 17 | 33 | ...... | 146,000 | 0.49 | 0.54 | 2.2 |
| 2350 | HT | 1425 | Oil | 1100 | 126,000 | 115,000 | 21 | 60 | 280 | 137,000 | 0.53 | 0.55 | 2.4 |
| 2350 | HT | 1425 | Oil | 900 | 160,000 | 150,000 | 16 | 54 | 350 | 172,000 | 0.41 | 0.54 | 1.9 |
| 2512 | HR | .... | ........ | ...... | 105,000 | 73,000 | 27 | 52 | 210 | 126,000 | 0.64 | 0.63 | 3.0 |
| 2512 | N | .... | ........ | ...... | 84,000 | 58,000 | 32 | 64 | ...... | 122,000 | 0.90 | 0.84 | 4.2 |
| 2512 | CD | .... | ........ | ...... | 140,000 | ........ | ...... | ...... | ...... | 130,000 | 0.68 | 0.67 | 3.1 |
| 2515 | HR | .... | ........ | ...... | 91,000 | 68,000 | 28 | 69 | 179 | 123,000 | 0.81 | 0.76 | 3.7 |
| 2515 | N | .... | ........ | ...... | 91,000 | 69,000 | 29 | 69 | 179 | 123,000 | 0.81 | 0.76 | 3.7 |
| 2515 | CD | .... | ........ | ...... | 115,000 | 75,000 | 28 | 60 | 226 | 130,000 | 0.62 | 0.63 | 2.9 |
| 3115 | HR | .... | ........ | ...... | 71,000 | 46,000 | 32 | 69 | 134 | 90,000 | 1.00 | 0.70 | 4.6 |
| 3115 | N | .... | ........ | ...... | 72,000 | 59,000 | 23 | 69 | 128 | 90,000 | 1.00 | 0.70 | 4.7 |
| 3115 | CD | .... | ........ | ...... | 77,000 | 68,000 | 25 | 68 | 163 | 92,000 | 0.85 | 0.60 | 3.9 |
| 3120 | HR | .... | ........ | ...... | 88,000 | 56,000 | 29 | 57 | 149 | 95,000 | 0.73 | 0.53 | 3.4 |
| 3120 | N | .... | ........ | ...... | 85,000 | 63,000 | 21 | 65 | 144 | 94,000 | 0.83 | 0.61 | 3.9 |
| 3120 | CD | .... | ........ | ...... | 98,000 | 80,000 | 20 | 63 | 174 | 119,000 | 0.60 | 0.55 | 2.8 |
| 3125 | HR | .... | ........ | ...... | 91,000 | 57,000 | 27 | 62 | 164 | 99,000 | 0.69 | 0.53 | 3.2 |
| 3125 | N | .... | ........ | ...... | 90,000 | 67,000 | 29 | 62 | 160 | 96,000 | 0.71 | 0.53 | 3.3 |
| 3125 | CD | .... | ........ | ...... | 111,000 | 92,000 | 16 | 58 | 187 | 127,000 | 0.50 | 0.49 | 2.3 |
| 3130 | HR | .... | ........ | ...... | 99,000 | 63,000 | 25 | 60 | 178 | 122,000 | 0.62 | 0.58 | 2.9 |
| 3130 | N | .... | ........ | ...... | 97,000 | 71,000 | 26 | 58 | 176 | 116,000 | 0.64 | 0.58 | 3.0 |
| 3130 | CD | .... | ........ | ...... | 118,000 | 113,000 | 14 | 55 | 210 | 132,000 | 0.45 | 0.46 | 2.1 |
| 3130 | HT | 1525 | Oil | 1000 | 131,000 | 102,000 | 20 | 62 | 262 | 141,000 | 0.45 | 0.48 | 2.1 |
| 3130 | HT | 1525 | Oil | 800 | 166,000 | 138,000 | 14 | 54 | 335 | 180,000 | 0.33 | 0.46 | 1.5 |
| 3140 | HR | .... | ........ | ...... | 112,000 | 74,000 | 20 | 50 | 228 | 128,000 | 0.53 | 0.52 | 2.4 |
| 3140 | N | .... | ........ | ...... | 110,000 | 79,000 | 21 | 50 | 207 | 127,000 | 0.54 | 0.49 | 2.5 |
| 3140 | CD | .... | ........ | ...... | 125,000 | 119,000 | 13 | 45 | 241 | 137,000 | 0.42 | 0.45 | 1.9 |
| 3140 | HT | 1500 | Oil | 1000 | 143,000 | 118,000 | 18 | 57 | 285 | 155,000 | 0.41 | 0.48 | 1.9 |
| 3140 | HT | 1500 | Oil | 800 | 175,000 | 157,000 | 13 | 48 | 370 | 195,000 | 0.31 | 0.47 | 1.4 |
| X3140 | HR | .... | ........ | ...... | 112,000 | 66,000 | 20 | 53 | 210 | 128,000 | 0.60 | 0.59 | 2.8 |
| X3140 | A | .... | ........ | ...... | 102,000 | 68,000 | 26 | 55 | 207 | 125,000 | 0.62 | 0.60 | 2.9 |
| X3140 | HT | 1500 | Oil | 1100 | 130,000 | 106,000 | 18 | 56 | 255 | 140,000 | 0.49 | 0.52 | 2.2 |
| X3140 | HT | 1500 | Oil | 840 | 186,000 | 166,000 | 12 | 47 | 384 | 213,000 | 0.29 | 0.48 | 1.4 |
| 3145 | HR | .... | ........ | ...... | 116,000 | 79,000 | 17 | 48 | 210 | 137,000 | 0.49 | 0.51 | 2.2 |
| 3145 | N | .... | ........ | ...... | 117,000 | 84,000 | 19 | 48 | 223 | 131,000 | 0.49 | 0.49 | 2.3 |
| 3145 | HT | 1475 | Oil | 1100 | 128,000 | 108,000 | 19 | 58 | 259 | 139,000 | 0.45 | 0.48 | 2.1 |
| 3145 | HT | 1475 | Oil | 900 | 170,000 | 143,000 | 15 | 52 | 332 | 186,000 | 0.33 | 0.47 | 1.5 |
| 3150 | HR | .... | ........ | ...... | 120,000 | 85,000 | 15 | 42 | 280 | 133,000 | 0.45 | 0.46 | 2.1 |
| 3150 | N | .... | ........ | ...... | 120,000 | 88,000 | 16 | 43 | 239 | 133,000 | 0.46 | 0.47 | 2.1 |
| 3150 | A | .... | ........ | ...... | 97,000 | 52,000 | 27 | 42 | 179 | 91,000 | 0.65 | 0.46 | 3.0 |

* A = annealed; CD = cold drawn; HR = hot rolled; HT = heat treated; N = normalized.

**Table 6.—Machinability Constants of Steel—*Continued***

| Steel, S.A.E. No. | Condition* | Heat Treatment | | | Physical Properties | | | | | Chip Pressure Constant $K_1$ | Multipliers for S.A.E. 1030 HR Values. See Tables 10 to 13 | | Relative Machinability |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Heat to, deg. F. | Quench in | Draw at, deg. F. | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elong. in 2 in., percent | Reduction of Area, percent | Brinell Hardness No. | | Speed | Hp. | |
| 3150 | HT | 1450 | Oil | 1100 | 132,000 | 112,000 | 18 | 57 | 262 | 142,000 | 0.44 | 0.47 | 2.0 |
| 3150 | HT | 1450 | Oil | 900 | 173,000 | 150,000 | 14 | 50 | 342 | 191,000 | 0.33 | 0.48 | 1.5 |
| 3230 | HR | .... | ........ | ...... | 113,000 | 54,000 | 21 | 60 | 221 | 129,000 | 0.55 | 0.54 | 2.5 |
| 3230 | N | .... | ........ | ...... | 90,000 | 65,000 | 33 | 65 | 174 | 96,000 | 0.74 | 0.55 | 3.5 |
| 3240 | HR | .... | ........ | ...... | 144,000 | 111,000 | 14 | 29 | 285 | 157,000 | 0.41 | 0.49 | 1.9 |
| 3240 | N | .... | ........ | ...... | 146,000 | 111,000 | 15 | 29 | 285 | 158,000 | 0.41 | 0.49 | 1.9 |
| 3240 | A | .... | ........ | ...... | 90,000 | 57,000 | 29 | 69 | 170 | 96,000 | 0.74 | 0.55 | 3.5 |
| 3240 | CD | .... | ........ | ...... | 107,000 | 102,000 | 19 | 65 | 207 | 126,000 | 0.58 | 0.57 | 2.7 |
| 3240 | HT | 1475 | Oil | 1000 | 160,000 | 140,000 | 17 | 55 | 330 | 172,000 | 0.36 | 0.47 | 1.6 |
| 3240 | HT | 1475 | Oil | 800 | 198,000 | 178,000 | 13 | 47 | 400 | 228,000 | 0.28 | 0.49 | 1.3 |
| 3245 | HR | .... | ........ | ...... | 142,000 | 78,000 | 14 | 37 | 263 | 155,000 | 0.41 | 0.48 | 1.9 |
| 3245 | A | 1385 | ........ | ...... | 101,000 | 66,000 | 25 | 62 | ...... | 125,000 | 0.62 | 0.59 | 2.8 |
| 3245 | HT | 1450 | Oil | 1000 | 154,000 | 144,000 | 18 | 55 | 331 | 167,000 | 0.36 | 0.47 | 1.7 |
| 3245 | HT | 1450 | Oil | 800 | 209,000 | 189,000 | 13 | 47 | 408 | 258,000 | 0.27 | 0.53 | 1.2 |
| 3250 | N | .... | ........ | ...... | 118,000 | 79,000 | 22 | 44 | 291 | 132,000 | 0.53 | 0.53 | 2.4 |
| 3250 | A | .... | ........ | ...... | 96,000 | 54,000 | 28 | 55 | 180 | 113,000 | 0.73 | 0.64 | 3.4 |
| 3250 | HT | 1450 | Oil | 1100 | 148,000 | 128,000 | 18 | 56 | 301 | 162,000 | 0.38 | 0.47 | 1.8 |
| 3250 | HT | 1450 | Oil | 900 | 185,000 | 165,000 | 14 | 48 | 372 | 214,000 | 0.31 | 0.58 | 1.4 |
| 3312 | HR | .... | ........ | ...... | 133,000 | 100,000 | 17 | 50 | 296 | 144,000 | 0.41 | 0.46 | 1.9 |
| 3312 | N | .... | ........ | ...... | 160,000 | 120,000 | 17 | 50 | 321 | 172,000 | 0.36 | 0.47 | 1.6 |
| 3312 | A | .... | ........ | ...... | 105,000 | 85,000 | 26 | 72 | 212 | 126,000 | 0.60 | 0.58 | 2.8 |
| 3312 | HT | 1400 | Oil | 1100 | 111,000 | 99,000 | 25 | 73 | 228 | 128,000 | 0.55 | 0.54 | 2.5 |
| 3325 | HT | 1500 | Oil | 1000 | 130,000 | 110,000 | 22 | 62 | 269 | 140,000 | 0.45 | 0.48 | 2.1 |
| 3325 | HT | 1500 | Oil | 800 | 171,000 | 143,000 | 17 | 54 | 352 | 186,000 | 0.33 | 0.47 | 1.5 |
| 3335 | HT | 1450 | Oil | 1000 | 147,000 | 126,000 | 20 | 59 | 302 | 162,000 | 0.40 | 0.49 | 1.8 |
| 3335 | HT | 1450 | Oil | 800 | 188,000 | 169,000 | 12 | 50 | 375 | 222,000 | 0.29 | 0.49 | 1.3 |
| 3340 | N | .... | ........ | ...... | 177,000 | 140,000 | 10 | 28 | 352 | 192,000 | 0.32 | 0.47 | 1.5 |
| 3340 | A | .... | ........ | ...... | 125,000 | 100,000 | 23 | 59 | 255 | 136,000 | 0.49 | 0.51 | 2.2 |
| 3340 | HT | 1425 | Oil | 1000 | 170,000 | 145,000 | 18 | 53 | 325 | 186,000 | 0.36 | 0.51 | 1.6 |
| 3340 | HT | 1425 | Oil | 800 | 211,000 | 183,000 | 13 | 47 | 395 | 258,000 | 0.28 | 0.55 | 1.3 |
| 3415 | N | .... | ........ | ...... | 81,000 | 47,000 | 32 | 65 | 163 | 92,000 | 0.82 | 0.58 | 3.8 |
| 3415 | A | 1545 | ........ | ...... | 75,000 | 46,000 | 40 | 69 | 153 | 91,000 | 0.91 | 0.64 | 4.2 |
| 3415 | HT | 1500 | Oil | 1000 | 130,000 | 112,000 | 21 | 67 | 268 | 140,000 | 0.45 | 0.48 | 2.1 |
| 3415 | HT | 1500 | Oil | 800 | 158,000 | 138,000 | 17 | 59 | 319 | 172,000 | 0.36 | 0.48 | 1.7 |
| 3415 | HT | 1500 | Oil | 600 | 181,000 | 160,000 | 14 | 52 | 360 | 204,000 | 0.31 | 0.48 | 1.4 |
| 3430 | N | .... | ........ | ...... | 170,000 | 130,000 | 15 | 40 | 341 | 186,000 | 0.33 | 0.47 | 1.5 |
| 3435 | N | .... | ........ | ...... | 105,000 | 75,000 | 17 | 57 | 207 | 126,000 | 0.54 | 0.52 | 2.5 |
| 3435 | A | 1500 | ........ | ...... | 106,000 | 67,000 | 27 | 59 | 217 | 126,000 | 0.59 | 0.57 | 2.7 |
| 3435 | CD | .... | ........ | ...... | 118,000 | 108,000 | 16 | 52 | 241 | 133,000 | 0.49 | 0.49 | 2.2 |
| 3435 | HT | 1450 | Oil | 1000 | 146,000 | 129,000 | 18 | 60 | 295 | 162,000 | 0.40 | 0.49 | 1.8 |
| 3435 | HT | 1450 | Oil | 800 | 184,000 | 162,000 | 15 | 55 | 360 | 204,000 | 0.32 | 0.51 | 1.5 |
| 3450 | HR | .... | ........ | ...... | 125,000 | 108,000 | 23 | 59 | 255 | 137,000 | 0.49 | 0.41 | 2.2 |
| 3450 | N | .... | ........ | ...... | 170,000 | 118,000 | 11 | 30 | 241 | 186,000 | 0.33 | 0.47 | 1.5 |
| 3450 | A | .... | ........ | ...... | 104,000 | 58,000 | 18 | 25 | 219 | 126,000 | 0.59 | 0.57 | 2.7 |
| 3450 | HT | 1425 | Oil | 1000 | 160,000 | 135,000 | 17 | 55 | 326 | 172,000 | 0.36 | 0.48 | 1.7 |
| 3450 | HT | 1425 | Oil | 800 | 195,000 | 168,000 | 13 | 52 | 388 | 232,000 | 0.29 | 0.52 | 1.3 |
| 4115 | HR | .... | ........ | ...... | 89,000 | 56,000 | 31 | 67 | 167 | 96,000 | 0.81 | 0.59 | 3.7 |
| 4115 | N | 1625 | ........ | ...... | 87,000 | 59,000 | 32 | 68 | 158 | 95,000 | 0.86 | 0.63 | 4.0 |
| 4130 | HR | .... | ........ | ...... | 109,000 | 70,000 | 22 | 52 | 229 | 124,000 | 0.54 | 0.51 | 2.5 |
| 4130 | N | .... | ........ | ...... | 100,000 | 65,000 | 26 | 58 | 217 | 118,000 | 0.59 | 0.53 | 2.7 |
| 4130 | HT | 1525 | Water | 1000 | 154,000 | 134,000 | 15 | 57 | 321 | 159,000 | 0.36 | 0.45 | 1.7 |
| 4130 | HT | 1525 | Water | 800 | 170,000 | 149,000 | 13 | 51 | 363 | 173,000 | 0.32 | 0.43 | 1.5 |
| X4130 | HR | .... | ........ | ...... | 95,000 | 72,000 | 22 | 56 | 192 | 111,000 | 0.65 | 0.55 | 3.0 |
| X4130 | HT | 1500 | Water | 1060 | 141,000 | 134,000 | 18 | 63 | 293 | 118,000 | 0.67 | 0.60 | 3.1 |
| 4135 | HR | .... | ........ | ...... | 131,000 | 86,000 | 16 | 48 | 272 | 142,000 | 0.44 | 0.48 | 2.0 |
| 4135 | N | .... | ........ | ...... | 114,000 | ....... | 18 | 53 | ...... | 127,000 | 0.49 | 0.48 | 2.3 |
| 4135 | HT | 1500 | Oil | 1000 | 183,000 | 163,000 | 15 | 55 | 364 | 185,000 | 0.32 | 0.46 | 1.5 |
| 4135 | HT | 1550 | Oil | 800 | 219,000 | 198,000 | 11 | 48 | 444 | 225,000 | 0.24 | 0.42 | 1.1 |
| 4140 | HR | .... | ........ | ...... | 134,000 | 92,000 | 20 | 45 | 269 | 142,000 | 0.44 | 0.48 | 2.0 |
| 4140 | N | .... | ........ | ...... | 137,000 | 81,000 | 19 | 42 | 277 | 145,000 | 0.42 | 0.48 | 2.0 |
| 4140 | HT | 1550 | Oil | 1000 | 187,000 | 171,000 | 11 | 46 | 363 | 191,000 | 0.29 | 0.42 | 1.3 |
| 4140 | HT | 1550 | Oil | 800 | 216,000 | 198,000 | 10 | 35 | 444 | 222,000 | 0.24 | 0.42 | 1.1 |
| 4150 | HR | .... | ........ | ...... | 144,000 | 108,000 | 16 | 47 | 287 | 156,000 | 0.40 | 0.47 | 1.8 |

* A = annealed; CD = cold drawn; HR = hot rolled; HT = heat treated; N = normalized.

**Table 6.—Machinability Constants of Steel—*Continued***

| Steel, S.A.E. No. | Condition* | Heat Treatment | | | Physical Properties | | | | | Chip Pressure Constant $K_1$ | Multipliers for S.A.E. 1030 HR Values. See Tables 10 to 13 | | Relative Machinability |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Heat to, deg. F. | Quench in | Draw at, deg. F. | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elong. in 2 in., percent | Reduction of Area, percent | Brinell Hardness No. | | Speed | Hp. | |
| 4150 | N | .... | ....... | ...... | 153,000 | ....... | 12 | 28 | 300 | 158,000 | 0.36 | 0.45 | 1.7 |
| 4340 | HR | .... | ....... | ...... | 198,000 | 159,000 | 9 | 22 | 401 | 202,000 | 0.26 | 0.40 | 1.2 |
| 4340 | N | .... | ....... | ...... | 203,000 | 147,000 | 12 | 20 | 415 | 207,000 | 0.27 | 0.42 | 1.2 |
| 4340 | HT | 1500 | Oil | 1200 | 150,000 | 129,000 | 20 | 57 | 302 | 155,000 | 0.40 | 0.47 | 1.8 |
| 4340 | HT | 1500 | Oil | 1000 | 184,000 | 165,000 | 16 | 48 | 375 | 186,000 | 0.31 | 0.45 | 1.4 |
| 4340 | HT | 1500 | Oil | 800 | 212,000 | 194,000 | 12 | 41 | 429 | 217,000 | 0.26 | 0.43 | 1.2 |
| 4345 | HT | 1450 | Oil | 800 | 241,000 | 220,000 | 11 | 46 | 444 | 250,000 | 0.24 | 0.47 | 1.1 |
| 4615 | HR | .... | ....... | ...... | 82,000 | 62,000 | 27 | 66 | 183 | 109,000 | 0.74 | 0.63 | 3.5 |
| 4615 | N | .... | ....... | ...... | 83,000 | 60,000 | 30 | 65 | 170 | 110,000 | 0.74 | 0.64 | 3.5 |
| 4615 | CD | .... | ....... | ...... | 107,000 | 102,000 | 14 | 58 | 212 | 125,000 | 0.54 | 0.51 | 2.5 |
| 4615 | HT | 1575 | Oil | 1000 | 115,000 | 89,000 | 27 | 70 | 229 | 128,000 | 0.54 | 0.53 | 2.5 |
| 4615 | HT | 1575 | Oil | 800 | 129,000 | 92,000 | 23 | 66 | 251 | 138,000 | 0.47 | 0.49 | 2.2 |
| 4615 | HT | 1575 | Oil | 600 | 141,000 | 99,000 | 20 | 60 | 271 | 147,000 | 0.42 | 0.48 | 2.0 |
| 4620 | N | .... | ....... | ...... | 85,000 | 61,000 | 31 | 68 | 179 | 111,000 | 0.74 | 0.63 | 3.4 |
| 4620 | HT | 1475 | Oil | 1000 | 120,000 | 97,000 | 24 | 67 | 246 | 131,000 | 0.49 | 0.49 | 2.3 |
| 4620 | HT | 1475 | Oil | 800 | 139,000 | 107,000 | 21 | 63 | 280 | 147,000 | 0.42 | 0.48 | 2.0 |
| 4620 | HT | 1475 | Oil | 600 | 178,000 | 156,000 | 15 | 60 | 345 | 182,000 | 0.32 | 0.45 | 1.5 |
| 4640 | HR | .... | ....... | ...... | 119,000 | 93,000 | 13 | 32 | 241 | 130,000 | 0.46 | 0.46 | 2.1 |
| 4640 | N | .... | ....... | ...... | 122,000 | 97,000 | 25 | 62 | 248 | 132,000 | 0.49 | 0.49 | 2.1 |
| 4640 | HT | 1500 | Oil | 1000 | 145,000 | 131,000 | 18 | 57 | 320 | 151,000 | 0.30 | 0.45 | 1.8 |
| 4640 | HT | 1500 | Oil | 800 | 182,000 | 169,000 | 14 | 42 | 392 | 182,000 | 0.29 | 0.42 | 1.4 |
| 4650 | HR | .... | ....... | ...... | 126,000 | 96,000 | 17 | 35 | 269 | 135,000 | 0.44 | 0.46 | 2.0 |
| 4650 | N | .... | ....... | ...... | 120,000 | 90,000 | 18 | 44 | 241 | 131,000 | 0.49 | 0.48 | 2.2 |
| 4650 | HT | 1475 | Oil | 1100 | 139,000 | 125,000 | 20 | 53 | 305 | 147,000 | 0.41 | 0.46 | 1.9 |
| 4650 | HT | 1475 | Oil | 900 | 175,000 | 160,000 | 15 | 50 | 382 | 178,000 | 0.31 | 0.42 | 1.4 |
| 4815 | N | .... | ....... | ...... | 94,000 | 68,000 | 28 | 62 | 187 | 118,000 | 0.67 | 0.60 | 3.1 |
| 4820 | HT | 1550 | Oil | 600 | 199,000 | 180,000 | 15 | 61 | 390 | 203,000 | 0.29 | 0.45 | 1.3 |
| 5120 | HR | .... | ....... | ...... | 82,000 | 48,000 | 28 | 61 | ...... | 110,000 | 0.78 | 0.66 | 3.6 |
| 5120 | N | .... | ....... | ...... | 73,000 | 50,000 | 33 | 65 | 149 | 105,000 | 0.86 | 0.69 | 4.0 |
| 5120 | CD | .... | ....... | ...... | 79,000 | ....... | 25 | ...... | 165 | 108,000 | 0.78 | 0.65 | 3.6 |
| 5120 | HT | 1560 | Water | 842 | 155,000 | 122,000 | 12 | 46 | 335 | 160,000 | 0.35 | 0.42 | 1.6 |
| 5120 | HT | 1600 | Water | 930 | 98,000 | 70,000 | 21 | 67 | 210 | 122,000 | 0.59 | 0.55 | 2.7 |
| 5130 | HR | .... | ....... | ...... | 105,000 | 66,000 | 24 | 63 | 207 | 125,000 | 0.60 | 0.58 | 2.8 |
| 5130 | N | .... | ....... | ...... | 104,000 | 67,000 | 24 | 63 | 201 | 125,000 | 0.60 | 0.58 | 2.8 |
| 5130 | HT | 1550 | Oil | 1200 | 120,000 | 99,000 | 22 | 67 | 235 | 136,000 | 0.53 | 0.55 | 2.5 |
| 5130 | HT | 1550 | Oil | 1000 | 148,000 | 130,000 | 18 | 60 | 302 | 154,000 | 0.40 | 0.47 | 1.8 |
| 5130 | HT | 1550 | Oil | 800 | 193,000 | 168,000 | 12 | 49 | 375 | 196,000 | 0.29 | 0.43 | 1.3 |
| 5135 | A | .... | ....... | ...... | 90,000 | 58,000 | 25 | 37 | 207 | 116,000 | 0.69 | 0.61 | 3.2 |
| 5140 | HR | .... | ....... | ...... | 114,000 | 75,000 | 20 | 58 | 223 | 131,000 | 0.53 | 0.54 | 2.5 |
| 5140 | N | .... | ....... | ...... | 110,000 | 70,000 | 22 | 61 | 212 | 128,000 | 0.55 | 0.55 | 2.6 |
| 5140 | HT | 1550 | Oil | 1200 | 124,000 | 100,000 | 22 | 63 | 241 | 138,000 | 0.49 | 0.52 | 2.3 |
| 5140 | HT | 1550 | Oil | 1000 | 152,000 | 130,000 | 18 | 55 | 302 | 157,000 | 0.38 | 0.47 | 1.8 |
| 5140 | HT | 1550 | Oil | 800 | 180,000 | 160,000 | 13 | 42 | 363 | 182,000 | 0.32 | 0.45 | 1.5 |
| 5150 | HR | .... | ....... | ...... | 135,000 | 101,000 | 15 | 50 | 277 | 145,000 | 0.42 | 0.48 | 2.0 |
| 5150 | N | .... | ....... | ...... | 135,000 | 98,000 | 15 | 50 | 285 | 145,000 | 0.41 | 0.46 | 1.9 |
| 5150 | HT | 1550 | Oil | 1200 | 133,000 | 114,000 | 21 | 57 | 269 | 144,000 | 0.46 | 0.51 | 2.1 |
| 5150 | HT | 1550 | Oil | 1000 | 168,000 | 150,000 | 17 | 45 | 352 | 171,000 | 0.35 | 0.45 | 1.6 |
| 5150 | HT | 1550 | Oil | 800 | 220,000 | 202,000 | 9 | 30 | 461 | 226,000 | 0.23 | 0.41 | 1.1 |
| 5210 | A | .... | ....... | ...... | 95,000 | 58,000 | 30 | 61 | 187 | 125,000 | 0.67 | 0.64 | 3.1 |
| 6115 | HR | .... | ....... | ...... | 83,000 | 65,000 | 30 | 63 | 174 | 123,000 | 0.81 | 0.76 | 3.7 |
| 6115 | N | .... | ....... | ...... | 81,000 | 64,000 | 30 | 63 | 170 | 122,000 | 0.82 | 0.72 | 3.8 |
| 6120 | HR | .... | ....... | ...... | 114,000 | 75,000 | 23 | 64 | 255 | 131,000 | 0.50 | 0.51 | 2.3 |
| 6120 | N | 1600 | ....... | ...... | 80,000 | 46,000 | 33 | 73 | 164 | 109,000 | 0.81 | 0.69 | 3.8 |
| 6120 | HT | 1625 | Water | 1150 | 112,000 | 95,000 | 24 | 72 | 220 | 130,000 | 0.55 | 0.55 | 2.6 |
| 6120 | HT | 1625 | Water | 800 | 150,000 | 130,000 | 17 | 63 | 310 | 155,000 | 0.38 | 0.46 | 1.8 |
| 6125 | HR | .... | ....... | ...... | 114,000 | 75,000 | 23 | 64 | ...... | 130,000 | 0.53 | 0.53 | 2.5 |
| 6125 | HT | 1625 | Water | 1200 | 127,000 | 113,000 | 23 | 66 | 260 | 140,000 | 0.49 | 0.53 | 2.3 |
| 6125 | HT | 1625 | Water | 1000 | 134,000 | 120,000 | 19 | 63 | 282 | 156,000 | 0.45 | 0.54 | 2.1 |
| 6125 | HT | 1625 | Water | 800 | 150,000 | 127,000 | 17 | 63 | 310 | 217,000 | 0.40 | 0.66 | 1.8 |
| 6130 | HR | .... | ....... | ...... | 112,000 | 69,000 | 21 | 61 | ...... | 130,000 | 0.53 | 0.53 | 2.5 |
| 6130 | N | 1625 | ....... | ...... | 103,000 | 72,000 | 26 | 65 | 207 | 126,000 | 0.64 | 0.63 | 3.0 |
| 6130 | HT | 1680 | Oil | 1230 | 94,000 | 64,000 | 24 | 65 | 195 | 118,000 | 0.72 | 0.65 | 3.3 |
| 6130 | HT | 1625 | Oil | 1100 | 141,000 | 126,000 | 19 | 62 | 305 | 177,000 | 0.42 | 0.58 | 2.0 |

* A = annealed; CD = cold drawn; HR = hot rolled; HT = heat treated; N = normalized.

### Table 6.—Machinability Constants of Steel—*Continued*

| Steel, S.A.E. No. | Condition* | Heat Treatment | | | Physical Properties | | | | | Chip Pressure Constant $K_1$ | Multipliers for S.A.E. 1030 HR Values. See Tables 10 to 13 | | Relative Machinability |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Heat to, deg. F. | Quench in | Draw at, deg. F. | Tensile Strength, lb. per sq. in. | Yield Point, lb. per sq. in. | Elong. in 2 in., percent | Reduction of Area, percent | Brinell Hardness No. | | Speed | Hp. | |
| 6130 | HT | 1720 | Oil | 900 | 127,000 | 105,000 | 18 | 54 | 264 | 140,000 | 0.47 | 0.51 | 2.2 |
| 6135 | A | 1600 | ....... | ...... | 108,000 | 76,000 | 23 | 64 | 217 | 127,000 | 0.60 | 0.59 | 2.8 |
| 6135 | HT | 1600 | Oil | 1200 | 133,000 | 120,000 | 23 | 64 | 275 | 156,000 | 0.47 | 0.51 | 2.2 |
| 6135 | HT | 1600 | Oil | 1000 | 150,000 | 134,000 | 18 | 61 | 308 | 217,000 | 0.41 | 0.67 | 1.9 |
| 6135 | HT | 1600 | Oil | 800 | 169,000 | 147,000 | 17 | 60 | 348 | 260,000 | 0.36 | 0.71 | 1.6 |
| 6140 | HR | .... | ........ | ...... | 133,000 | 112,000 | 19 | 44 | 277 | 156,000 | 0.46 | 0.57 | 2.1 |
| 6140 | N | .... | ........ | ...... | 126,000 | 99,000 | 22 | 47 | 262 | 139,000 | 0.49 | 0.53 | 2.3 |
| 6140 | HT | 1575 | Oil | 1200 | 120,000 | 103,000 | 22 | 62 | 241 | 130,000 | 0.53 | 0.53 | 2.4 |
| 6140 | HT | 1575 | Oil | 1000 | 149,000 | 134,000 | 18 | 55 | 311 | 213,000 | 0.41 | 0.66 | 1.9 |
| 6140 | HT | 1575 | Oil | 800 | 190,000 | 173,000 | 13 | 52 | 401 | 260,000 | 0.29 | 0.60 | 1.4 |
| 6145 | HR | .... | ........ | ...... | 140,000 | 81,000 | 10 | 26 | 302 | 173,000 | 0.38 | 0.52 | 1.8 |
| 6145 | HT | 1525 | Oil | 900 | 183,000 | 166,000 | 15 | 50 | 401 | 280,000 | 0.32 | 0.69 | 1.5 |
| 6145 | HT | 1565 | Oil | 800 | 231,000 | 206,000 | 11 | 44 | 444 | 370,000 | 0.26 | 0.74 | 1.2 |
| 6150 | HR | .... | ........ | ...... | 141,000 | 107,000 | 18 | 44 | 285 | 177,000 | 0.44 | 0.60 | 2.0 |
| 6150 | N | .... | ........ | ...... | 136,000 | 100,000 | 20 | 47 | 277 | 159,000 | 0.46 | 0.57 | 2.1 |
| 6150 | HT | 1575 | Oil | 1200 | 133,000 | 110,000 | 19 | 55 | 262 | 158,000 | 0.47 | 0.58 | 2.2 |
| 6150 | HT | 1575 | Oil | 1000 | 171,000 | 150,000 | 12 | 45 | 341 | 265,000 | 0.35 | 0.70 | 1.6 |
| 6150 | HT | 1575 | Oil | 800 | 228,000 | 210,000 | 10 | 38 | 461 | 300,000 | 0.24 | 0.55 | 1.1 |
| 30905 | HT | 1950 | Water | ...... | 91,000 | 55,000 | 71 | 80 | 187 | 126,000 | 0.36 | 0.35 | 1.6 |
| 30905 | A | .... | ........ | ...... | 75,000 | 33,000 | 57 | 63 | 142 | 122,000 | 0.45 | 0.42 | 2.1 |
| 51210 | HT | 1290 | Air cool | | 93,000 | 75,000 | 26 | 74 | 198 | 118,000 | 0.67 | 0.60 | 3.1 |

* A = annealed; CD = cold drawn; HR = hot rolled; HT = heat treated; N = normalized.

### Table 7.—Machinability Constants of Cast Iron

| Chemical Composition | | | | Physical Properties | | | Chip Pressure Constant $K_2$ | Multipliers for Cast Iron of 177 Brinell. See Tables 14 to 17 | | Relative Machinability |
|---|---|---|---|---|---|---|---|---|---|---|
| Total Carbon | Combined Carbon | Silicon | Manganese | Tensile Strength, lb. per sq. in. | Compressive Strength, lb. per sq. in. | Brinell Hardness No. | | Speed | Hp. | |
| 2.80 | 0.90 | 2.85 | 0.41 | 50,000 | 159,000 | 224 | 111,000 | 0.59 | 1.11 | 2.4 |
| 2.90 | ....... | 1.12 | 0.71 | 44,000 | 152,000 | 218 | 104,000 | 0.66 | 1.16 | 2.7 |
| 3.30 | 1.05 | 0.88 | 0.94 | 36,000 | 111,000 | 199 | 66,000 | 0.70 | 0.78 | 2.8 |
| 3.04 | ....... | 1.70 | 0.69 | 40,000 | 129,000 | 198 | 82,000 | 0.71 | 0.98 | 2.9 |
| 3.16 | ....... | 2.50 | 0.79 | 37,000 | 118,000 | 194 | 74,000 | 0.71 | 0.88 | 2.9 |
| 2.65 | ....... | 2.34 | 1.05 | 38,000 | 130,000 | 205 | 84,000 | 0.71 | 1.01 | 2.9 |
| 2.70 | 0.65 | 2.00 | 0.60 | 38,000 | 133,000 | 204 | 87,000 | 0.76 | 1.12 | 3.1 |
| 2.92 | 0.88 | 1.57 | 0.36 | 41,000 | 139,000 | 195 | 92,000 | 0.80 | 1.24 | 3.3 |
| 3.13 | ....... | 1.51 | 1.06 | 33,000 | 111,000 | 183 | 66,000 | 0.91 | 1.02 | 3.7 |
| 2.88 | 0.32 | 1.47 | 0.60 | 30,000 | 102,000 | 177 | 59,000 | 1.00 | 1.00 | 4.1 |
| 3.03 | ....... | 1.68 | 0.98 | 33,000 | 120,000 | 181 | 74,000 | 1.03 | 1.28 | 4.2 |
| 3.10 | ....... | 1.89 | 0.80 | 36,000 | 116,000 | 182 | 71,000 | 1.04 | 1.25 | 4.2 |
| 3.20 | ....... | 2.01 | 0.79 | 33,000 | 122,000 | 182 | 76,000 | 1.05 | 1.35 | 4.3 |
| 3.39 | 0.80 | 1.60 | 0.52 | 30,000 | 102,000 | 161 | 59,000 | 1.10 | 1.10 | 4.5 |
| 2.98 | 0.24 | 2.41 | 0.14 | ......... | .......... | 144 | 36,000 | 1.20 | 0.74 | 4.9 |
| 3.37 | 0.18 | 2.34 | 0.50 | 18,500 | 81,000 | 156 | 42,000 | 1.32 | 0.97 | 5.4 |
| 3.51* | 0.10 | 2.31 | 0.69 | 33,000 | 121,000 | 168 | 76,000 | 1.48 | 0.91 | 6.1* |
| 3.71 | ....... | 1.80 | 0.91 | 17,000 | 66,000 | 127 | 31,000 | 1.67 | 0.88 | 6.8 |
| 2.46† | ....... | 1.00 | 0.26 | ......... | .......... | 137 | 70,000 | 1.10 | 1.31 | 4.5† |

* Centrifugally cast. † Malleable iron.

### Table 8.—Cutting Speeds for Wrought Monel Metal

| Feed, in. per Revolution | Depth of Cut, in. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 1/32 | 1/16 | 1/8 | 1/4 | 3/8 | 1/2 | 5/8 | 3/4 | 1 |
| | Cutting Speed, ft. per min. | | | | | | | | |
| 0.002 | 262 | 223 | 205 | 192 | 190 | 186 | 185 | 183 | 182 |
| .004 | 209 | 179 | 163 | 153 | 149 | 147 | 147 | 145 | 144 |
| .008 | 168 | 144 | 138 | 124 | 121 | 118 | 116 | 114 | 113 |
| 1/64 | 139 | 121 | 110 | 103 | 98 | 95 | 93 | 92 | 90 |
| 1/32 | 118 | 104 | 68 | 57 | 52 | 49 | 47 | 45 | 43 |
| 1/16 | 104 | 60 | 48 | 39 | 34 | 31 | 29 | 26 | 25 |
| 1/8 | 69 | 48 | 34 | 29 | 24 | 21 | 18 | 17 | 14 |

**Table 9.—Relative Machinability of Non-ferrous Metals**

Based on cutting speed

| Material | Condition | Chemical Composition | | | | | Relative Machinability |
|---|---|---|---|---|---|---|---|
| | | Copper | Tin | Lead | Zinc | Nickel | |
| Muntz metal | Wrought | | | | | | 15.2 |
| Muntz metal | Cast | | | | | | 11.8 |
| Soft brass | Wrought | | | | | | 13.6 |
| Soft brass | Cast | | | | | | 10.5 |
| Phosphor bronze | Wrought | | | | | | 11.5 |
| Phosphor bronze | Cast | | | | | | 7.1 |
| Navy rolled brass | Wrought | | | | | | 11.5 |
| Navy rolled brass | Cast | | | | | | 7.1 |
| Navy "G" | Wrought | 88.0 | 8.0 | | 4.0 | | 11.5 |
| Navy "G" | Cast | 88.0 | 8.0 | | 4.0 | | 7.1 |
| Hard brass | Wrought | | | | | | 12.1 |
| Hard brass | Cast | | | | | | 9.4 |
| Bronze | Wrought | 90.0 | 10.0 | | | | 12.1 |
| Bronze | Cast | 90.0 | 10.0 | | | | 9.4 |
| Bronze | Wrought | 85.0 | 6–11 | 0–1 | | | 9.2 |
| Bronze | Cast | 85.0 | 6–11 | 0–1 | | | 7.1 |
| Bronze | Wrought | 84.0 | 13.0 | | | 3.0 | 9.2 |
| Bronze | Cast | 84.0 | 13.0 | | | 3.0 | 7.1 |
| Marine bronze | Wrought | | | | | | 11.5 |
| Marine bronze | Cast | | | | | | 7.1 |
| Tobin bronze | Wrought | | | | | | 11.5 |
| Tobin bronze | Cast | | | | | | 7.1 |
| Bronze | Wrought | 88.0 | 11.0 | 1.0 | | | 11.5 |
| Bronze | Cast | 88.0 | 11.0 | 1.0 | | | 7.1 |
| Monel metal | Forged | | | | | | 4.1 |
| Monel metal | Cast | | | | | | 3.5 |
| Government babbitt | Cast | | | | | | 13.9 |
| Aluminum | Wrought | | | | | | 40.0 |
| Aluminum | Wrought | | | | | | 34.0 |
| Copper | Cast | | | | | | 9.5 |
| Hard rubber | | | | | | | 12.9 |
| Bakelite | | | | | | | 5.0 |

## 4. CUTTING FLUIDS

The various fluids used in metal cutting operations serve the following purposes: 1. Cooling the tool; 2. Cooling the work; 3. Improving the quality of the surface produced; 4. Lubricating the surfaces in contact between the tool and work, or preventing welding of chip or work to the tool, thus aiding in reducing the temperature of the tool; 5. Flushing away the chips; 6. Reducing the pressure of the chip on the tool.

For cooling tool and work and flushing away chips, the quantity of fluid is more important than the kind. A thin soluble oil solution, approximately 40–1, or a solution of soda or borax in water is very satisfactory. A pure oil or a rich mixture of soluble oil does not appear to be so efficient a coolant as a fluid which is mainly water.

An oil containing a large percentage of sulphur appears to reduce the tendency of a steel chip to weld to the tool. Such oil is advantageous when a good quality of machined finish is desired. For finishing operations an oil with a high degree of oiliness generally is used.

**QUALITY OF SURFACE.**—There are few data available as to the effect of cutting fluids on the quality of the surface produced. Where high quality of finished steel surface is required, a sulphurized mineral, or a sulphurized mineral lard oil generally is used, except that water or soluble oil generally is used to obtain a high grade of finish with a spring tool on a lathe.

**EFFECT OF QUANTITY OF CUTTING FLUID.**—When a cutting fluid is used it is particularly important that it be applied steadily; intermittent application may tend to crack the tool. When machining surfaces which are not easily accessible, special means should be provided, for supplying the cutting fluid at the cutting edge of the tool; for example, oil holes through the boring bar used for boring deep holes.

Table 19, based on the work of George Schlesinger as reported in *Stahl u. Eisen*, Vol. 48, p. 341, gives an indication of the effect of variation in the quantity of fluid used on the rate of metal removal. Frederick W. Taylor had recognized the importance of an adequate quantity of cutting fluid and recommended the use of at least 3 gal. per minute on

(*Continued on* p. 21-22)

### Table 10.—Cutting Speed and Horsepower Required for Cutting Steel with Carbon Steel Round-nose Tools. Tool life, 1 hr.

Steel: S.A.E. 1030, hot rolled. Tool: 1.15 Carbon Steel; nose radius, 1/8 in.; side-cutting-edge angle, 0; front-cutting-edge angle, 6 deg.; side rake, 14 deg.; back rake, 8 deg.; clearance, 6 deg. See Fig. 11F

| Feed, in. per rev. | Depth of Cut, in. 1/32 | | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 5/8 | | 3/4 | | 1 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 103 | 0.1 | 71 | 0.1 | 62 | 0.2 | 56 | 0.4 | 54 | 0.6 | 52 | 0.8 | 52 | 1.0 | 51 | 1.2 | 51 | 1.7 |
| .004 | 66 | .1 | 47 | .2 | 40 | .3 | 36 | .5 | 34 | .7 | 33 | .9 | 33 | 1.1 | 32 | 1.3 | 31 | 1.8 |
| .008 | 42 | .1 | 32 | .2 | 26 | .3 | 23 | .5 | 22 | .7 | 21 | 1.0 | 21 | 1.2 | 20 | 1.4 | 20 | 1.9 |
| 1/64 | 28 | .1 | 21 | .2 | 18 | .3 | 15 | .6 | 15 | .8 | 14 | 1.1 | 13 | 1.3 | 13 | 1.5 | 13 | 2.1 |
| 1/32 | 20 | .2 | 15 | .2 | 12 | .4 | 11 | .7 | 10 | 1.0 | 9 | 1.2 | 8 | 1.5 | 8 | 1.7 | 8 | 2.2 |
| 1/16 | 15 | .2 | 12 | .3 | 10 | .5 | 8 | .9 | 7 | 1.2 | 6 | 1.4 | 6 | 1.6 | 5 | 1.9 | 5 | 2.3 |
| 1/8 | 13 | .4 | 10 | .5 | 8 | .8 | 6 | 1.2 | 5 | 1.6 | 4 | 1.8 | 4 | 1.9 | 3 | 2.1 | 3 | 2.3 |

### Table 11—Cutting Speed and Horsepower Required for Cutting Steel with High-speed Steel Side-cutting Tools. Tool life, 1 hr.

Steel: S.A.E. 1030, hot rolled. Tool: 18–4–1 high-speed steel; nose radius, 0; side-cutting-edge angle, 0; side rake, 14 deg.; back rake, 8 deg. See Fig. 11A

| Feed, in. per rev. | Depth of Cut, in. 1/32 | | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 5/8 | | 3/4 | | 1 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 328 | 0.2 | 327 | 0.5 | 327 | 1.1 | 325 | 2.2 | 325 | 3.4 | 323 | 4.7 | 323 | 5.9 | 323 | 7.2 | 323 | 9.9 |
| .004 | 209 | .3 | 212 | .6 | 206 | 1.1 | 206 | 2.4 | 205 | 3.7 | 205 | 5.1 | 203 | 6.5 | 203 | 7.9 | 203 | 10.8 |
| .008 | 136 | .4 | 135 | .7 | 134 | 1.3 | 130 | 2.7 | 130 | 4.1 | 130 | 5.7 | 129 | 7.1 | 129 | 8.6 | 128 | 11.7 |
| 1/64 | 92 | .5 | 91 | .8 | 87 | 1.5 | 86 | 3.0 | 85 | 4.5 | 84 | 6.1 | 83 | 7.7 | 83 | 9.3 | 82 | 12.5 |
| 1/32 | 64 | .8 | 61 | 1.2 | 59 | 2.0 | 56 | 3.7 | 55 | 5.4 | 54 | 7.1 | 53 | 8.7 | 52 | 10.4 | 51 | 13.7 |
| 1/16 | 65 | 1.2 | 44 | 1.9 | 41 | 2.9 | 39 | 4.8 | 36 | 6.6 | 35 | 8.5 | 34 | 10.2 | 33 | 11.9 | 32 | 15.4 |
| 1/8 | 66 | 1.9 | 42 | 2.9 | 31 | 5.0 | 27 | 7.0 | 24 | 8.8 | 23 | 10.7 | 22 | 12.5 | 21 | 14.1 | 20 | 17.5 |

### Table 12.—Cutting Speed and Horsepower Required for Cutting Steel with High-speed Steel Round-nose Cutting Tools. Tool life, 1 hr.

Steel: S.A.E. 1030, hot rolled. Tool: 18–4–1 high-speed steel; nose radius, 1/4 in.; side-cutting-edge angle, 30 deg.; side rake, 14 deg.; back rake, 8 deg. See Fig. 11F

| Feed, in. per rev. | Depth of Cut, in. 1/32 | | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 5/8 | | 3/4 | | 1 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 737 | 0.7 | 515 | 1.0 | 447 | 1.6 | 408 | 3.0 | 390 | 4.5 | 382 | 5.9 | 379 | 7.4 | 371 | 8.9 | 368 | 12.1 |
| .004 | 475 | .8 | 342 | 1.1 | 285 | 1.8 | 254 | 3.3 | 245 | 4.8 | 241 | 6.7 | 236 | 8.0 | 234 | 9.7 | 231 | 13.0 |
| .008 | 302 | .9 | 220 | 1.2 | 197 | 2.1 | 164 | 3.6 | 157 | 5.3 | 151 | 7.5 | 149 | 8.9 | 147 | 10.4 | 146 | 14.1 |
| 1/64 | 199 | 1.0 | 144 | 1.3 | 120 | 2.2 | 106 | 3.9 | 102 | 5.7 | 98 | 7.6 | 96 | 9.4 | 95 | 11.3 | 93 | 15.1 |
| 1/32 | 134 | 1.2 | 95 | 1.6 | 80 | 2.5 | 70 | 4.5 | 65 | 6.4 | 63 | 8.4 | 61 | 10.1 | 60 | 12.2 | 59 | 16.3 |
| 1/16 | 94 | 1.5 | 68 | 2.0 | 55 | 3.1 | 47 | 5.3 | 43 | 7.3 | 41 | 9.4 | 40 | 11.6 | 39 | 13.6 | 36 | 17.6 |
| 1/8 | 81 | 2.2 | 55 | 2.9 | 42 | 4.2 | 34 | 6.8 | 30 | 9.0 | 28 | 11.1 | 27 | 13.5 | 27 | 16.3 | 23 | 19.6 |

### Table 13.—Cutting Speed and Horsepower Required for Cutting Steel with High-speed Steel Parting Tools. Tool life, 1 hr.

Steel: S.A.E. 1030 hot rolled. Tool: 18–4–1 high-speed steel; corner radius, 1/64 in.; setting angle, 90 deg.; back rake, 20 deg.; relief, 6 deg. See Fig. 10

| Feed, in. per rev. | Width of Cut, in. 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 3/4 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 383 | 0.7 | 378 | 1.3 | 378 | 2.6 | 378 | 3.9 | 378 | 5.3 | 378 | 8.1 |
| .004 | 240 | .8 | 240 | 1.5 | 239 | 2.8 | 239 | 4.3 | 236 | 5.8 | 236 | 8.8 |
| .006 | 187 | 1.0 | 187 | 1.6 | 186 | 3.1 | 183 | 4.6 | 183 | 6.2 | 183 | 9.4 |
| .008 | 156 | 1.0 | 154 | 1.7 | 154 | 3.3 | 151 | 4.8 | 151 | 6.5 | 150 | 9.7 |
| .010 | 136 | 1.1 | 134 | 1.9 | 133 | 3.4 | 133 | 5.1 | 130 | 6.7 | 130 | 10.1 |
| .012 | 122 | 1.2 | 120 | 1.9 | 118 | 3.5 | 116 | 5.1 | 116 | 6.9 | 115 | 10.3 |

**Table 14.—Cutting Speed and Horsepower Required for Cutting Cast Iron with Carbon Steel Round-nose Cutting Tools. Tool life, 1 hr.**

Cast Iron: Brinell Hardness, 177; total carbon, 2.88%; combined carbon, 0.32%; silicon, 1.47%; manganese, 0.60%. Tool: 1.15 carbon steel; nose radius 1/8 in.; side-cutting-edge angle, 0.; front-cutting-edge angle, 6 deg.; side rake, 14 deg.; back rake, 8 deg.; clearance, 6 deg. See Fig. 11F

| Feed, in. per rev. | Depth of Cut, in. | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/32 | | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 5/8 | | 3/4 | | 1 | |
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 95 | 0.1 | 77 | 0.2 | 69 | 0.3 | 64 | 0.5 | 63 | 0.7 | 62 | 0.9 | 61 | 1.1 | 60 | 1.3 | 60 | 1.7 |
| .004 | 71 | .1 | 58 | .2 | 52 | .3 | 48 | .6 | 47 | .8 | 46 | .9 | 45 | 1.3 | 45 | 1.5 | 44 | 1.9 |
| .008 | 54 | .2 | 45 | .2 | 42 | .4 | 37 | .6 | 35 | .9 | 34 | 1.2 | 34 | 1.5 | 33 | 1.7 | 33 | 2.3 |
| 1/64 | 49 | .2 | 36 | .3 | 32 | .5 | 29 | .8 | 27 | 1.1 | 26 | 1.4 | 25 | 1.7 | 25 | 2.1 | 24 | 2.6 |
| 1/32 | 33 | .3 | 29 | .4 | 26 | .6 | 24 | 1.1 | 22 | 1.4 | 20 | 1.7 | 20 | 2.1 | 19 | 2.4 | 18 | 3.0 |
| 1/16 | 29 | .4 | 26 | .6 | 23 | .9 | 20 | 1.4 | 18 | 1.9 | 16 | 2.2 | 15 | 2.5 | 14 | 2.8 | 13 | 3.4 |
| 1/8 | 27 | .6 | 24 | .9 | 22 | 1.4 | 18 | 2.1 | 15 | 2.6 | 13 | 2.9 | 12 | 2.1 | 11 | 3 4 | 9 | 3 7 |

**Table 15.—Cutting Speed and Horsepower Required for Cutting Cast Iron with High-speed Steel Side-cutting Tools. Tool life, 1 hr.**

Cast Iron: Brinell Hardness, 177; total carbon, 2.88%; combined carbon, 0.32%; silicon, 1.47%; manganese, 0.60%. Tool: 18–4–1 high-speed steel; nose radius, 0; side-cutting-edge angle, 0; side rake, 14 deg.; back rake, 8 deg. See Fig. 11A

| Feed, in. per rev. | Depth of Cut, in. | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/32 | | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 5/8 | | 3/4 | | 1 | |
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 169 | 0.2 | 168 | 0.3 | 167 | 0.6 | 166 | 1.1 | 166 | 1.7 | 166 | 2.3 | 166 | 2.8 | 166 | 3.4 | 166 | 4.5 |
| .004 | 127 | .2 | 127 | .4 | 126 | .7 | 126 | 1.4 | 125 | 2.0 | 124 | 2.7 | 124 | 3.3 | 124 | 4.0 | 123 | 5.3 |
| .008 | 97 | .3 | 97 | .4 | 96 | .9 | 94 | 1.6 | 93 | 2.4 | 93 | 3.2 | 93 | 4.0 | 92 | 4.7 | 92 | 6.2 |
| 1/64 | 77 | .4 | 76 | .6 | 74 | 1.1 | 73 | 2.0 | 71 | 2.9 | 71 | 3.8 | 70 | 4.7 | 69 | 5.5 | 68 | 7.2 |
| 1/32 | 63 | .6 | 61 | .9 | 58 | 1.5 | 56 | 2.5 | 54 | 3.6 | 53 | 4.6 | 52 | 5.6 | 52 | 6.6 | 51 | 9.0 |
| 1/16 | 62 | .9 | 52 | 1.6 | 48 | 2.3 | 45 | 3.6 | 43 | 4.7 | 41 | 5.9 | 40 | 7.0 | 39 | 8.1 | 37 | 11.3 |
| 1/8 | 62 | 1.4 | 49 | 1.7 | 41 | 4.0 | 37 | 4.6 | 34 | 5.[illegible] | 32 | 8.1 | 31 | 9.3 | 29 | 10.2 | 28 | 12.4 |

**Table 16.—Cutting Speed and Horsepower Required for Cutting Cast Iron with High-speed Steel Round-nose Cutting Tools. Tool life, 1 hr.**

Cast Iron: Brinell Hardness, 177; total carbon, 2.88%; combined carbon, 0.32%; silicon, 1.47%; manganese, 0.60%. Tool: 18–4–1 high-speed steel; nose radius, 1/8 in.; side-cutting-edge angle, 30 deg.; side rake, 14 deg.; back rake, 8 deg. See Fig. 11E

| Feed, in. per rev. | Depth of Cut, in. | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/32 | | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 5/8 | | 3/4 | | 1 | |
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.002 | 284 | 0.4 | 230 | 0.5 | 207 | 0.8 | 192 | 1.5 | 187 | 2.1 | 185 | 2.7 | 182 | 3.4 | 181 | 4.0 | 180 | 5.3 |
| .004 | 212 | .4 | 172 | .6 | 154 | 1.0 | 144 | 1.7 | 140 | 2.5 | 137 | 3.2 | 136 | 4.0 | 135 | 4.7 | 134 | 6.2 |
| .008 | 160 | .5 | 130 | .7 | 122 | 1.2 | 107 | 2.0 | 104 | 2.9 | 103 | 3.8 | 101 | 4.6 | 100 | 5.5 | 99 | 7.1 |
| 1/64 | 122 | .6 | 100 | .9 | 89 | 1.4 | 82 | 2.4 | 79 | 3.4 | 77 | 4.4 | 77 | 5.4 | 76 | 6.3 | 74 | 8.2 |
| 1/32 | 96 | .8 | 78 | 1.1 | 69 | 1.7 | 63 | 2.9 | 60 | 4.1 | 58 | 5.2 | 58 | 6.4 | 56 | 7.5 | 55 | 9.6 |
| 1/16 | 77 | 1.0 | 63 | 1.4 | 56 | 2.2 | 50 | 3.7 | 47 | 5.0 | 45 | 6.4 | 43 | 7.6 | 42 | 8.9 | 41 | 11.2 |
| 1/8 | 69 | 1.5 | 56 | 2.1 | 47 | 3.1 | 41 | 5.0 | 3[illegible] | 6.5 | 35 | 7.9 | 34 | 9.3 | 33 | 11.1 | 31 | 13.1 |

**Table 17.—Cutting Speed and Horsepower Required for Cutting Cast Iron with High-speed Steel Parting Tools. Tool life, 1 hr.**

Cast Iron: Brinell Hardness, 177; total carbon, 2.88%; combined carbon, 0.32%; silicon, 1.47%; manganese, 0.62%. Tool: 18–4–1 high-speed steel; corner radius, 1/64 in.; setting angle, 90 deg.; back rake, 20 deg.; relief, 6 deg. See Fig. 10

| Feed, in. per rev. | Width of Cut, in. | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/16 | | 1/8 | | 1/4 | | 3/8 | | 1/2 | | 3/4 | |
| | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. | Ft. per min. | Hp. |
| 0.003 | 94 | 0.3 | 94 | 0.6 | 94 | 1.1 | 94 | 1.5 | 94 | 2.0 | 93 | 2.9 |
| .005 | 76 | .4 | 76 | .7 | 75 | 1.2 | 75 | 1.7 | 74 | 2.2 | 74 | 3.3 |
| .007 | 67 | .5 | 67 | .7 | 66 | 1.3 | 66 | 1.9 | 65 | 2.5 | 65 | 3.6 |
| .010 | 58 | .5 | 58 | .8 | 57 | 1.5 | 56 | 2.1 | 56 | 2.7 | 56 | 4.0 |
| .015 | 50 | .6 | 49 | 1.0 | 49 | 1.7 | 48 | 2.4 | 47 | 3.1 | 47 | 4.4 |
| .020 | 45 | .8 | 44 | 1.1 | 43 | 1.9 | 43 | 2.6 | 42 | 3.3 | 41 | 4.7 |

roughing cuts. It will be seen from Table 18 that the use of 3 gal. per min. of water results in a condition close to the maximum for the experiments on which this table is based.

For light cuts on steel, where the chip thickness is less than 0.015 in., the use of a copious supply of cutting fluid permits the use of cutting speeds approximately 16% greater than could otherwise be used. The effect of the cutting fluid appears to vary with the thickness of the chip from 16% for thin chips up to approximately 50% for chips approximately 0.090 in. thick.

**GENERAL RECOMMENDATIONS.**—Table 19 is based on experiment, and on the recommendations of the suppliers of cutting fluid. Where possible, it is advisable to obtain the recommendation of a reliable manufacturer concerning the fluid to be used for a particular application.

**Table 18.—Effect of Quantity of Cooling Water on Permissible Cutting Speed**

Rough turning chrome-nickel steel with high-speed steel tools; 0.079 in. depth of cut; 0.039 in. feed. Tool with 45-deg. side-cutting-edge angle, 30-deg. front-cutting-edge angle, sharp nose, 14-deg. rake

| | | | | | | |
|---|---|---|---|---|---|---|
| Water per min., gallons | 0 | 0.2 | 1.1 | 2.2 | 3.3 | 4.4 |
| Cutting speed, ft. per min.* | 67 | 71 | 81 | 84 | 85 | 85 |
| Relative rate of metal removal | 1.00† | 1.05 | 1.21 | 1.25 | 1.26 | 1.27 |

* For 20 min. tool life. † The dry condition is taken as unity.

**Table 19.—Recommended Practice for the Use of Cutting Fluid for Metals**

| Material Cut | Rough Turning and Planing | Finishing, Forming and Turning and Planing |
|---|---|---|
| Cast iron | Dry<br>Air blast<br>Soluble oil, 50 to 1<br>Water, with 1 1/2 % borax or soda | Dry<br>Air blast<br>Soluble oil, 50 to 1<br>Water, with 1 1/2 % borax or soda |
| Malleable iron | Dry<br>Air blast<br>Soluble oil, 50 to 1<br>Water, with 1 1/2 % borax or soda | Dry<br>Air blast<br>Soluble oil, 50 to 1<br>Water, with 1 1/2 % borax or soda |
| Wrought iron, or low-carbon steel | Soluble oil, 40 to 1<br>Sulphurized mineral oil<br>Water, with 1 1/2 % borax or soda | Soluble oil, 40 to 1<br>Sulphurized mineral oil<br>Water, with 1 1/2 % borax or soda |
| Medium-carbon steel | Soluble oil, 30 to 1<br>Sulphurized mineral oil<br>Water, with 1 1/2 % borax or soda | Soluble oil, 30 to 1<br>Sulphurized mineral oil<br>Mineral lard oil<br>Sulphurized mineral lard oil<br>Water, with 1 1/2 % borax or soda |
| High-carbon steel | Soluble oil, 20 to 1<br>Mineral oil<br>Water, with 1 1/2 % borax or soda | Soluble Oil, 20 to 1<br>Mineral oil<br>Water, with 1 1/2 % borax or soda |
| Alloy steel<br>Monel metal<br>Nickel | Soluble oil, 20 to 1<br>Sulphurized mineral oil<br>Water, with 1 1/2 % borax or soda | Soluble oil, 20 to 1<br>Sulphurized mineral oil<br>Water, with 1 1/2 % borax or soda<br>Mineral lard oil |
| Copper | Soluble oil, 50 to 1<br>Light mineral oil<br>Sulphurized mineral oil<br>Water, with 1 1/2 % borax or soda | Light mineral oil<br>Water, with 1 1/2 % borax or soda |
| Brass or bronze | Soluble oil, 50 to 1 | Light mineral oil |
| Aluminum, duralumin, pewter, zinc | Light mineral oil | Soluble oil, 50 to 1<br>Very light mineral oil<br>Water, with 1 1/2 % borax or soda |

## 5. FACTORS AFFECTING RATE OF METAL REMOVAL

**SET-UP MATERIAL.**—Provision of an adequate quantity of well-chosen and well-arranged set-up material will save much time in making ready the average job. More time is lost on non-repetitive work through failure to provide an adequate equipment of set-up material, with racks or cabinets for keeping it in an orderly manner, than through any other failure on the part of management.

A supply of the frequently-used equipment should be kept at each machine tool. The cost is low, when compared with the reduction in set-up time made possible by a carefully engineered equipment of make ready material, and should prove to be a very profitable investment.

**EFFECT OF LENGTH OF TOOL EXTENSION.**—Tools should be so clamped in the tool post as to be supported, as closely as possible, directly beneath the active cutting edge. Excessive overhang may cause chatter and make necessary an appreciable reduction of speed, feed, or both. The following figures, based on the shop practice of a leading manufacturer of machine tools in the use of large forged planer tools, give an indication of this effect:

| Tool extension, in. | 0–8 | 8–12 | 12–20 | 20–34 | 34 and over |
|---|---|---|---|---|---|
| Reduction in cut and feed, percent | 0 | 25 | 50 | 60 | 65 |

**EFFECT OF WORK SET-UP ON RATE OF METAL REMOVAL.**—The size of chip removed and the cutting speed used depend on the stiffness of the work, the amount it overhangs the supporting surface, the amount of unbalance, and the rigidity with which it is clamped. Experience has shown that at times, when the part is weak, and cannot be solidly held, it is necessary to reduce the rate of removing metal as much as 75% over that possible with stiff, well-held work.

**EFFECT OF ATTENUATION OF WORK ON RATE OF METAL REMOVAL.**—When long, slim work must be turned on centers, the ratio between length and diameter of the work piece has an appreciable effect on the size of chip which may be taken, and on the cutting speed that may be used. Fig. 15 shows the effect on production rates of variations in the ratio of unsupported length to the diameter of the work, when cutting S.A.E. No. 1035 hot-rolled steel, with a tool having a $^1/_{16}$-in. nose radius, 90-deg. entering angle, 6-deg. nose drag angle, 14-deg. side rake, 8-deg. back rake, 6-deg. relief angle, when taking a cut $^1/_{16}$ in. deep, with a feed of 0.033 in., used for the roughing cut. The finishing cut was taken with a similar tool having a $^1/_{16}$ in. flat, with a depth of cut of 0.015 in. and a feed of 0.033 in. The curves of Fig. 15 have been used successfully, in the shop in which they were developed, for setting production rates on a variety of long slim shafts approximately 1 to 2 in. diameter, supported on centers and made in lots of 1 to 5. The extent to which they are applicable to conditions different from those for which they were developed is uncertain. They, at least, may be used as an indication of the reduction in the rate of chip production which becomes necessary as work on centers decreases in stiffness.

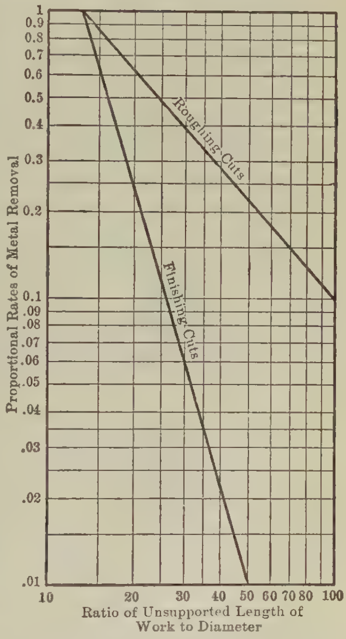
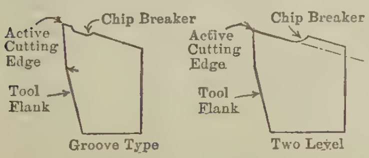


Fig. 15. Effect on Metal Removal of Ratio of Unsupported Length and Diameter of Work. Steel Work Piece Supported on Centers

**CHIP BREAKERS.**—The chip tends to flow over the face of a tool freshly ground in the usual way as a straight ribbon, gradually changing to a helix, which decreases in radius as the tool becomes worn. These long, straight chips are difficult to handle, and it frequently is desirable to force the chip into helical form by a device called a chip breaker.

A chip breaker either may be ground into the face of the tool, or may take the form of a supplemental piece, attached to the face of the tool. Fig. 16 shows two common forms of chip breaker which are ground into the face of the tool. A chip breaker is needed only when the chip comes off in the form of a long continuous helix. It is not necessary when cutting cast iron or other metals where the chip tends to break into short pieces.



Groove Type

Two Level

Fig. 16. Chip Breakers

**CHATTER** may be affected by several variables, of which the following probably are the most important: 1. Material cut; 2. Chip proportions, as affected by *a*, depth of cut, *b*, feed, *c*, tool contour; 3. Cutting speed; 4. Rigidity of work; 5. Stiffness of tool; 6. Stiffness of work support; 7. Stiffness of tool support; 8. Vibrations caused by nature of design of machine tool, such as the gear ratios and tooth forms used; 9. Setting of tool with respect to work.

Many of these variables are fixed by the design of the part to be machined, and the machine tool available for doing the work. It appears that the best method of controlling chatter under a given set of conditions is to vary the chip proportions, cutting speed, the rigidity of the work support, or the setting of the tool with respect to the work.

**Chip Proportions.**—Study of experimental data indicates that a cut taken with a relatively thick chip has less tendency to chatter than one taken with a thin chip. It would seem that it is desirable, in order to reduce chatter, to increase the feed, or to so change the shape of tool as to increase the thickness of chip with respect to its width. A tool having a nose radius greater than half the depth of cut is very apt to cause chatter unless a heavy feed is used.

**Cutting Speed.**—A reduction in cutting speed may reduce or eliminate chatter. Some experimenters have reported that an *increase* in cutting speed may reduce chatter.

**Rigidity of Work Support.**—Use of a center or follower rest frequently will eliminate chatter. The use of opposed tools, to balance the normal components of the cutting pressure, makes a very efficient method of reducing chatter.

**Setting of Tool.**—It has been found that a change in the setting of the nose of a tool with respect to the lathe center may affect the amount of chatter. Carl G. Barth has related to the author that in an aggravated case, setting of the tool nose considerably above the work center eliminated chatter. It is believed that this elimination resulted from the steadying effect of the contact between tool flank and work.

**TOOL GRINDING.**—Improper grinding of metal-cutting tools leads to appreciable losses in production. Failure to use the most effective tool contours and angles requires a reduction in the cutting speed. Grinding of tools by the individual machine operator causes a loss in production, as the machine tool must stand idle while tools are being ground. There is a growing tendency toward the grinding of all tools in a central tool room.

**Table 20.—Recommended Grinding Wheel Practice for Grinding Single-point Carbon Steel, High-speed Steel, and Stellite Tools**

| Grinding | | | | Wheel | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Grade | |
| Machine | Grind on | Wet or Dry | Rough or Finish | Diam., in. | Abrasive | Bond | Grain | Norton Co. Classification | Carborundum Co. Classification |
| Off-hand....... | Per.* | Dry | R | 6–12 | Al. oxide | Vit.‡ or Sil.§ | 24–40 | N–Q | H–K |
| " ....... | " | " | F | 6–12 | " | " | 46–80 | L–N | J–M |
| " ....... | " | Wet | R & F | 12–24 | " | " | 24–36 | O–P | I–K |
| " ....... | " | " | R & F | 24–36 | " | " | 24–36 | O–P | H–J |
| " ....... | " | " | R & F | 36–48 | " | Sil.§ | 24–30 | O–P | H–J |
| " ....... | Face† | " | R | ...... | " | Vit.‡ or Sil.§ | 30–36 | K–L | M–N |
| " ....... | " | " | F | ...... | " | " | 46–60 | J–K | N–O |
| Sellers semi-precision..... | Per.* | " | R & F | 15 | " | " | 36 | J–N | K–P |
| Sellers semi-precision..... | " | " | R & F | 24 | " | " | 24 | N–Q | H–K |
| Gisholt semi-precision..... | Face† | " | R & F | ...... | " | " | 24 | L–M | K–P |

* Periphery of wheel. † Cup wheel. ‡ Vitreous. § Silicate.

**Table 21.—Recommended Grinding Wheel Practice for Grinding Cemented Carbide Tools**

| Grinding | | | Wheel | | | | |
|---|---|---|---|---|---|---|---|
| | | | | | | Grade | |
| Machine | Grind on | Rough or Finish | Abrasive | Bond | Grain | Norton Co. Classification | Carborundum Co. Classification |
| Off-hand............ | Side | R | Silicon Carbide | Vit.‡ | 60–80 | H–I | P–S |
| " ............ | " | F | " | " | 100–120 | H–I | R–T |
| " ............ | | R & F | " | " | 60–120 | H–I | R–T |
| " ............ | Per.* | R | " | " | 60–80 | H–J | O–R |
| " ............ | " | F | " | " | 100–120 | H–J | O–R |
| Sellers semi-precision. | Per.* | R & F | " | " | 60–100 | H–I | P–T |
| Gisholt " | Face† | R & F | " | " | 60–100 | H–I | P–T |
| Lapping............ | Face<br>cylindrical lap | ......<br>......<br>...... | Diamond<br>Silicon Carbide<br>Boron Carbide | ...... | 280–320 | ...... | ...... |

* Periphery of wheel. † Cup wheel. ‡ Vitreous.

The type of grinding wheel to be used for tool grinding depends on the tool material, the area of contact between tool and wheel, whether the grinding operation is roughing or finishing, the surface of the wheel on which the grinding takes place, whether the tool is rigidly or loosely held with respect to the grinding wheel, and whether or not water is used.

Tables 20 and 21 summarize the recommendations of the two largest manufacturers of grinding wheels concerning the wheel that should be used under different conditions.

## 6. TOOL MATERIALS

The materials used for making tools for the cutting of metals may be classified in the following groups: 1. Carbon tool steel; 2. Low-alloy tool steels; 3. High-speed steel; 4. Non-ferrous alloys; 5. Cemented carbides; 6. Diamond.

**CARBON TOOL STEEL.**—A tool steel containing little or no alloying metals, and with a carbon content of from 1.10 to 1.30% is used for cutting operations where a hard, keen cutting edge is required, at low cutting speeds, particularly when a high quality of finished surface is desired. It is used at times for forming tools, particularly where it is desirable to form the tool before hardening. It also is used at times where the size of the tool or work, coupled with speeds of available shop equipment, limit cutting speeds.

**LOW-ALLOY TOOL STEELS** include the fast finishing steels, and some steels used for twist drills.

**HIGH-SPEED STEEL** is a steel containing at least 12% tungsten, 7% molybdenum or an equivalent combination of the two, together with appreciable percentages of chromium and vanadium, and, at times, cobalt. Analyses of the steels in most common use are given in Table 22. High-speed steel tools are much more generally used than those of other tool materials. They combine superior cutting qualities, strength, toughness and relatively low first cost for a very wide range of general machining operations. Cutting qualities vary with composition. See Table 23. For history of the development of high-speed steel see F. W. Taylor, The Art of Cutting Metals, *Trans.* A.S.M.E., xxviii, 1, 1908.

In general, the cobalt steels may be run at higher speeds, particularly on heavy roughing cuts. They are more brittle and weaker, when hardened to secure maximum cutting speeds, and are appreciably more expensive than the standard high-speed steels. They must be hardened at temperatures approximately 50 deg. F. higher than the equivalent standard steels. Particular care, therefore, is necessary in hardening to prevent surface burning and decarburization. The use of cobalt steel largely is confined to tools in which the contour is formed by grinding after hardening, such as lathe and planer roughing and finishing tools, tool bits, and inserted blades for milling cutters.

**NON-FERROUS ALLOYS.**—Of the several tool materials in this class, the best known are Stellite and No. 548 alloy. These materials occupy a position between the high-speed steels and the cemented carbides. They can be run at cutting speeds approximately 50% greater than those possible with high-speed steel, particularly when roughing cast iron, brass, and bronze. Their price is approximately 3 $^1/_3$ times that of high-speed steel. Tools of these materials are extremely brittle, and great care is necessary to prevent tool breakage or chipping of the cutting edge.

Many useful data on the performance of Stellite tools will be found in a paper by W. A. Becker, F. E. Gordon and W. A. Wissler, *Trans.* A.S.M.E. MSP–53–11a, 1931. They state that Stellite has been found most useful for cutting cast-iron, although it has been used successfully on steel under certain conditions. A typical cast-iron turning job ran at 87 ft. per min. with a $^5/_{32}$-in. cut and a feed of 0.05 in. per rev. The tool had a 5-deg. side rake and 6-deg. relief angle. Turning of 1020 S.A.E. steel was done at 110 ft. per min., with a feed of 0.021 in. per rev. and a cut of

**Table 22.—Materials Used for Metal Cutting Tools**

| Type | Carbon | Tungsten | Molybdenum | Chromium | Vanadium | Cobalt |
|---|---|---|---|---|---|---|
| **High-speed Steels** | | | | | | |
| 18–4–1 | 0.70–0.75 | 18.0 | ....... | 4.0 | 1.0 | ....... |
| 18–4–2 | 0.80–0.85 | 18.0 | 0.75 | 4.0 | 2.0 | ....... |
| 18–4–3 | 0.85–1.10 | 18.0 | ....... | 4.0 | 3.0 | ....... |
| 14–4–2 | 0.70–0.75 | 14.0 | ....... | 4.0 | 2.0 | ....... |
| Moly-Tungsten | 0.80–0.85 | 1.4 | 9.0 | 4.0 | 1.0 | ....... |
| Molybdenum | 0.80–0.85 | ....... | 10.0 | 4.0 | 1.0 | ....... |
| **Cobalt Steels** | | | | | | |
| 18–4–1 + 4% Cobalt | 0.70–0.75 | 18.0 | 0.5 | 4.0 | 1.0 | 4.0 |
| 18–4–2 + 7% Cobalt | 0.75–0.80 | 18.0 | 0.75 | 4.0 | 2.0 | 7.0 |
| 20–2 $^1/_2$–1 $^1/_2$ + 12% Cobalt | 0.80–0.85 | 20.0 | 1.00 | 4.5 | 1.5 | 12.0 |
| 14–4–2 + 4% Cobalt | 0.75–0.80 | 14.0 | 0.25 | 4.0 | 2.0 | 4.0 |
| Moly-Tungsten + 5% Cobalt | 0.80–0.85 | 1.4 | 9.0 | 4.0 | 1.0 | 7.0 |

1/8 in. The tool had 10 deg. side rake, and 7 deg. relief angle. For turning bronze at 131 ft. per min. with a feed 0.021 in. per rev., and a cut of 1/16 in., a tool with 10 deg. side rake, 10 deg. relief, and 5 deg. negative back rake was used.

**CEMENTED CARBIDE TOOLS** are made of tungsten, tantalum and titanium carbides in different proportions, cemented into a material usable for metal cutting tools by means of a matrix of cobalt. Such tools usually are in the form of small tips, which are brazed to a tool shank made of a relatively tough and inexpensive steel.

The cemented carbides are extremely hard and brittle, and may be run at cutting speeds approximately 3 1/2 times those possible with high-speed steels for the same cuts. Great care must be used to prevent chipping of the cutting edge. Such tools are particularly useful when cutting materials of low tensile strength which have an abrasive action on the tool. Carbide tools used for cutting steel usually contain appreciable proportions of tantalum and titanium carbides, which reduce the tendency of the chip to adhere to the tool.

The following tool angles are recommended for cemented carbide tools:

| | Cast Iron | | | Steel | | | Stainless | Steel Casting | | Bronze | Alum- |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Material | Soft | Hard | Chilled | Soft | Hard | 12%Mn | Steel | Soft | Hard | Brass | inum |
| Lip angle, deg. | 74–80 | 74–80 | 82–60 | 60–65 | 65–74 | 80–84 | 65–74 | 68–73 | 73–78 | 65–75 | 50–55 |
| Relief angle, deg. | 5 | 4 | 3 | 6 | 5 | 4 | 5 | 5 | 5 | 6 | 8 |

For uninterrupted cuts, or for material with blowholes, etc., or where there is excessive vibration, the back rake and relief angles should be reduced to give a larger cutting angle and bring the point of support more nearly under the cutting edge.

Thomas Prosser & Sons, New York, give the values in Table 23 as good averages for cutting speeds in ft. per min., feeds in in. per rev., and maximum depths of cut, in. for cemented carbide tools.

**Table 23.—Feeds and Speeds for Cemented Carbide Tools**

| | Aluminum | Bronze | Cast Iron | Cast Iron | Cast Iron ‡ | Steel | Steel | Steel | Stainless Steel § | 12%Mn Steel | Steel Casting ¶ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Hardness* | | | 200B † | 400B † | 75–90S | 130B † | 225B † | 410B | 225B | 225B | 280B |
| | | | | | Roughing Cuts | | | | | | |
| Speed | 3300 | 1000 | 250–300 | 130–250 | 13–20 | 325–375 | 200–230 | 80–115 | 115–150 | 25–65 | 100–130 |
| Feed | 0.040 | .040 | .040–.080 | .040–.080 | .080–.120 | .040 | .040 | .040 | .040 | .012–.020 | .040 |
| Cut Depth | 3/16–3/8 | 3/16–3/8 | 3/16–3/8 | 3/16–3/8 | 1/8–1/4 | 3/16–3/8 | 3/16–3/8 | 3/16–3/8 | 5/32–5/16 | 1/8–3/8 | 3/16–3/8 |
| | | | | | Finishing Cuts | | | | | | |
| Speed | 4000 | 1000–1650 | 260–400 | 165–260 | 13–30 | 725–1000 | 230–300 | 100–130 | 200–230 | 65–80 | 130–165 |
| Feed | 0.008 | .008 | .008 | .008 | .160–.320 | .008 | .008 | .008 | .008 | .008 | .008 |
| Cut Depth | .040 | .040 | .040 | .040 | .040 | .040 | .040 | .040 | .040 | .040 | .040 |

* B = Brinell; S = scleroscope. † Maximum. ‡ Chilled. § 18–8 type.
¶ Surfaces containing sand and blowholes.

**DIAMONDS.**—Diamonds cut and polished to the desired contour, and set in an appropriate form of holder, are used as cutting tools, particularly when it is desired to turn or bore a part with extreme accuracy; for example, the piston rod bushings of a gasoline engine. They usually are used for cutting relatively soft materials, as bronze, hard rubber, or molded synthetic resins, which may have considerable abrasive action on the tool.

## 7. HEAT TREATMENT OF TOOL STEELS

**CARBON STEEL** used for cutting tools should be hardened as follows:

**Anneal.**—Heat to 375–1425 deg. F., and cool slowly, either in furnace or buried in an insulating material, as lime, ashes, or infusorial earth. Tools should be annealed after each hot or cold working, to reduce hardness and prevent the formation of a coarse crystalline structure.

**Normalize.**—Heat tool uniformly to 1575–1650 deg. F., until there is complete penetration of the heat and refinement of the grain, and cool in still air.

**Preheat.**—Preheating of carbon steel tools is unnecessary unless the tool is exceptionally thick, or is extremely irregular in shape. If preheating is believed to be necessary, the tool may be placed in a cold furnace which is brought slowly to hardening temperature; or it may be given a preheat of approximately 900 deg F. before being placed in the hardening furnace.

**Hardening.**—Carbon tool steel should be hardened at temperatures which vary with the carbon content of the steel, as follows:

| Carbon content, % | 0.65–0.80 | 0.80–0.95 | 0.95–1.10 | 1.10 and over |
|---|---|---|---|---|
| Hardening temperature, deg. F. | 1450–1550 | 1410–1460 | 1390–1430 | 1380–1420 |

**Quenching.**—Carbon steel tools usually are quenched in a brine, consisting of sodium or calcium chloride in water, at about 70 deg. F. They may be removed from the brine and placed in oil while still retaining a considerable portion of their heat.

**Tempering.**—The hardening of steel tends to set up internal stresses which cause brittleness and a tendency for the cutting edge to break under cut. These conditions may be relieved by tempering as follows: To relieve strains, temper in oil at 300–375 deg. F.; to relieve strains and reduce brittleness, temper in oil at 375–500 deg. F.; to relieve strains and toughen, temper in lead at 500–650 deg. F. The strength of an untempered carbon steel tool, as measured by the torsion impact test, is approximately one-fourth of its strength when tempered to develop maximum strength.

**HIGH-SPEED STEEL.—Annealing.**—All high-speed steel should be annealed before machining, after forging, or before rehardening. Pack bars or tools in annealing boxes, allowing about 1 $^{1}/_{2}$ in. between tools and sides of box, using lime, mica, infusorial earth, ashes or, at times, charcoal, as the packing medium. Each box should be sealed with fireclay, raised slowly to 1550–1650 deg. F. in the furnace, and allowed to remain at the annealing temperature long enough to anneal thoroughly. This time varies with the size of the tools and the weight of the charge; it will range from 5 to 10 hr. Boxes and work should be allowed to cool slowly in the furnace.

**Preheat.**—Because of the low thermal conductivity of high-speed steel, it usually is brought to one or more intermediate temperatures in other furnaces, before it is placed in the hardening furnace. Tools usually are brought to a temperature of 1550 deg F. in a single stage of preheat. Large, intricate tools may be heated to approximately 1000 deg. F. in the first preheat furnace, and to 1550 deg. F. in the second, before hardening.

**Hardening.**—High-speed steels must be heated to a point close to their fusion point, in an atmosphere as nearly neutral as possible, to develop maximum efficiency as metal cutting tools. This may be obtained by manual control of the fuel and air supply; by enclosing the tool in a container lined with a material which gives off carbon monoxide when heated; by heating in a closed chamber the atmosphere of which is controlled by use of an auxiliary gas burner at the door; or by use of an inert gas, such as hydrogen, in a closed furnace chamber. There is a tendency toward the use of furnaces having some method of atmospheric control.

To obtain maximum practical cutting speeds, high-speed steel should be hardened at approximately the following temperatures:

| | 18–4–1 | 18–4–2 | 14–4–2 | 18–4–1+Co | 14–4–2+Co | W–Mo | W–Mo+Co |
|---|---|---|---|---|---|---|---|
| Hardening temp., furnace, deg. F. | 2400 | 2400 | 2350 | 2450 | 2400 | 2250 | 2300 |
| Hardening temp., bath, deg. F. | 2350 | 2350 | 2300 | 2400 | 2350 | 2200 | 2250 |

High-speed tools should remain in the hardening furnace sufficiently long to make certain that they have reached a uniform temperature, and that the thermal reactions have taken place completely. For a modern furnace, in which there is little tendency to burn the surface and edges of the tool, the following times are approximately correct:

| Tool thickness, in. | $^{1}/_{4}$ | $^{1}/_{2}$ | $^{3}/_{4}$ | 1 | 1 $^{1}/_{2}$ | 2 | 3 | 4 | 5 | 6 | 8 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Heating time in furnace, minutes | 2.0 | 3.0 | 4.0 | 5.0 | 7.0 | 8.5 | 12 | 15 | 18 | 20 | 25 | 30 |

These times are based on complete penetration of the heat. It frequently is undesirable or impracticable to use hardening times as long as those required for the larger thicknesses. For very thick tools, the practical procedure is to harden to a depth sufficient to give a satisfactory cutting edge, leaving the interior of the tool relatively soft.

**Quenching.**—High-speed steel usually is quenched in oil, or an air blast. If large tools are cooled too rapidly, there is danger of causing shrinkage cracks, and it may be desirable first to quench in a lead bath at 900–1000 deg. F., and to complete the quenching in oil.

**Straightening When Quenching.**—A tool that warps during quenching may be straightened within the few minutes immediately following quenching. The tool cannot be straightened later, as it becomes hard within a very short time after quenching.

**Tempering.**—After the hardened tool has been reduced to atmospheric temperature, it should be reheated to approximately 1000–1100 deg. F., held at that temperature for from 15 to 240 min. and cooled in air, in order to obtain maximum cutting efficiency and maximum transverse strength. This second heat treatment both increases the cutting qualities and the strength of the tool, and reduces brittleness. The maximum strength under impact, as measured by the torsion impact test, is obtained when the tool is given a secondary heat treatment at 900 deg. F.

**Heat Treatment in Small Shops.**—Where it is impracticable to install a modern type of heat treating furnace in a shop, tools should be purchased in the hardened condition, preferably in the form of bits. It is unwise to attempt to harden high-speed steel in the blacksmith's forge because of the uneven results obtained.

# PLANER WORK

**By Forrest E. Cardullo**

The planer is primarily a tool for generating flat surfaces, although by the use of special tools, fixtures and attachments, it may be used for forming surfaces, for planing curved surfaces, helixes, arcuate surfaces, and for doing various types of special work. Commercial planers are built in sizes to take work from 2 ft. wide by 2 ft. high by 5 ft. long, to work 16 ft. wide by 12 ft. high by 60 ft. long.

**FIELD OF THE PLANER.**—Work of the type usually performed on a planer, often may be done on a milling machine, shaper, slotter, or even on a lathe or boring mill. While each of these tools is preeminent in certain fields, in other fields they are competitive. The principal competitor of the planer is the milling machine, and it is desirable, therefore, to delimit the respective fields of these two tools.

The planer is preeminent in roughing large flat surfaces, especially when much finish is to be removed; in the accurate finishing of flat surfaces on medium-sized and larger work, especially if the material be cast iron; in the accurate finishing of such work as guides or slides when a number of pieces of identical cross-section may be placed in line and finished in one setting; in machining extremely hard materials.

The milling machine is preeminent in machining small surfaces, especially when the depth of cut is small, the material is soft, or extremely high cutting speeds may be used. The milling machine is preferable also when a large number of pieces are to be formed, or when the area to be machined is a small fraction of the area passed over by the tools, as is the case when raised bosses and strips are to be finished.

On ductile materials, the planer produces a more accurate but less smooth finish than the milling machine. On cast iron the planer produces a smoother, as well as a more accurate finish. Large steel castings usually are better planed than milled.

The planer is more versatile and adaptable than the milling machine for all types of work for which it is fitted. It is adapted to perform a number of operations in sequence on a single piece, or a few pieces. The set-up costs and tool costs are comparatively small. In the hands of a skilled operator it is extremely accurate. These properties obviously fit it for work on large pieces, for jobbing work, for tool room work, for work in shops building a general line of machinery, for emergency and repair work, and for work where accuracy is paramount, as in the building of machine tools. There is an overlapping field where either machine will produce satisfactory work at practically the same cost.

**POWER REQUIREMENTS OF PLANERS.**—Planers commonly are designed so that the power required at the moment of reversal is somewhat less than the maximum power delivered under a heavy cut. With the belts in proper condition, they will transmit about 4 Hp. per inch of width of cutting belt. The G. A. Gray Co., Cincinnati, recommends the following sizes of motors for planers. For light service, belt-driven: 24-in., 5 Hp.; 30-in., 7 1/2 Hp.; 36-in., 10 Hp. For medium service, belt-driven: 36-in., 15 Hp.; 42-in., 15 Hp.; 48-in. and over 20 Hp. For medium and heavy service, reversing motor drive: 36-in., 20 Hp.; 42-in., 25 Hp.; 48-in., 35 Hp.; 60-in. and 72-in., 50 Hp.; switch and frog planers, 50 Hp. The powers specified are ample for the simultaneous use of two tools. For more than two tools or for extra large planers, more power than is given above sometimes may be used to advantage, especially if the cutting speed be high. In such cases the planer builder should be consulted.

The power required to drive a planer depends on the rate of metal removal. When working to good advantage a planer tool cuts a rather thick chip, without much distortion, a condition favorable to economy of power. Planers usually are provided with from one to four tool-carrying heads, each head designed to withstand a tool pressure of from 5000 to 40,000 lb., depending on the size and type of planer. Such tool pressures can be obtained only under favorable circumstances. Usually the tool pressure is limited by the depth of cut, the feed which can be taken without damage to the tool or work, the extension of the tool, and the rigidity and strength of the work. It frequently is desirable to substitute a gang of three or four tools for a single heavy tool in order to avoid undue shock of entry, the breaking out of the work at the end of the cut, and the distortion of the work which would be produced by a heavy feed when taking a roughing cut. In general, the more tools that can be made to cut at once, the greater the economy of operation. Planers have been built in which 24 roughing tools and 16 finishing tools were simultaneously at work on eight pieces. Large heavy planers should have 75 Hp.

motors. For unusual service more power may be used to advantage. A 48-in. planer, roughing steel forging with four tools, may use 70 Hp. advantageously, and special multiple tooled planers, using a coolant at high speed, may require over 100 Hp.

**CUTTING AND RETURN SPEEDS.**—Cutting speed is dictated by the material being cut, and the kind of tool used. The return speed then is adjusted to give the greatest number of strokes in a given time. Belt-driven planers usually have only one cutting speed, from 30 to 45 ft. per min., and one return speed, from 80 to 100 ft. per min. As tool steels improve, the tendency is to increase the speed, but return speeds higher than 100 ft. per min. are not practical for belt-driven planers. When the speeds can be varied, as where planers are driven by a variable speed reversing motor, the usual cutting speeds are as follows:

| | |
|---|---|
| Cast iron (roughing) | 35 to 70 ft. per min. |
| " " (finishing) | 25 to 45 " " " |
| " " (chilled) | 2 to 6 " " " |
| Steel castings | 30 to 60 " " " |
| " forgings (soft) | 40 to 60 " " " |
| Steel forgings (hard alloys) | 12 to 30 ft. per min. |
| Bronzes | 40 to 60 " " " |
| Brass | 50 to 75 " " " |
| Aluminum alloys | 70 to 200 " " " |

All metals except cast iron are roughed and finished at the same speed. When large surfaces are to be finished accurately it is necessary to conserve the edge of the tool by using a wide feed and low cutting speed.

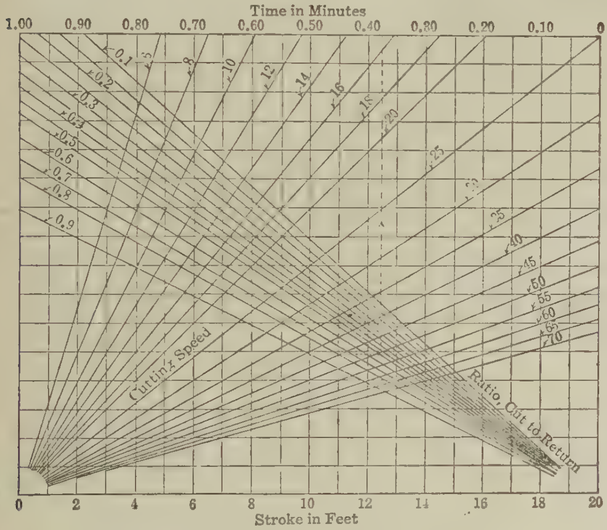


Fig. 1. Chart for Determining Cutting Speeds

**PLANER FEEDS.**—For rough planing cast iron, feeds range from $^1/_{16}$ in. to $^3/_8$ in. according to the depth of cut, power of the machine, and stiffness of the work and tools. For steel, the corresponding feeds are one-half those for cast iron. Down feed and undercut work require fine feeds. Frequently the feed must be reduced to prevent the cut from breaking out at the end. Cast iron finishing feeds range from $^1/_4$ in. to $1\,^1/_2$ in.

**TIME REQUIRED FOR PLANING.**—The number of round trips may be determined by dividing the width of the surface in inches by the feed in inches or fractions. From one to three extra round trips usually are necessary.

The time required for a round trip is the time required for the cut stroke plus the time required for the return stroke plus the time lost at the two reversals. The length of stroke usually is from 2 to 8 in. greater than the length of the work. To reduce this overrun, the feed should occur during the return stroke whenever this is possible. When gang tools are used, the overrun is increased by the distance between the cutting edges of the front and rear tools of the gang. The time lost during the two reversals varies from 0.015 to 0.020 min. with a first-class belt drive, to 0.03 min. with a good reversing motor, or 0.06 min. with a logy motor or one running at very high return speed.

Let $L$ = length of stroke, ft.; $B$ = backing or return speed, ft. per min.; $C$ = cutting speed, ft. per min.; $R$ = time lost in reversal per round trip, min.; $S$ = time per round trip, min.; $N$ = round trips per hr.; $F$ = feed, in.; $W$ = width of surface planed, in.; $T$ = time; $A = C/B$. Then

$$S = (L/B) + (L/C) + R \text{ minutes} = (L/C)\,(1 + A) + R \text{ minutes.}$$

$$N = 60/S = (60\,C) \div \{L(1 + A) + CR\} \text{ strokes per hour.}$$

$$T = WS/60F = W/NF \text{ hours.}$$

The chart, Fig. 1, gives values of $S$ which can be substituted in the above equations. To use the chart, first divide the cutting speed by the return speed, to obtain the value $A$.

From the intersection of the diagonal representing the cutting speed $C$, and the vertical representing the length of stroke $L$, proceed horizontally to the intersection of this horizontal with the diagonal representing $A$, thence vertically to read the value $S$ at the top of the chart. The chart assumes $R = 0.01$. The dotted line shows the method of finding $S$, which is 0.377 min., for a stroke of 11 ft., a cutting speed of 45 ft. per min., and a return speed of 90 ft., which gives a value of $A$ of 0.5. The time required to rough a piece of work requiring 11 ft. stroke, having a width of 18 in., and using a feed of $^1/_8$ in., and the above speeds, would be $T = (18 \times 0.377)/(60 \times {}^1/_8) = 0.904$ hr. To this, the time for setting up and removing work setting and adjusting tools, etc., must be added.

**DEVELOPMENTS IN PLANERS.**—The great majority of planers at present (1937) in service were built prior to the introduction of modern features of design. Helical or herringbone gearing, forced feed lubrication to vees and main bearings, oil clarifying systems, centralized lubrication of heads and gear boxes, power rapid traverse, improved feed mechanisms, which replace the old-fashioned friction feed, improved rail clamping mechanisms, and many other features have been introduced since 1920. These make machine operation and maintenance easier, increase rigidity and accuracy, reduce time for repairs, and increase the efficiency as a machine tool. Estimated savings due to such features vary from 20% to 40%, making it possible to replace three old-type planers with two modern machines, and at the same time obtain more accurate and satisfactory work.

**TOOLS AND FIXTURES.**—In many shops planers are not well equipped with tools and fixtures, and hence are able to operate at only a fraction of potential capacity. A study of the planer department often will result in increased efficiency when proper tools, fixtures, gages and other accessories are provided.

**High-speed Steel Tools** generally are used for roughing. High tungsten and tungsten-cobalt steels are preferred. High carbon steels often are used for fine finishing, but high tungsten steels with a high carbon content (special high temper) give practically as good results at much higher speeds. High carbon steels are preferred for planing chilled cast iron, being used at very low speeds. Blades of high-speed steel or of stellite sometimes are brazed to alloy steel shanks. High-speed steel bits also are used in carbon or alloy steel holders. A large variety of such bits and holders are used both for roughing and finishing and are successful and economical for a wide variety of work.

**The Superhard Cutting Tools** are not as satisfactory in planer service at the present time (1937) as high-speed steel. They lack the strength and toughness which normal planer service requires, and are apt to be chipped by careless handling of the machine. They are of use in planing dirty castings, in cleaning castings deficient in finish, where scale must be cut, and in planing work containing hard inclusions. While their usefulness at the present time is limited, it is entirely possible that their physical properties may be developed to the point where they will largely displace high-speed steel.

**A Set of Tools for General Work** includes right- and left-cut round-nose tools for roughing cast iron; the same for steel; a round-nose tool without side rake, for grooving; square-nose tools for second cuts and for "cutting down"; right- and left-cut side finishing tools; and gooseneck finishing tools. All of these tools may be obtained either in the forged type or the inserted bit type. The inserted bit tools usually are superior as finishing tools and for light cuts at high speeds, because of the superiority of their heat treatment. Forged tools usually are necessary when working in constricted openings, or close to shoulders or ledges.

**Other Useful Tools,** are chamfering tools, slotting tools, dovetail roughing tools, corner rounding tools, filleting tools, elling tools, self-relieving undercutting tools, and forming tools. Fig. 2 shows general purpose tools recommended by the G. A. Gray Co.

Unlike work done on machines intended for mass production, planer work seldom receives the attention of the production engineer. It usually is planned by the operator and his methods must be adapted to the fixtures and equipment available. Without the necessary tools and fixtures, even the most skilled and intelligent operator cannot work efficiently. Each planer should be supplied with two or more tools of each type that it is likely to require, so that it need not be shut down for forging or sharpening tools. As to fixtures and equipment, a variety of clamps should be provided, and also clamp bolts of various lengths, stop pins with adjustable screws, jacks, parallel blocks, washers, slot nuts, "file pieces" for holding down thin work, "pipe struts" for bracing high work, slot blocks for resetting parallel work, angle plates, squaring plates which can be doweled to the table for setting work when two surfaces are required to be square with each other, and special fixtures for holding work which must be done frequently. Since the planer can remove metal rapidly a good part of the floor to floor time on a planer job is set-up time. Adequate equipment for setting up work is essential to the efficient use of a planer. Such equipment cuts down set-up time, enables the operator to take heavier cuts, reduces the chance of accident, and produces more accurate work.

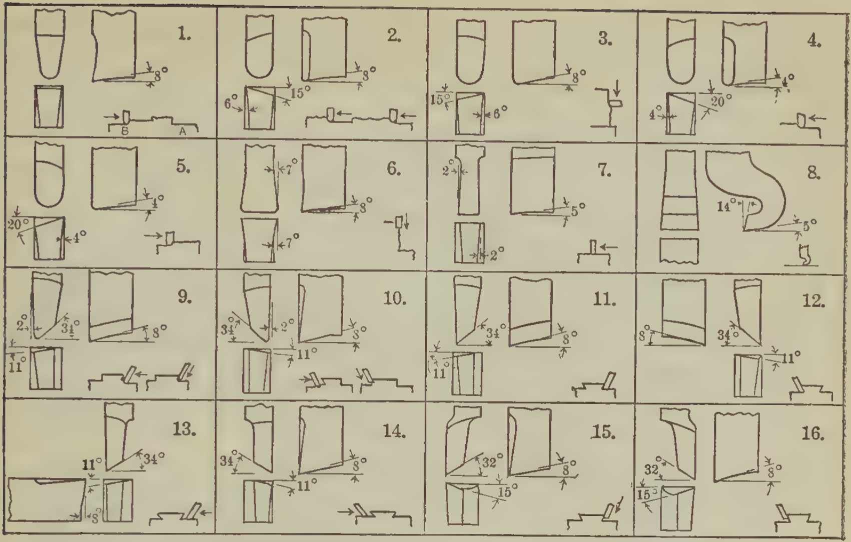


Fig. 2. Standard Shapes of Planer Tools. (See notes below)

In addition to the above equipment for each planer, the planer department should have an accurate and rigid four-sided cast iron square with diagonal bracing, a long and stiff straight edge with parallel sides, a sine bar, a large adjustable outside micrometer, a large adjustable inside micrometer, and two or three dial gages with universal mounting attachments. Dial gages may be used to advantage on the micrometers, and in gaging angles.

Finally, careful design of parts to be planed often will save much time. Lugs for supporting, bracing, stopping, measuring, or clamping pieces frequently are of advantage, later being planed or chipped off. Holes for lifting by chains, hooks, or clamps often are advantageous. Parts frequently are made in one piece at great expense, when a cheaper, more accurate and even more rigid design could be made in two pieces.

## Notes on Planer Tools. (See Fig. 2)

**Tool No. 1. Round-nose Roughing Tool for Cast Iron.**—Material, high-speed steel. For general-purpose, light roughing. Can be used in feeding from left to right or right to left. In planing two surfaces A and B at the same time, the tool is first fed from the right over surface A and then without changing setting, is fed from left over surface B. Tool has no side rake and maximum depth of cut is $^1/_2$ in.

**Tools No. 2 and No. 3. Right-cut and Left-cut Round-nose Roughing Tools for Cast Iron.**—Material, high-speed steel. Two of each of these tools required for a planer with two rail heads. Tool No. 2 feeds from right to left; tool No. 3 from left to right; also feeds down with the right-hand side head.

**Tools No. 4 and No. 5. Right-cut and Left-cut Round-nose Roughing Tools for Steel.**—Material, high-speed steel. Similar to tools Nos. 2 and 3, but ground for roughing cuts in steel. If used for cast iron, they will pull in and cause chatter. Tool No. 4 is used for planing from right to left and tool No. 5 from left to right, or for feeding down with right-hand side head. For removing a large amount of stock with these tools, use a deep cut, heavy feed, and slow speed.

**Tool No. 6. Square-nose Roughing Tool for Cast Iron.**—Material, high-speed steel. Used for roughing on flat surfaces where a sharp corner is to be made. Also can be used for straightening or heavy finishing cuts (depth of cut 0.004 to 0.005 in.). For lighter cuts and finer finishes use tool No. 8. Tool No. 6 can be made by brazing a high-speed steel tip to a machinery steel shank.

**Tool 7. Square-nose Finishing Tool for Cast Iron.**—Material, high carbon steel. A general purpose tool for straightening and finishing cuts. Advisable to have several different widths from $^3/_8$ to 1 in. Cutting edge of tool should be exactly parallel to surface of work. Feed should be coarse, almost the width of the tool.

**Tool No. 8. Gooseneck Finishing Tool for Cast Iron and Steel.**—Material, high carbon steel. Used to finish flat surfaces in any metal. Used with a very shallow cut and coarse feed. Cutting edge of tool should be finished with oilstone by hand, leaving a keen straight edge. Cutting edge should be exactly parallel to surface of work.

**Tools No. 9 and No. 10. Right-hand and Left-hand Dovetailing End Cutting Roughing Tools for Cast Iron.**—Material, high-speed steel. Cutting edge of tool at the end. Corner rounded off to avoid breakdown in taking roughing cut. Tool No. 9 is fed downward or from right to left. Tool No. 10 is fed from left to right and downward. Tools Nos. 9 and 10 should be followed by tools Nos. 11 and 12 to leave a sharp corner.

**Tools No. 11 and No. 12. Right-hand and Left-hand Dovetailing End Cutting Roughing Tools for Cast Iron.** Material, high-speed steel. These tools are similar to tools Nos. 9 and 10, and are intended to clean out corners after most of the metal has been removed by the other tools. Not as well suited for general dovetail roughing as tools Nos. 9 and 10, because the sharp corner breaks down. Tool No. 11 feeds downward or from right to left; tool No. 12 feeds in the opposite direction and downward.

**Tools No. 13 and No. 14. Right-hand and Left-hand Dovetail End Cutting Finishing Tools for Cast Iron.**—Material, high carbon steel. For finishing flat surfaces with cutting edge at end of tool. Used after roughing cuts with tools Nos. 9, 10, 11, and 12. Tool No. 13 feeds from right to left and tool No. 14 from left to right.

**Tools No. 15 and No. 16. Right-hand and Left-hand Dovetail-side Cutting Finishing Tools for Cast-iron.**—Material, high carbon steel. Used for finishing angular surface of dovetail. Tool No. 15 feeds downward with cross feed, taking very light cut. Tool No. 16 feeds downward with cross feed and light cut.

**General Notes on All Tools.**—For fair finish, set roughing tool to final dimensions. The finishing tool merely removes the high spots. For accurate finish, leave work 0.006 in. oversize when roughing, followed by straightening cut leaving 0.001 in. to be removed by finishing cut. Tools should be made from the following sizes of bar. From 24-in. to 36-in. standard planers, 1 × 1 1/2 in. bar; 36-in. to 48-in. maximum service planers, 1 1/8 × 1 3/4 in. bar; 56-in. to 72-in. maximum service planers, 1 1/4 × 2-in. bar.

## Planer Drives

Small planers usually are driven by shifting belts, the "cut" belt being crossed and the "return" belt open. Such belts must be kept pliable, but not sticky. They are stretched somewhat more tightly than is good practice in other types of machinery, in order to get a belt narrow enough to shift readily, and still deliver the requisite power. Belt-driven planers usually operate at about 33 ft. per min. cutting speed and 90 ft. per min. return speed and deliver about 4 Hp. per inch of width of cutting belt.

The maximum satisfactory belt width is 4 in. Belt drives usually are used for very small planers and, when direct current is not available, for reversing motor drive. Belt-driven planers reverse more quickly than reversing motor-driven planers, although modern (1937) reversing motors are much more efficient in this respect than motors built a few years previously. The belt drive gives only one cutting speed, unless provided with means for varying the speed of the overhead cutting belt pulley independently of the speed of the overhead return belt pulley. In such cases, cutting speeds ranging from 25 to 60 ft. per min. usually are provided.

For heavy work, planers 36 in. wide and over usually are operated by 230-volt, reversing, variable-speed, direct-current motors with field control. Such motors are made, having speeds ranging from 250 to 1000 r.p.m., from 200 to 1200 r.p.m., and from 150 to 900 r.p.m. Planers usually are so geared that ten revolutions of the motor shaft moves the table one foot or more. Switch planers are so geared that 13 to 15 revolutions of the motor shaft moves the table one foot. The field control should be so arranged that the entire speed range is available for both cut and return speeds. Table speeds of 25 to 100 ft. per min. formerly were standard. Higher speeds are coming into favor, and maximum table speeds of 120 to 200 ft. per min. are not unusual. Variable-voltage drives consisting of a special motor-generator set for each planer, energizing a variable-voltage motor, are used when extreme speed ranges are required. Such drives, while expensive, give motor speeds ranging from 40 to 1200 r.p.m., and make possible table speeds ranging from 4 to 120 ft. per min., or with special gear ratios, speeds up to 300 ft. per min.

Hydraulically-driven planers lose less time in reversal than reversing-motor or belt-driven planers. For short strokes and light and medium work, such planers often are found to be satisfactory. However, the hydraulic cylinder located immediately below the table becomes very hot in service, causing unequal expansion of the planer members, and making precision work difficult. Hydraulic planers are not adapted for heavy duty work or long stroke work because of their lack of power and their tendency to chatter, due to the compressibility of the liquid.

In Europe, some planers have cutting speeds of 100 to 250 ft. per min. There are no tools available to operate at such speeds and give reasonable tool life, except on aluminum, similar alloys, and some non-metallic materials.

# MILLING MACHINE PRACTICE

By Hans Ernst and Mario Martellotti

## 1. ANALYSIS OF THE PROCESS OF MILLING

**FUNDAMENTAL CONCEPT.**—The milling process may be considered as a method of metal removal in which the portion of the work piece that is to be removed is projected into the space dominated by a rotating body carrying one or more individual cutting teeth. The profile of the finished surface produced on the work piece, therefore, is determined by the intersection of a plane, perpendicular to the motion of the work piece, and the outline of the body of revolution generated by the rotating cutter.

Two main classes of milling operations are:

1. **Face Milling,** in which the finished surface mainly is parallel to the face of the cutter.
2. **Peripheral Milling** (including slab and form milling), in which the finished surface mainly is parallel to the periphery of the cutter.

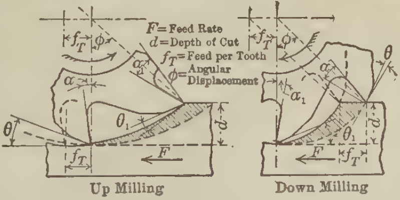
**CHARACTERISTIC FORM OF MILLING CHIP.**—Because of the limited period of engagement of each tooth with the work piece, the removal of metal in milling is accomplished by the separation of small individual chips. The rotary motion of the cutter and the translatory motion of the work piece combine to produce a chip whose cross-section varies from instant to instant as the direction of motion of a tooth changes with respect to direction of motion of the work piece. Chip thickness is a maximum at the instant that these motions are perpendicular to each other, and substantially zero when they are parallel.

**SURFACE GENERATED IN MILLING.**—In peripheral milling, two distinct methods are used:

**Up-milling.**—The work piece advances toward the cutter from the side where the teeth are moving upward. The cutter rotates against the direction of the feed. The chip thickness is a minimum at the beginning of the cut and a maximum at the end.

**Down-milling.**—The work piece advances toward the cutter from the side where the teeth are moving downward. The cutter rotates in the same direction as the feed. The chip thickness is a maximum at the beginning of the cut, and a minimum at the end.

Fig. 1 compares the shape of milling chips produced by these two methods, the difference in shape being exaggerated by the use of a large feed per tooth. In both methods the path of a tooth through the work material is a curtate trochoid. Its actual shape will vary with changes in rate of feed, diameter of cutter, depth of cut, and cutting speed. Fig. 1 shows that the actual rake and clearance angles change continually as the tooth forms a chip. In up-milling, the actual clearance angle $\theta$ is greatest at the beginning of the tooth contact with the work, decreasing to a minimum value of $\theta_1$ when the tooth leaves the work. The actual rake angle, however, has a minimum value $\alpha$ at the beginning of the cut, increasing to maximum value $\alpha_1$ at the point of leaving the work. In down-milling, the actual clearance angle $\theta$ is least at the beginning of the cut and greatest, $\theta_1$, at the end; the actual rake angle $\alpha$ is greatest at the beginning of the cut, and least, $\alpha_1$, at the end.



Up Milling    Down Milling

FIG. 1

In this respect, milling differs radically from other machining processes, as turning, planing, etc., in which clearance and rake angles remain constant throughout the cut.

**CHIP FORMATION.**—Up-milling and down-milling produce surfaces of entirely different character, the reason lying in the nature of the process of chip formation. When a cutting tool engages a work piece, the material immediately ahead of the cutting edge is at first compressed and escapes by flowing in a direction parallel to the face of the tool. See Fig. 2. As the tool continues to advance, however, escape in this direction becomes increasingly difficult; the pressure on the tool rises; the stress in the material further ahead of the tool increases, finally reaching a value great enough to cause rupture or flow in a plane roughly perpendicular to the face of the tool. If the material is relatively brittle, or the thickness of chip is great, a definite rupture will occur in this plane. The entire portion of severed material then will move out of the path of the advancing tool, and permit the engagement of a fresh portion. The chip will not be a continuous ribbon, but a series of individual segments more or less closely joined. This type of chip is known as a segmental chip.

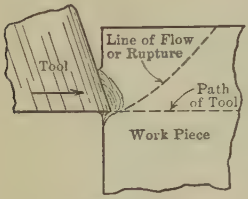


Fig. 2

If the material is relatively ductile, definite successive ruptures will not take place; instead, the material ahead of the tool will yield by plastic flow over a broad region. As this material (*A*, Fig. 3) is forced out of the path of the advancing tool, it separates from the initially compressed portion *B*, which is held by frictional contact with the tool face. The body of the chip passes off as a substantially solid ribbon, with flow lines perpendicular to the tool face. The wedge-shaped portion *B*, with flow lines parallel to the tool face, remains to form the so-called *built-up edge.* This type of chip generally is known as a flow-type chip. The built-up edge is added to continuously by material compressed at point *C*, but is periodically reduced by fragments, as *D* and *E*, torn off both the chip and work piece, respectively.

Roughness of a machined surface is due solely to the presence of fragments such as *E*; consequently, for good finish, their magnitude should be kept to a minimum by so arranging conditions that the stable size of the built-up edge itself is a minimum. Such arrangements include: 1. Decrease in chip thickness; 2. Increase in rake angle; 3. Increase in tool sharpness; 4. Increase of cutting speed; 5. Use of cutting fluid; 6. Use of work material of lower ductility. See H. Ernst and M. Martellotti, Metal Cutting, *Mech. Engg.*, Aug., 1935.

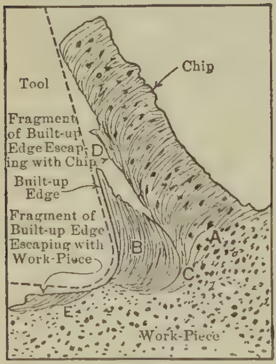

Chip
Tool
Fragment of Built-up Edge Escaping with Chip
D
Built-up Edge
Fragment of Built-up Edge Escaping with Work-Piece
B
A
C
E
Work-Piece

Fig. 3

In up-milling, the portion of the finished surface which is produced by a tooth is formed at the beginning of its contact with the work piece, when the chip thickness is extremely small. Hence, the built-up edge itself is very small, and the resultant surface is smooth and shiny. In down-milling an element of the finished surface is produced at the end of the tooth engagement with the work. With a ductile material such as a low-carbon steel, where the built-up edge is large, the latter persists to the extreme end of the chip; thus the escape of fragments likewise persists, and produces a somewhat torn finish similar to that obtained by planing or shaping. With a less ductile material, such as a high carbon steel, the built-up edge is smaller; hence, the difference in finish between up-and-down methods of milling is less marked. (See discussion of paper by O. W. Boston and C. E. Kraus, The Elements of Milling, *Trans.* A.S.M.E., RP 54–4, 1932, p. 96.)

## 2. MILLING CUTTERS

### Forms of Milling Cutters

The following classification and nomenclature relative to milling cutters are based on the standards approved by the American Standards Association, and adopted by the leading manufacturers. See A.S.A. *Bulletin*, 5C–1930. Tables 1 to 9 show principal dimensions of the most commonly used types of milling cutters, as approved by the American Standards Association. These dimensions have been adopted as standard by the cutter manufacturers.

**CLASSIFICATION BASED ON RELIEF OF TEETH.—1. Profile Cutters.**—All forms of cutters which are sharpened by grinding on the periphery of the teeth, the clearance (or relief) being obtained by grinding a narrow land back of the cutting edge. Where the cutting edges are of curved or irregular shape, they are designated as Shaped Profile Cutters.

**2. Formed Cutters.**—All cutters where the eccentric relief back of the cutting edge is of the same contour as the cutting edge itself. These cutters are sharpened by grinding the face of the teeth.

**CLASSIFICATION BASED ON METHOD OF MOUNTING.—1. Arbor Cutters.**—A cutter with hole for mounting on arbor.

**2. Shank Cutters** have either a straight or taper shank integral with the cutter.

**3. Facing Cutters** are designed to be attached directly to spindle end or to stub arbor.

**GENERAL TYPES OF CUTTERS.**—The commonly used types of cutters, and the work to which they are adapted are:

**The Plain Milling Cutter** is a cylinder with teeth on the circumferential surface only. It is used to produce a flat surface parallel to the axis of the cutter. Plain milling cutters are made in a wide variety of diameters and widths, for the various requirements of slab milling. They generally have helical teeth. The helical form enables each tooth to take a cut gradually, thus reducing shock and minimizing tendency to chatter. Cutters with a helix angle of 25 to 45 deg. commonly, but incorrectly, are termed *spiral mills*; if helix angle is more than 45 deg. the cutter generally is known as a *helical mill.*

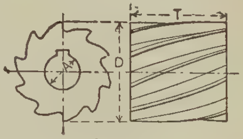


Fig. 4

The helical mill, which may be either of the shank or hole type, is free cutting and particularly desirable for light cuts on soft steel or brass. It can run at high speeds and, therefore, produces a good finish. It can be used to advantage for profiling work such as cam milling, and for milling intermittent surfaces. The shank type, with pilot end, commonly is used for elongating slots. For general slab milling, the helical mill is not as efficient as the common "spiral" mill.

**Table 1.—Dimensions of Heavy Duty Plain Milling Cutters**

See Fig. 4

| Cutter Diam., *D*, in. | | | Diam. of Hole, *A*, in. | | Nominal Widths of Face, *T*, in. |
|---|---|---|---|---|---|
| Nominal | Max. | Min. | Max. | Min. | |
| 2 1/2 | 2.515 | 2.485 | 1.001 | 1.000 | 2, 2 1/2, 3, 4 |
| 3 | 3.015 | 2.985 | 1.251 | 1.250 | 2, 2 1/2, 3, 4, 6 |
| 4 | 4.015 | 3.985 | 1.501 | 1.500 | 2, .... 3, 4, 6 |
| 4 1/2 | 4.515 | 4.485 | 2.001 | 2.000 | .. .... .. 6, 12 |

Limits of *T*

| | | | | | | |
|---|---|---|---|---|---|---|
| Nominal | 2.0 | 2 1/2 | 3.0 | 4.0 | 6.0 | 12.0 |
| Maximum | 2.010 | 2.520 | 3.020 | 4.020 | 6.020 | 12.020 |
| Minimum | 2.000 | 2.500 | 3.000 | 4.000 | 6.000 | 12.000 |

**The Side Milling Cutter** is a plain milling cutter of cylindrical form with teeth on the circumferential surface and on both sides. The side teeth extend a portion of the distance from the circumference to the axis. Half side and interlocking side milling cutters also are made. Side milling cutters are used in a large variety of work. Two or more of them often are placed on the same arbor with a space between them, they then being known as straddle mills.

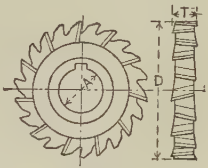


Fig. 5

**Straddle Mills** are used to advantage where the work is to be milled on two parallel sides; *e.g.*, bolt heads, tongues, etc.

**Table 2.—Dimensions of Stagger Tooth Milling Cutters**

See Fig. 5

| Cutter Diam., *D*, in. | | | Diam. of Hole, *A*, in. | | Nominal Widths of Face, *T*, in. |
|---|---|---|---|---|---|
| Nominal | Max. | Min. | Max. | Min. | |
| 2 1/2 | 2.515 | 2.485 | 0.876 | 0.875 | .... 1/4, 5/16, 3/8, .... 1/2 ... ... .... .. |
| 3 | 3.015 | 2.985 | 1.001 | 1.000 | 3/16, 1/4, 5/16, 3/8, .... 1/2, 5/8, 3/4 .... .. |
| 4 | 4.015 | 3.985 | 1.251 | 1.250 | .... 1/4, 5/16, 3/8, 7/16, 1/2, 5/8, 3/4, 7/8 .. |
| 5 | 5.015 | 4.985 | 1.251 | 1.250 | .... ... .... ... .... 1/2, 5/8, 3/4 .... .. |
| 6 | 6.015 | 5.985 | 1.251 | 1.250 | .... ... .... 3/8, .... 1/2, 5/8, 3/4, 7/8, 1 |
| 8 | 8.015 | 7.985 | 1.501 | 1.500 | .... ... .... 3/8, .... 1/2, 5/8, 3/4, ... 1 |

Limits of *T*

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Nominal | 3/16 | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 | 7/8 | 1 |
| Maximum | 0.1875 | .2500 | .3125 | .3750 | .4375 | .5000 | .6250 | .7500 | .8750 | 1.0000 |
| Minimum | 0.1870 | .2495 | .3120 | .3745 | .4370 | .4995 | .6245 | .7495 | .8740 | .9990 |

**Interlocking Cutters** are used to mill slots to a standard width. They are maintained at a constant width by thin shims or collars between the inner hubs.

**Staggered Tooth Milling Cutters** are cylindrical cutters with the cutting teeth on the circumferential surface only. Alternate teeth are of opposite helix angle. See Fig. 5. The side teeth extending from the circumference a short distance towards the axis are for chip clearance only. They are not ground for cutting purposes. This type of cutter is used to obtain exact width of slots and is the most efficient type for milling slots where depth exceeds width. Because of the alternate right- and left-hand helix angle of the teeth, with considerable undercut, these cutters can remove large amounts of metal without vibration or chatter. The free cutting action makes possible an increased feed and speed, without detriment to the production of smooth, accurate work.

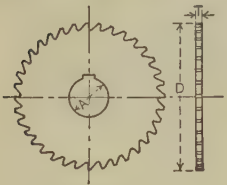


Fig. 6

**Metal Slitting Saws** are plain milling cutters with sides relieved or "dished" to give side clearance. They generally are made 3/16 in. thick or less, and usually have more teeth for a given diameter than a plain milling cutter. They are used to cut off work, or to mill very narrow slots. Slitting saws also are made with side teeth similar to side milling cutters, and with staggered teeth, similar to staggered tooth milling cutters. The metal slitting saw with staggered teeth usually is made 3/8 to 3/16 in. thick, and is used for heavy sawing in steel.

**Table 3.—Dimensions of Metal Slitting Saws**

See Fig. 6

| Cutter Diam., *D*, in. | | | Diam. of Hole, *A*, in. | | Nominal Thicknesses, *T*, in. | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal | Max. | Min. | Max. | Min. | | | | | | | |
| 2 1/2 | 2.515 | 2.485 | 0.876 | 0.875 | 1/32, | 3/64, | 1/16, | 3/32, | 1/8 | .... | ... |
| 3 | 3.015 | 2.985 | 1.001 | 1.000 | 1/32, | 3/64, | 1/16, | 3/32, | 1/8, | 5/32 | ... |
| 4 | 4.015 | 3.985 | 1.001 | 1.000 | 1/32, | 3/64, | 1/16, | 3/32, | 1/8, | 5/32, | 3/16 |
| 5 | 5.015 | 4.985 | 1.001 | 1.000 | .... | .... | 1/16, | 3/32, | 1/8, | 5/32, | 3/16 |
| | | | 1.251 | 1.250 | .... | .... | .... | .... | 1/8 | .... | ... |
| 6 | 6.015 | 5.985 | 1.001 | 1.000 | .... | .... | 1/16, | 3/32, | 1/8, | .... | 3/16 |
| | | | 1.251 | 1.250 | .... | .... | .... | .... | 1/8, | .... | 3/16 |
| 7 | 7.015 | 6.985 | 1.001 | 1.000 | .... | .... | 1/16, | .... | 1/8 | .... | ... |
| | | | 1.251 | 1.250 | .... | .... | .... | .... | 1/8, | .... | 3/16 |
| 8 | 8.015 | 7.985 | 1.001 | 1.000 | .... | .... | .... | .... | 1/8 | .... | ... |
| | | | 1.251 | 1.250 | .... | .... | .... | .... | 1/8, | .... | 3/16 |

Limits of *T*

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Nominal............. | 1/32 | 3/64 | 1/16 | 3/32 | 1/8 | 5/32 | 3/16 |
| Maximum............ | 0.0322 | .0478 | .0635 | .0947 | .1260 | .1572 | .1885 |
| Minimum............ | 0.0302 | .0458 | .0615 | .0927 | .1240 | .1552 | .1865 |

**Angle Milling Cutters** are made both **single angle** and **double angle**, with teeth on the conical surfaces. See Figs. 7 and 8. Single angle cutters are made both with and without teeth on one or both of the flat sides. Angle cutters are used for various fluting operations, or for milling the edge of a piece to a given angle.

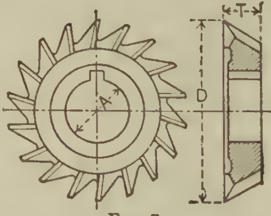


Fig. 7

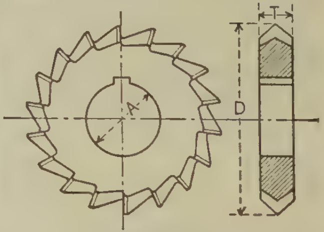


Fig. 8

**Table 4.—Dimensions of Angle Milling Cutters**

| Cutter Diam., *D*, in. | | | Diam. of Hole, *A*, in. | | Single Angle Cutter, Fig. 7* Thickness, *T*, in. | | | Double Angle Cutter, Fig. 8† Thickness, *T*, in. | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Nominal | Max. | Min. | Max. | Min. | Nominal | Max. | Min. | Nominal | Max. | Min. |
| 2 1/2 | 2.515 | 2.485 | 0.876 | 0.875 | 1/2 | 0.515 | 0.485 | 1/2 | 0.515 | 0.485 |
| 2 3/4 | 2.765 | 2.735 | 1.001 | 1.000 | 1/2 | .515 | .485 | | | |
| 3 | 3.015 | 2.985 | 1.251 | 1.250 | 1/2 | .515 | .485 | | | |

* Included angle 45 or 60 deg. ± 10 min. † Included angle, 45, 60 or 90 deg. ± 10 min.

**End Mills** are cutters with teeth on the circumferential surface and on one end. See Fig. 9. The teeth may be either parallel to the axis of rotation or helical, and either right or left hand. End mills of moderate angle commonly are called spiral end mills. End mills are made of five general types, *viz.*, Solid end mill; two-lip end mill or slotting mill; shell end mill; hollow mill; helical end mill.

**Solid End Mills** are used for light milling operations, as the milling of slots, profiling, and facing narrow surfaces. The teeth are integral with the shank, which may be either straight or tapered.

**Two-lip End Mills** comprise a shank cutter with two cutting teeth on the circumferential surface, and end teeth cut to the center. Flutes are either straight or helical. The cutter can be sunk into the material and then fed longitudinally. A depth of cut equal to one-half the diameter of the mill usually can be taken from the solid stock.

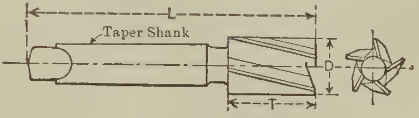


FIG. 9

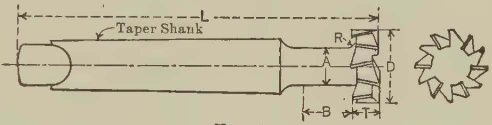


FIG. 10

**Shell End Mill.**—Teeth are cut on the circumferential surface and on one end. See Fig. 11. The tooth end is recessed to receive a nut or screw head for holding the cutter on the stub arbor. It usually is driven from the key slot across the back face. The teeth may be parallel to the axis of rotation, or helical, and either right or left hand. This type of milling cutter should be used in preference to the solid end mill whenever possible, because it is cheaper to replace when worn out or broken.

**Hollow Mill.**—A cutter of tubular construction with teeth on one end, and having an internal clearance. The internal clearance sometimes is obtained by a plain tapered hole with back taper and sometimes by internal cleared flutes. The hollow mill generally is used for producing bosses or cylindrical projections from solid bodies.

**T-slot Cutter.**—A shank cutter designed for milling T-slots. See Fig. 10. The teeth are on the circumferential surface and both sides. In making a T-slot, a groove first is cut with an ordinary side milling cutter or a two-lipped end mill. The wide groove at the bottom then is cut with the T-slot cutter.

**Table 5.—Dimensions of Taper Shank End Mills.** (See Fig. 9)

| Diam. of Cutter, *D*, in. | | | Brown & Sharpe Taper Shank No. | Length of Cut, *T*, in. | Overall Length, *L*, in. | Diam. of Cutter, *D*, in. | | | Brown & Sharpe Taper Shank No. | Length of Cut, *T*, in. | Overall Length, *L*, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal | Max. | Min. | | | | Nominal | Max. | Min. | | | |
| 1/4 | 0.2650 | 0.2500 | 5 | 5/8 | 2 13/16 | 3/4 | 0.7650 | 0.7500 | 7 | 1 1/4 | 5 1/4 |
| 5/16 | .3275 | .3125 | 5 | 11/16 | 2 7/8 | 7/8 | 0.8900 | 0.8750 | 7 | 1 7/16 | 5 7/16 |
| 3/8 | .3900 | .3750 | 5 | 3/4 | 2 15/16 | 1 | 1.0150 | 1.0000 | 7 | 1 5/8 | 5 5/8 |
| 7/16 | .4325 | .4375 | 5 | 7/8 | 3 1/16 | 1 1/8 | 1.1400 | 1.1250 | 9 | 1 3/4 | 7 |
| 1/2 | .5150 | .5000 | 5 | 15/16 | 3 1/8 | 1 1/4 | 1.2650 | 1.2500 | 9 | 2 | 7 1/4 |
| 1/2 | .5150 | .5000 | 7 | 15/16 | 4 15/16 | 1 1/2 | 1.5150 | 1.5000 | 9 | 2 1/4 | 7 1/2 |
| 9/16 | .5775 | .5625 | 7 | 1 | 5 | 1 3/4 | 1.7650 | 1.7500 | 9 | 2 1/2 | 7 3/4 |
| 5/8 | .6400 | .6250 | 7 | 1 1/8 | 5 1/8 | 2 | 2.0150 | 2.0000 | 9 | 2 3/4 | 8 |

**Table 6.—Dimensions of Taper Shank T-slot Cutters.** (See Fig. 10)

| Nominal Bolt Size, in. | Cutter Diam., *D*, in. | | Thickness of Cutter, *T*, in. | | Diam. of Neck,* *A*, in. | | Length of Neck, *B*, in. | Fillet Radius, *R*, in. | Brown & Sharpe Taper Shank No. | Overall Length, *L*, in. |
|---|---|---|---|---|---|---|---|---|---|---|
| | Max. | Min. | Max. | Min. | Max. | Min. | Min. | Max. | | |
| 1/4 | 0.562 | 0.552 | 0.234 | 0.229 | 0.265 | 0.260 | 3/8 | 0.007 | 5 | 2 5/8 |
| 5/16 | .656 | .646 | .265 | .260 | .328 | .323 | 7/16 | .007 | 5 | 2 23/32 |
| 3/8 | .781 | .771 | .328 | .323 | .406 | .401 | 9/16 | 1/64 | 7 | 4 13/16 |
| 1/2 | .968 | .958 | .390 | .385 | .531 | .526 | 11/16 | 1/64 | 7 | 5 |
| 5/8 | 1.249 | 1.239 | .484 | .479 | .656 | .651 | 7/8 | 1/64 | 7 | 5 1/4 |
| 3/4 | 1.468 | 1.458 | .625 | .620 | .781 | .776 | 1 1/16 | 1/64 | 9 | 6 7/8 |
| 1 | 1.843 | 1.833 | .828 | .823 | 1.031 | 1.026 | 1 1/4 | 1/64 | 9 | 7 1/4 |
| 1 1/4 | 2.218 | 2.208 | 1.093 | 1.088 | 1.281 | 1.276 | 1 9/16 | 1/64 | 9 | 7 13/16 |
| 1 1/2 | 2.655 | 2.645 | 1.343 | 1.338 | 1.531 | 1.526 | 1 15/16 | 1/64 | 10 | 10 3/8 |

* Neck diameters as given are to be used with American Standard T-slots.

**The Woodruff Key Seat Cutter** is made in the **shank** and **arbor** type. The shank type generally has teeth on the circumferential surface only, with the sides slightly relieved for clearance. The arbor type generally is used in sizes larger than 2 in. diameter. These cutters also are made with staggered teeth. They are made with teeth on both sides which are ground for clearance only and not for cutting. Both types of cutters are used for the specific purpose of milling semi-cylindrical keyways in shafts, to permit the use of Woodruff keys.

**Formed Cutters** usually are of curved irregular outline. They are sharpened by grinding the faces of the teeth. So long as the face of a tooth is maintained in its original plane with respect to the axis of rotation, the contour of the tooth will remain unchanged. Formed cutters are used for duplicate interchangeable work over and over again until they have been ground to a point where the teeth are too slender to stand the strain of the work.

In the formed cutter class are included such cutters as: *Gear Cutters, Multiple Gear Cutters, Sprocket Cutters, Convex Cutters, Concave Cutters, Corner Rounding Cutters, Spline Cutters, Thread Milling* and *Hobbing Cutters*, etc.

**A Fly Cutter** consists of a single cutter similar in shape to a planer tool, held and rotated by an arbor. As it has but a single cutting edge, it is used but rarely outside of the experimental room or tool room. The fly cutter can be formed exactly to any desired shape. Its field is operations that would not bear the expense of a special form cutter.

**Inserted-tooth Facing Cutter.**—This type of cutter, generally known as a **Face Mill,** is a cylindrical body with slots on its periphery. Blades of various cutting materials, as high-speed steel, Stellite, or blades tipped with cemented carbide, are inserted in the slots. The teeth of the facing cutter are ground to cut both on the circumferential surface and on the side, as in the **Side Mill.** These cutters are adapted to be attached directly to the end of the spindle, or to a stub arbor.

**Face Mills** are used to face large surfaces, and are very efficient in the removal of metal.

*The Heavy Duty Face Mill* is made with a stronger body, and thicker and fewer blades than the light duty type. It generally is used for roughing cuts.

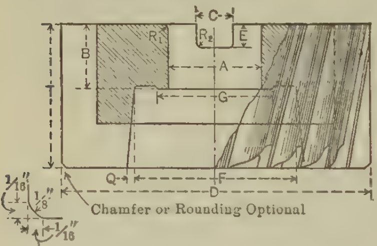


Fig. 11

*The Light Duty Face Mill* has a lighter body and a larger number of teeth than the heavy duty type, and is used mostly for finishing cuts. Face mills with high-speed steel inserted blades, tipped with cemented carbide, or with stellite blades, are used almost exclusively for milling cast iron and non-ferrous materials. They have more teeth and usually are run from two to three times faster than face mills with high-speed steel blades.

The inserted tooth milling cutter is most economical and useful when cutters of large dimensions are required, although they are obtainable also in diameters as low as 3 in. Almost all types of milling cutters may be made with inserted teeth. Their use is becom-

**Table 7.—Dimensions of Shell End Mills.** (See Fig. 11)

| Size or Diam., *D*, in. | Width, *T*, in. | Hole | | | | Driving Slot | | | | Fillet Radius, *R*, in. | Counterbore | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diam., *A*, in. | | Depth, *B*, in. | | Width, *C*, in. | | Depth, *E*, in. | | | Diam., *F*, in. | Diam., *G*, in. | Angular Increase, *Q*, deg. |
| | | Max. | Min. | Max. | Min. | Max. | Min. | Max. | Min. | | | | |
| 1 1/4 | 1 | 0.5005 | 0.5000 | 41/64 | 5/8 | 0.262 | 0.258 | 11/64 | 5/32 | 1/64 | 11/16 | 5/8 | 0 |
| 1 1/2 | 1 1/8 | .5005 | .5000 | 41/64 | 5/8 | .262 | .258 | 11/64 | 5/32 | 1/64 | 11/16 | 5/8 | 0 |
| 1 3/4 | 1 1/4 | .7505 | .7500 | 49/64 | 3/4 | .324 | .320 | 13/64 | 3/16 | 1/32 | 15/16 | 7/8 | 0 |
| 2 | 1 3/8 | .7505 | .7500 | 49/64 | 3/4 | .324 | .320 | 13/64 | 3/16 | 1/32 | 15/16 | 7/8 | 0 |
| 2 1/4 | 1 1/2 | 1 0005 | 1.0000 | 49/64 | 3/4 | .387 | .383 | 15/64 | 7/32 | 1/32 | 1 1/4 | 1 3/16 | 0 |
| 2 1/2 | 1 5/8 | 1.0005 | 1 0000 | 49/64 | 3/4 | .387 | .383 | 15/64 | 7/32 | 1/32 | 1 3/8 | 1 3/16 | 0 |
| 2 3/4 | 1 5/8 | 1.0005 | 1.0000 | 49/64 | 3/4 | .387 | .383 | 15/64 | 7/32 | 1/32 | 1 1/2 | 1 3/16 | 5 |
| 3 | 1 3/4 | 1.2505 | 1.2500 | 49/64 | 3/4 | .512 | .508 | 19/64 | 9/32 | 1/32 | 1 21/32 | 1 1/2 | 5 |
| 3 1/2 | 1 7/8 | 1.2505 | 1.2500 | 49/64 | 3/4 | .512 | .508 | 19/64 | 9/32 | 1/32 | 1 11/16 | 1 1/2 | 5 |
| 4 | 2 1/4 | 1.5005 | 1.5000 | 1 1/64 | 1 | .637 | .633 | 25/64 | 3/8 | 1/16 | 2 1/32 | 1 7/8 | 5 |
| 4 1/2 | 2 1/4 | 1.5005 | 1.5000 | 1 1/64 | 1 | .637 | .633 | 25/64 | 3/8 | 1/16 | 2 1/16 | 1 7/8 | 10 |
| 5 | 2 1/4 | 1.5005 | 1.5000 | 1 1/64 | 1 | .637 | .633 | 25/64 | 3/8 | 1/16 | 2 9/16 | 1 7/8 | 10 |
| 5 1/2 | 2 1/4 | 2.0005 | 2.0000 | 1 1/64 | 1 | .762 | .758 | 29/64 | 7/16 | 1/16 | 2 13/16 | 2 1/2 | 10 |
| 6 | 2 1/4 | 2.0005 | 2.0000 | 1 1/64 | 1 | .762 | .758 | 29/64 | 7/16 | 1/16 | 2 13/16 | 2 1/2 | 15 |

ing more and more common, even in small diameter cutters, particularly with cemented carbides.

**DIRECTION OF ROTATION.**—A cutter is **right-hand** if it rotates counter-clockwise, and **left-hand** if it rotates clockwise, when viewed from the front end as mounted on the spindle.

**MATERIAL FOR MILLING CUTTERS.**—Four classes of materials commonly used for milling cutters are: Carbon steel; high-speed steel; Stellite; cemented carbide.

**High-speed Steel Cutters** predominate, although carbon steel is used to some extent for screw slotting and slitting cutters and other light duty work. Practically all heavy-duty plain milling cutters are made of high-speed steel. For most classes of work, high-speed steel cutters can be run from 2 to 2 1/2 times as fast as carbon steel cutters.

**Stellite** is employed generally in the form of inserted blades in face mills for use on cast iron. The life of Stellite cutters is generally longer than that of similar high-speed steel cutters and the permissible cutting speed higher.

**Cemented Carbide.**—Inserted-blade face milling cutters, tipped with various types of cemented carbides, are (1937) replacing other types of cutters for use on cast iron. Cutting speeds normally employed are from 2 to 3 times as high as those used with high-speed steel cutters, and a considerable improvement in finish usually is obtained. Because of the extreme hardness of these carbides, a smaller allowance for finish is permissible, thus further increasing production and reducing costs.

**NUMBER OF TEETH.**—No standard rule exists for the number of teeth in milling cutters. Modern cutters (1937) generally are of the " coarse tooth " type, the spacing offered commercially by cutter manufacturers having been found satisfactory for most purposes.

The coarse tooth cutter provides a great chip space between successive teeth, which permits a free flow of the chip and prevents clogging of the teeth. The efficiency of the modern coarse tooth cutter is considerably higher than that of the old style fine tooth cutter. See A. L. DeLeeuw, Milling Cutters and their Efficiency, *Trans.* A.S.M.E., xxxiii, p. 245, 1911, also J. A. Hall and B. P. Graves, Effect of Variation in Design of Milling Cutters on Power Requirements and Capacity, *Trans.* A.S.M.E., lxv, p. 165, 1923.

**Table 8.—Dimensions of Concave Milling Cutters.** (See Fig. 12)

| Diameter of Circle, *C*, in. | | | Diam. of Cutter, *D*, in. | Thickness of Cutter, *T* ± 0.010, in. | Diameter of Hole, *A*, in. | | |
|---|---|---|---|---|---|---|---|
| Nominal | Max. | Min. | | | Nominal | Max. | Min. |
| 1/8 | 0.1280 | 0.1240 | 2 | 1/4 | 7/8 | 0.876 | 0.875 |
| 3/16 | .1905 | .1865 | 2 | 3/8 | 7/8 | .876 | .875 |
| 1/4 | .2530 | .2490 | 2 | 7/16 | 7/8 | .876 | .875 |
| 5/16 | .3155 | .3115 | 2 1/4 | 9/16 | 7/8 | .876 | .875 |
| 3/8 | .3780 | .3740 | 2 1/4 | 5/8 | 7/8 | .876 | .875 |
| 7/16 | .4405 | .4365 | 2 1/4 | 3/4 | 7/8 | .876 | .875 |
| 1/2 | .5040 | .4980 | 2 1/4 | 13/16 | 7/8 | .876 | .875 |
| 5/8 | .6290 | .6230 | 2 3/4 | 1 | 1 | 1.001 | 1.000 |
| 3/4 | .7540 | .7480 | 3 | 1 3/16 | 1 | 1.001 | 1.000 |
| 7/8 | .8790 | .8730 | 3 1/4 | 1 3/8 | 1 | 1.001 | 1.000 |
| 1 | 1.0050 | .9980 | 3 1/4 | 1 9/16 | 1 | 1.001 | 1.000 |

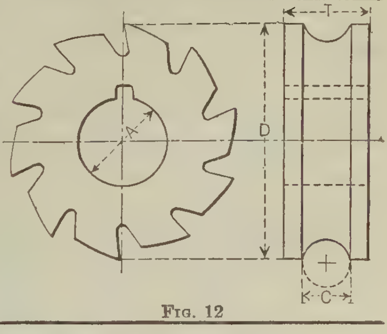


FIG. 12

**Table 9.—Dimensions of Convex Milling Cutters.** (See Fig. 13)

| Diameter of Circle- *C*, in. | | | Diam. of Cutter, *D*, in. | Nominal Thickness, *T*, in. | Diameter of Hole, *A*, in. | | |
|---|---|---|---|---|---|---|---|
| Nominal | Max. | Min. | | | Nominal | Max. | Min. |
| 1/8 | 0.1270 | 0.1230 | 2 | 1/8 | 7/8 | 0.876 | 0.875 |
| 3/16 | .1895 | .1855 | 2 | 3/16 | 7/8 | .876 | .875 |
| 1/4 | .2520 | .2480 | 2 | 1/4 | 7/8 | .876 | .875 |
| 5/16 | .3145 | .3105 | 2 1/4 | 5/16 | 7/8 | .876 | .875 |
| 3/8 | .3770 | .3740 | 2 1/4 | 3/8 | 7/8 | .876 | .875 |
| 7/16 | .4395 | .4355 | 2 1/4 | 7/16 | 7/8 | .876 | .875 |
| 1/2 | .5030 | .4970 | 2 1/4 | 1/2 | 7/8 | .876 | .875 |
| 5/8 | .6280 | .6220 | 2 3/4 | 5/8 | 1 | 1.001 | 1.000 |
| 3/4 | .7530 | .7470 | 3 | 3/4 | 1 | 1.001 | 1.000 |
| 7/8 | .8780 | .8720 | 3 1/4 | 7/8 | 1 | 1.001 | 1.000 |
| 1 | 1.0040 | .9960 | 3 1/4 | 1 | 1 | 1.001 | 1.000 |

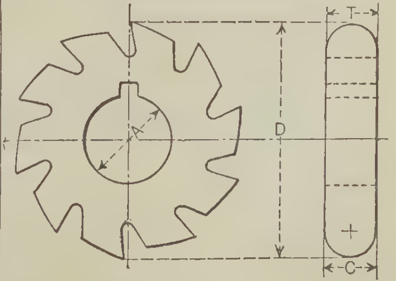


FIG. 13

Table 10.—Cutting Speed of Milling Cutters on Various Materials

| Material | High-speed Steel | Stellite | Cemented Carbide |
|---|---|---|---|
| | Feet per Minute | | |
| Cast iron, 160 B* | 70–80 | 130–180 | 200–350 |
| " " 200 B* | 60–70 | 100–140 | 100–250 |
| Malleable iron | 75–95 | 95–110 | ......... |
| 1020 S.A.E. steel | 75–95 | ......... | ......... |
| 1050 S.A.E. steel | 70–80 | ......... | ......... |
| 3145 S.A.E. steel | 60–70 | ......... | ......... |
| Tool steel, annealed | 50–55 | ......... | ......... |
| Low-carbon steel forging | 75–95 | ......... | ......... |
| Tough alloy steel forging | 60–75 | ......... | ......... |
| Steel castings | 65–75 | ......... | ......... |
| Yellow brass | 200–500 | 500–700 | 500–1000 |
| Bronze, 40 R† | 200–350 | 350–500 | 500–1000 |
| Bronze, 75 R† | 75–95 | 150–250 | ......... |
| Aluminum | 800–2500 | 1000–4000 | 1000–4000 |

* B = Approximate Brinell hardness number. †R = Rockwell B scale.

## 3. USE OF MILLING CUTTERS

**CUTTING SPEED.**—Because of the countless variations in conditions encountered in milling practice, fixed rules for cutting speeds are impossible. The most effective speeds are governed by the structure and hardness of the part to be machined, the amount of metal to be removed, the type of cutting material, etc. In general, cutting speed is selected to give the best compromise between maximum production and maximum life of the cutter. The values given in Table 10 are conservative and represent average practice on production jobs.

The cutting speed for finishing milling often is 40% to 80% higher than the roughing speed.

**FEED RATE.**—The rate of production in milling depends directly on the rate at which the work passes the cutter, but the selection of the proper feed rate for milling a given work piece involves several factors, among them being the limitations of machine, cutter, work piece, and fixture; the finish desired, etc.

Estimates of feed rates usually are based on the assumption of a certain value for the feed per tooth. This is multiplied by the number of tooth passages per minute, as determined from the desired cutting speed and tooth spacing, to obtain the feed rate in inches per minute. This then must be checked with the above limitations.

In practice the feed per tooth is made as large as possible, consistent with these limitations and the permissible tooth load. Table 11 gives average values of permissible feed per tooth for various combinations of cutting material and work material. These values must be reduced when using cutters, or work, of a relatively fragile nature.

**DIAMETER OF CUTTER** usually is determined by the limitations of the work piece, either in regard to the size of the surface to be milled, or interference with adjacent

Table 11.—Permissible Feed per Tooth of Milling Cutters

| Material being cut | Feed per tooth, in. | | |
|---|---|---|---|
| | High-speed Steel Cutter | Face Mill with Stellite Blades | Face Mill Tipped with Tungsten Carbide |
| Cast iron, 160 B * | 0.018 | 0.014 | 0.012 |
| " " 200 B * | .015 | .012 | .010 |
| Malleable iron | .012 | .008 | .... |
| 1020 S.A.E. steel | .016 | .... | .... |
| 1050 S.A.E. steel | .014 | .... | .... |
| 3145 S.A.E. steel | .008 | .... | .... |
| Tool steel, annealed | .010 | .... | .... |
| Low carbon steel forging | .016 | .... | .... |
| Tough alloy steel forging | .010 | .... | .... |
| Steel castings | .015 | .... | .... |
| Yellow brass | .020 | .018 | .012 |
| Bronze, 40 R † | .018 | .016 | .012 |
| Bronze, 75 R † | .016 | .014 | .010 |
| Aluminum | .020 | .020 | .018 |

* B = Approximate Brinell Hardness Number. † R = Rockwell B Scale.

portions of work or fixture. In the case of cutters of small diameters, the limitation frequently is the size of the arbor required to support it properly.

**CLEARANCE ANGLE (Relief).**—The sole function of clearance or relief of the flank surface, is to prevent undue interference between it and the surface of the work. Where the direction of the motion of the flank surface, relative to the work, is parallel to the finished surface (as in the case of a shaper tool or planer tool) the clearance angle need be only sufficient to allow for the slight expansion of fragments of the built-up edge which escape with the work piece as the latter passes under the tool. Any clearance greater than the minimum amount required increases the likelihood of chatter, and reduces the life of the tool.

In the case of a peripheral cutting edge of a milling cutter, the direction of motion of the tooth relative to the direction of the feed varies with the position of the tooth in its arc of contact with the work piece. In up-milling, the motion of the tooth is parallel to the feed at the instant of engagement, and inclined thereto when leaving the work. Throughout its contact with the work piece, there is an increasing component of the feed tending to cause interference with the flank surface, and thus an additional clearance must be provided.

The additional clearance angle necessitated by the feed component depends on the diameter of the cutter, $D$, the feed per revolution, $F_r$, and the depth of cut $d$. Its value may be computed from

$$\theta = \frac{180}{\pi}\left\{\frac{2\,F_r}{D}\sqrt{\frac{\pi d(D-d)}{\pi(\pi D + F_r)^2 - F_r\,d}}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where $\theta$ is given in degrees, and $D$, $F_r$ and $d$ are in inches.

The minimum permissible clearance angle also is affected by the width of the flank (or *land*) back of the cutting edge. This should be kept to a small value, usually not greater than $^1/_{32}$ in., in order that the true clearance at the cutting edge may be as small as possible. As shown in Fig. 14, a secondary clearance angle beyond this land usually is provided to avoid interference, and to reduce the amount of material which must be removed when regrinding the cutter. The values of clearance angles commonly used range from 3 to 5 deg. for steel, to about 10 deg. for aluminum.

**RAKE ANGLE** of a milling cutter (see Fig. 14) is defined as the angle by which the face of the tooth is displaced back of the radial line drawn from the center of rotation to the cutting edge. Efficiency of metal removal increases with increase in the rake angle. Consequently, the latter should be as large as possible consistent with strength. In practice, rake angles ranging from 10 to 15 deg. commonly are used.

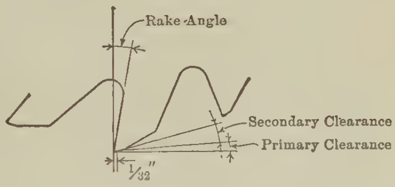


FIG. 14

**CUTTING FLUIDS,** consisting generally of oils or emulsions, are used to protect the cutter, and improve the quality of the finish. They have three main functions:

**1. Cooling.**—A large amount of heat is generated in metal cutting, particularly with strong ductile materials such as steel, where a flow type chip is produced. Heating of the tool point causes a reduction in resistance to abrasion, thus hastening dullness or wear. Heating of the material also may impair the finish by increasing the size of the built-up edge. For efficient cooling, a cutting fluid should have a high specific heat coefficient.

**2. Lubrication.**—By lubrication of the tool face, resistance to chip flow is reduced, thus reducing the size of the built-up edge and improving the finish, and also reducing the total force on the tool. This will increase the tool life. For efficient lubrication, the cutting fluid should have a low surface tension with respect to the tool and work material, and a high film strength to withstand the high specific pressures encountered.

**3. Flushing of Chips.**—The cutting fluid should be supplied in sufficient quantities to flush the chips away from the cutter and work, thus reducing the likelihood of marring the finished surface.

**Composition.**—Emulsions consisting of soluble oils and water are used widely in milling, because of their effectiveness as coolants, and the ease with which they may be used in large quantities for flushing away the chips. For milling of tough alloy steel forgings, however, heavy cutting oils often are used, the composition varying with the nature of the work and the finish required. Special compounded oils, such as sulphurized oils or chlorinated oils, are being used to an increasing extent because of their value as extreme pressure lubricants.

**COMPARISON OF UP-MILLING AND DOWN-MILLING.**—At various times during the past 50 years down-milling (or milling with the feed) has been advocated for particular classes of work, in view of certain advantages inherent in this method. In recent years this method has been used with special types of feed systems, designed to prevent an overrunning of the milling machine table due to the action of the cutter on the work.

When using thin metal slitting cutters the likelihood of producing crooked slots is less when cutting down than when cutting up. In milling the flutes in twist drills, and other similar operations, a longer tool life and more desirable finish has been obtained when milling down, and the holding of the work piece during the milling operation is greatly simplified. On the other hand, when milling a low carbon steel or other ductile material, the finish is much inferior than with the conventional up-milling method. The greatest practical objection to the down-milling method for heavy milling work is the danger of a serious accident if the work piece or fixture should be insecurely fastened to the table.

From the standpoint of power required there is little to choose between the two methods of milling. Careful tests have shown in most cases that there is very little difference between the power actually used by the cutter when removing chips either by up-milling or down-milling.

**POWER REQUIRED FOR MILLING** is a function of many factors, including the cutting speed, the feed, the material being cut, the rake angle, the sharpness of the cutter, the depth of cut, the smoothness of the face of the teeth, the type of cutting fluid used, etc.

An empirical formula which can be used for a rough determination of the power required in milling is

$$\text{Hp.} = KFWD \quad \ldots\ldots\ldots\ldots \quad [2]$$

where Hp. = horsepower actually used in cutting; $F$ = feed rate, in. per min.; $W$ = width of cut, in.; $D$ = depth of cut, in.; $K$ = a constant which is different for different materials; values are given in Table 12.

In order to obtain the total actual power required to perform a milling operation in any given type of milling machine, it is necessary to add to the result of calculation by formula [2], the idle power required to drive the machine. Idle power varies in relation to the type of milling machine used. However, it generally is found that the idle power is approximately constant within the range of loads encountered in ordinary milling machine operation. For more exact determination of the power involved in milling, the following formula can be used:

$$\text{Hp.} = CWf^{a}D^{b} \quad \ldots\ldots\ldots\ldots \quad [3]$$

where Hp. = actual horsepower used in cutting; $W$ = depth of cut, in.; $f$ = feed per tooth, in.; $D$ = depth of cut, in.; $C$ = a constant.

Constants $C$ and exponents $a$ and $b$ are determined by experiment. They vary with the material being cut, the type of cutter used, the nature of the cut and with up-milling or down-milling. O. W. Boston and C. E. Kraus (*Trans.* A.S.M.E., RP–56–1, 1934) discuss the results of tests with a single point tool and of a great number of commercial tests on milling. They present values of $C$, $a$ and $b$ for three steels, *viz.* S.A.E. 1020, S.A.E. 1112 and S.A.E. 3250, and also a free-cutting leaded brass of analysis Cu, 0.62; Zn, 0.34; Pb, 0.03. Cuts were made both in a groove and on a land, with cutters whose helix angles ranged from 0 to 40 deg., and whose rake angles ranged from 0 to 15 deg. The range of the values of the exponents and constants was as given in Table 13.

The following summarizes the variation of the exponents $a$ and $b$ and the constant $C$.

With down-milling, $C$ in most cases is lower than with up-milling. In general, $C$ for a 25-deg. rake angle is about 55% of its value for a 0-deg. rake angle, and varies but little for the greater

**Table 12.—Values of Constant *K***

| Material | *K* | Material | *K* |
|---|---|---|---|
| Cast iron | 0.5 | Alloy Steel | 2.0 to 2.5 |
| Malleable iron | 0.8 | Brass | 0.35 to 0.5 |
| Mild steel | 2.2 | Aluminum | 0.25 to 0.35 |
| High carbon steel | 2.2 to 2.5 | | |

**Table 13.—Values of Exponents and Constants in Horsepower Formula**

| | In Groove | | | On Land | | |
|---|---|---|---|---|---|---|
| | *a* | *b* | *C* | *a* | *b* | *C* |
| 1020 Steel | 0.70–0.79 | 0.87–1.07 | 5450–20,200 | 0.65–0.74 | 0.83–0.94 | 4130– 9,040 |
| 1112 " | 0.71–0.82 | 0.82–0.95 | 5270–12,780 | 0.68–0.78 | 0.82–0.90 | 3710– 8,350 |
| 3250 " | 0.68–0.77 | 0.86–0.98 | 6780–18,450 | 0.67–0.84 | 0.85–0.93 | 5280–13,360 |
| Brass | 0.80–0.91 | 0.89–1.07 | 3115– 9,540 | 0.77–0.89 | 0.86–0.97 | 2365– 4,785 |

angles of rake. The value of $C$ remains practically constant with all helix or side rake angles up to 30 deg., with a sharp increase at 40 deg., due probably to excessive metallic distortion of the chip.

The feed exponent $a$ remains practically constant as the rake angle increases, although a slight decrease is apparent for land cutting. It also remains practically constant as the helix increases, except that for extreme values of helix in groove cutting the value of $a$ is high.

The depth exponent $d$ decreases slightly with high rake angles, and increases slightly for high helix angles when milling in a groove. The exponent remains practically constant when milling is on a land.

**TYPICAL MILLING JOBS. SPEEDS. FEEDS.**—The following are representative examples of good commercial practice:

**Automotive Practice.**—1. Rough mill cylinder blocks, top and bottom, manifold, bearing cap slots, and valve cover pads, using Stellite tools: Cutting speed, 110 ft. per min.; feed, 10.5 in. per min.; depth of cut, $1/8$ in.; production, 13.6 pieces per hr.; pieces per grind, 200.

2. Mill clutch housing on cylinder blocks, using cemented carbide tools: Cutting speed, 250 ft. per min.; feed, 12 $1/2$ in. per min.; depth of cut, $1/32$ in.; production, 60 pieces per hr.; pieces per grind, 6720.

3. Finish-mill rear end of cylinder blocks, using cemented carbide tools: Cutting speed 250 ft. per min.; feed, 16 in. per min.; depth of cut, 0.015 in.; production, 44 pieces per hr.; pieces per grind, 3168.

4. Mill mounting pads of master cylinder body, using Stellite cutters: Cutting speed, 140 ft. per min.; feed, 11 $1/4$ in. per min.; depth of cut, $1/8$ in.; production, 130 pieces per hr.

5. Mill bosses on shock absorber body, using Stellite cutters: Cutting speed, 128 ft. per min.; feed, 25 in. per min.; depth of cut, $3/16$ in.

Table 14 gives data concerning the milling operations on a 12-cylinder automobile motor block.

**General Practice.**—1. Bottle mold halves; mill parting faces and tongue, using tungsten carbide and high-speed steel cutters: Material, cast iron; cutting speed, tungsten carbide cutters, 170 ft. per min.; high-speed steel cutters, 53 ft. per min.; feed rate, tungsten carbide cutters, 9 $7/8$ in. per min.; high-speed steel cutters, 3 $5/8$ in.; depth of cut, $1/16$ in.

2. Roller-spider; mill three cams using high-speed form cutter: Material, steel forging; cutting speed, 78 ft. per min.; feed variable, average, 5 $1/2$ in. per min.; depth of cut, $1/16$ in.; time per piece, 1.22 min.; production, 42 pieces per hour.

3. Transmission case; mill face, using high-speed steel cutters: Material, cast iron; cutting speed, 60 ft. per min.; feed, 5 in. per min.; production, 8 pieces per hr.

4. Compressor cylinder; Material, S.A.E. 1112 steel; high-speed steel slotting cutter: cutting speed, 64 ft. per min.; feed, 5 $3/8$ in. per min.; depth of cut, 0.025 in.; production, 229 pieces per hour.

5. Master cylinder body; mill mounting pads, using Stellite face mill cutter: Material, cast iron; cutting speed, 140 ft. per min.; table feed, 11 $1/4$ in. per min.; production, 130 pieces per hour.

6. Block plunger feed trip; mill slot and angle using a gang of high-speed steel cutters: Material, S.A.E. 1112 steel; cutting speed, 65 ft. per min.; feed rate, 2.4 in. per min.; depth of cut, 1 $1/2$ in.; production, 12 pieces per hr.

7. Fork; mill inside and outside of fork; mill radius and end, form-mill other radius and end. Material, steel casting; 1st and 2nd operation: Use gang of four high-speed steel cutters; cutting speed, 55 ft. per min.; feed rate, 5 $3/8$ in. per min.; depth of cut, $1/8$ in. 3rd operation: Use high-speed steel form cutter; cutting speed, 50 ft. per min.; feed rate, 3 in. per min.; depth of cut, $1/8$ in.; production, 6 pieces per hr.

8. Bracket; mill bottom and inside edges using high-speed steel face mill cutter and high-speed steel end mill: Material, cast iron; cutting speed, roughing, 80 ft. per min.; finishing, 120 ft. per min.; feed rate, roughing, 11 $3/4$ in. per min.; finishing, 9 $1/4$ in. per min.; depth of cut, $3/16$ in.; production, 2 pieces per hr.

## 4. MILLING MACHINES

**CLASSIFICATION AND USES.**—The types and sizes of milling machines are numerous and their designs merge into one another. It is, therefore, difficult to classify them. The following classification is based on general appearance or design.

**1. Knee and Column Type** is so called because the spindle is fixed in the column or main body, and the platen or table is capable of adjustment longitudinally, transversely, and vertically. Machines of this type are of rather universal application and generally

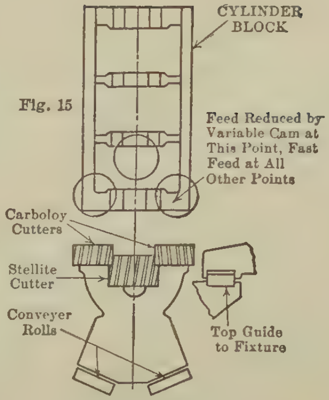


Fig. 15

1st and 2nd Operation
Rough and Finish Oil Pan Seal and Main Bearing Blocks

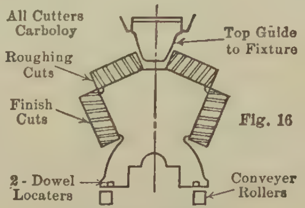


Fig. 16

3rd Operation
Rough Head Contact Face and Finish Water Back

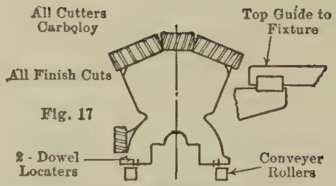


Fig. 17

4th Operation
Finish Head Contact, Oil Seal, Oil Relief

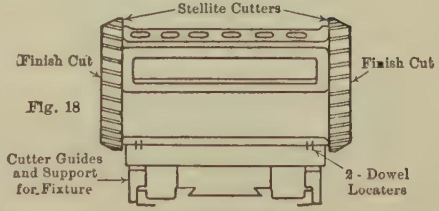


Fig. 18

5th Operation
Mill Ends

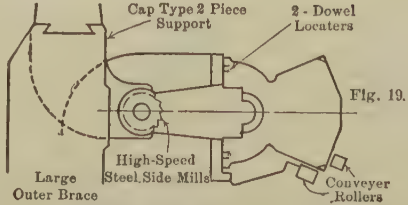


Fig. 19.

6th Operation
Straddle Mill Main Bearings

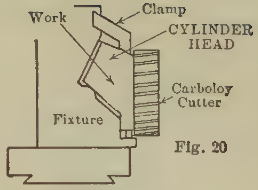


Fig. 20

1st Operation
Mill Bottom

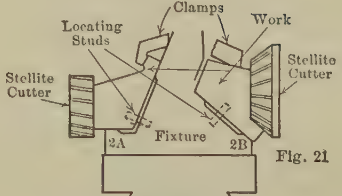


Fig. 21

2nd Operation
Mill Manifold. Valve Cover Face and Bosses

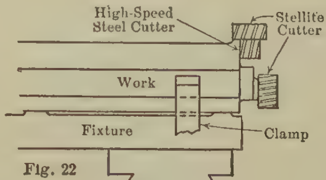


Fig. 22

3rd Operation
Mill End and Seal Pad

FIGS. 15 to 22

are selected for that class of production milling which involves relatively small parts, manufactured in limited quantities, with frequent set-ups. They also are used in tool room work and in jobbing shops.

These machines can be equipped with fixtures and multi-spindle attachments to permit their use as production units. Such procedure pays in many shops, where certain jobs are manufactured in sufficient quantities to justify a knee and column type machine for high production, with the idea that it easily can be stripped of its high-production elements and put back on regular run of work.

Knee and column machines are subdivided as follows:

*a*. *Plain Horizontal*, with spindle located horizontally.

*b*. *Universal*, equipped with a swiveling table, for the cutting of helical grooves.

*c*. *Vertical*, with the spindle located vertically.

*d*. *Automatic*, which combines the advantages of the bed type and the knee and column type millers.

**2. Hand Millers** usually are small machines of the knee and column type, in which the feeding movement of the table is controlled by hand. They are used in the manufacture of comparatively small quantities of light parts. Hand millers are subdivided as follows: *a*. Floor type. *b*. Bench type. The latter are used for the machining operations of very light work.

**3. Fixed Bed Type** machines are intended for use in the manufacture of parts in large quantities, and for this reason they are more rigidly built and simpler than the knee and column type millers. These machines are built on the plan of standard unit construction, which permits the make-up, from standard and complementary units, of a great variety of spindle combinations and feed cycles to meet the specific requirements of manufacture of a given product. See Figs. 15 to 22.

Fixed bed machines often are called semi-automatic, because, with proper fixtures,

**Table 14.—Milling Operations Performed on Cylinder Blocks and Heads of a Twelve-cylinder Motor**

| Operation | Pieces per Hour | Cutters | | | | R.p.m. | Cutting Speed, ft. per min. | Feed, in. per min. | Chip per Tooth, in. | Stock Removed, in. |
|---|---|---|---|---|---|---|---|---|---|---|
| | | No. | Type | Diam., in. | Material | | | | | |
| **Cylinder Blocks** | | | | | | | | | | |
| Mill oil-pan face, slot for bearing caps (Fig. 15) | 16 | 2 | Inserted-blade face mills | 6 1/2 | Tungsten Carbide | 107 | 183 | 19 | 0.0089 | 1/8 |
| | | 1 | Inserted-blade end-mill | 6 3/4 | J-Stellite | 64 | 113 | 10 | 0.0098 | 1/8 to 3/16 |
| Rough-mill cylinder head seats and finish-mill valve cover faces (Fig. 16) | 18 | 2 | Inserted-blade face mills | 7 | Tungsten Carbide | 111 | 204 | 12 3/4 | 0.0058 | 1/8 |
| | | 1 | Inserted-blade face mills | 5 1/2 | Tungsten Carbide | 155 | 223 | 12 3/4 | 0.0052 | 1/8 |
| Finish-mill cylinder head seats and mill top and water-jacket pad (Fig. 17) | 18 | 2 | Inserted-blade cutters | 7 | Tungsten Carbide | 111 | 204 | 14 5/16 | 0.0064 | 1/32 |
| | | 2 | Inserted-blade cutters | 4 | Tungsten Carbide | 189 | 199 | 14 5/16 | 0.0076 | 1/8 |
| Mill cylinder block ends (Fig. 18) | 13 | 2 | Inserted-blade face mills | 18 | J-Stellite | 31 | 146 | 7 | 0.0057 | 1/8 |
| Straddle-mill main bearing sides (Fig. 19) | 52 | 6 | Half side-mills | 6 | High-speed steel | 49 | 77 | 5 | 0.0085 | 1/8 |
| **Cylinder Heads** | | | | | | | | | | |
| Mill cylinder head bottom (Fig. 20) | 24 | 1 | Inserted-blade face mill | 7 | Tungsten Carbide | 110 | 202 | 15 | 0.0069 | 3/32 |
| Mill manifold, valve cover face and hold-down bosses (Fig. 21) | 14 | 1 | Inserted-blade shell end-mill | 4 1/2 | Stellite | 111 | 131 | 10 1/2 | 0.0086 | 1/8 |
| | | 1 | Inserted-blade formed mill | 6 | Stellite | 86 | 120 | 10 1/2 | 0.0100 | 1/8 |
| Mill ends and seal pad (Fig. 22) | 57 | 2 | Solid shell end-mills | 4 | Stellite | 111 | 117 | 8 | 0.0036 | 1/8 |
| | | 1 | Shell end-mill | 2 1/4 | High-speed steel | 111 | 66 | 8 | 0.0072 | 1/8 |

the work may be clamped and the machine started through a definite automatic cycle, such as rapid advance of work to cutter, cutting feed, rapid return to the starting position, and stop. These machines are subdivided as follows: *a.* Plain (single spindle); *b.* Duplex; *c.* Multiple spindle; rail; bridge; special types.

4. **Rotary and Drum Type Millers** are provided with fixtures and are used for mass production work in which the cutting is continuous, with no idle time. The parts are loaded on one side of the machine, pass first the roughing, and then the finishing cutters, and return to the loading position, where they are replaced. Drum type machines derive their name from the fact that the parts to be milled are mounted on a drum-like fixture which rotates continuously.

5. **Planetary Milling Machines** are used to mill plain or formed internal surfaces. The cutter is inserted in the bore to be milled and, while rotating, is fed to depth radially, makes a single sweeping cut about the bore, and is withdrawn first radially, then axially.

6. **Thread Milling Machines** are designed to cut threads and worms. In these operations milling cutters are used rather than single-point cutting tools.

7. **Die-sinking and Profiling Machines** are designed to reproduce accurately forms and contours from a master by means of a tracer, contact with which maintains the master form throughout the operation, thus controlling the relative position of work and cutter. These machines are subdivided into:

*a.* Hand controlled, in which the tracer is guided by hand over the master form.

*b.* Automatic, where the tracer, once the cycle is started, follows automatically the master form without requiring the assistance of the operator.

**MILLING MACHINE ATTACHMENTS.**—The range of work that a milling machine can do is greatly increased by the use of attachments. An almost endless variety of attachments has been devised for special requirements. Many of these attachments, however, have become standard equipment units of the milling machine. Among these the most important are the following: 1. Universal dividing head. 2. Index head. 3. Heavy vertical milling attachment. 4. Circular milling attachment. 5. Universal spiral milling attachment. 6. Slotting attachment. 7. Rack milling attachment. 8. Standard index base. 9. Cam milling attachment. 10. Toolmaker's vise.

# DRILLS AND DRILLING*

## 1. DRILLING PRACTICE

**CONSTANT FOR FINDING SPEEDS OF DRILLS.** (See Table 1.)—For finding the speed in feet when the number of revolutions is given, or the number of revolutions, when the speed in feet is given.

Constant = 12 ÷ (size of drill × 3.1416).
Number of revolutions = Constant × speed in feet.
Speed in feet = Number of revolutions ÷ constant.

**Table 1.—Constants for Drill Speeds**

| Size of Drill, in. | Constant | Size of Drill, in. | Constant | Size of Drill, in. | Constant | Size of Drill, in. | Constant | Size of Drill, in. | Constant |
|---|---|---|---|---|---|---|---|---|---|
| 1/8 | 30.55 | 3/4 | 5.09 | 1 3/8 | 2.78 | 2 | 1.91 | 2 5/8 | 1.45 |
| 3/16 | 20.38 | 13/16 | 4.70 | 1 7/16 | 2.66 | 2 1/16 | 1.85 | 2 11/16 | 1.42 |
| 1/4 | 15.28 | 7/8 | 4.36 | 1 1/2 | 2.55 | 2 1/8 | 1.80 | 2 3/4 | 1.39 |
| 5/16 | 12.22 | 15/16 | 4.07 | 1 9/16 | 2.44 | 2 3/16 | 1.75 | 2 13/16 | 1.36 |
| 3/8 | 10.19 | 1 | 3.82 | 1 5/8 | 2.35 | 2 1/4 | 1.70 | 2 7/8 | 1.33 |
| 7/16 | 8.73 | 1 1/16 | 3.59 | 1 11/16 | 2.26 | 2 5/16 | 1.65 | 2 15/16 | 1.30 |
| 1/2 | 7.64 | 1 1/8 | 3.39 | 1 3/4 | 2.18 | 2 3/8 | 1.61 | 3 | 1.27 |
| 9/16 | 6.79 | 1 3/16 | 3.22 | 1 13/16 | 2.11 | 2 7/16 | 1.57 | 3 1/16 | 1.25 |
| 5/8 | 6.11 | 1 1/4 | 3.06 | 1 7/8 | 2.04 | 2 1/2 | 1.53 | 3 1/8 | 1.22 |
| 11/16 | 5.56 | 1 5/16 | 2.91 | 1 15/16 | 1.97 | 2 9/16 | 1.49 | 3 1/4 | 1.18 |

**DRILL SPEEDS.**—The correct speed of drill will vary with the composition and hardness of the material, type of machine used, depth of hole, lubricant, and other conditions. The speed and feed to be used for any particular job is largely a matter of judgment and trial. The Cleveland Twist Drill Co. recommends the speeds given in Table 2 as a basis for good practice with high-speed steel drills. Speed of carbon steel drills should be from 40 to 50% of the speed of high-speed steel drills.

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* Staff contribution.

While an occasional drill will withstand for days a cutting speed of 150 ft. per minute, in either cast iron or steel (the latter under a lubricant), it rarely is expedient to drive any but very small ones faster than 100 ft. per min. Operating drills at an excessive speed usually is expensive, and it is more economical to err in the other direction. Table 3 gives the speeds and feeds recommended by the Cincinnati-Bickford Tool Co. as representative of good commercial drilling practice. The table is based on cutting speeds of 100 ft. per min. in cast iron, 80 ft. per min. in medium steel, and 40 to 60 ft. per min. in unusually hard steel.

**DRILL FEEDS.**—The rate at which a drill may be advanced per revolution depends on the toughness of the material to be drilled, the ability of the machine to resist thrust without forfeiture of alignment and upon the knowledge that is exercised in the grinding of the drill—the size of its included angle, the width of its chisel point, and the keenness and evenness of its cutting edges, all being deciding factors. Were it not for the weakening effect upon the drill it could be said that the stiffer the machine, the less the included angle, the narrower the chisel point, the smaller the degree of the spiral, the greater the uniformity of the cutting lips and the more efficacious the lubricant in minimizing the frictional resistance of the chips, the coarser becomes the feed it is permissible to use. But, inasmuch as the durability of the drill must not be impaired, the advantage obtainable through the application of these axioms has its limitations.

The size of the drill and the material drilled govern, in practice, the feed used. A general rule is a feed of 0.001 to 0.002 in. per revolution for drills smaller than 1/8 in., 0.002 to 0.004 in. per revolution for drills 1/8 to 1/4 in., 0.004 to 0.007 in. per revolution for drills 1/4 to 1/2 in., 0.007 to 0.015 in. per revolution for drills 1/2 to 1 in., and 0.015 to

**Table 2.—Drill Speeds for Various Materials**

| Material | Speed, ft. per min. | Material | Speed, ft. per min. |
|---|---|---|---|
| Mild machinery steel, 0.2–0.3 carbon | 80–110 | Malleable iron | 80– 90 |
| Steel 0.4–0.5 carbon | 70– 80 | Ordinary brass and bronze | 200–300 |
| Tool steel 1.2 carbon | 50– 60 | High tensile bronze | 70–150 |
| Steel forgings | 50– 60 | Monel metal | 40– 50 |
| Alloy steel | 50– 70 | Slate, marble and stone | 15– 25 |
| Stainless steel | 30– 40 | Aluminum and its alloys | 200–300 |
| Soft cast iron | 100–150 | Magnesium and its alloys | 250–400 |
| Hard chilled cast iron | 70–100 | Bakelite | 100–150 |
| | | Wood | 300–400 |

**Table 3.—Drilling Feeds for Cast Iron and Steel**

(Cincinnati-Bickford Tool Co., Cincinnati)

| Drill Size, in. | Cast Iron, Average Quality | | | Medium Steel | | | Unusually Tough Steel | | |
|---|---|---|---|---|---|---|---|---|---|
| | R.p.m. | Feed | | R.p.m. | Feed | | R.p.m. | Feed | |
| | | In. per rev. | In. per min. | | In. per rev. | In. per min. | | In. per rev. | In. per min. |
| 3/16 | 2000 | 0.004–.006 | 8–12 | 1600 | 0.003–.008 | 4 3/4–13 | 800–1000 | 0.002–.004 | 2–4 |
| 1/4 | 1500 | .008–.012 | 12–16 | 1200 | .006–.012 | 7–14 | 600–900 | .004–.008 | 3–6 |
| 3/8 | 1000 | .012–.020 | 12–20 | 800 | .008–.012 | 6–9 | 400–600 | .006–.008 | 2 1/2–5 |
| 1/2 | 750 | .015–.025 | 11–19 | 600 | .010–.015 | 6–9 | 300–460 | .008–.010 | 2–4 1/2 |
| 5/8 | 600 | .015–.025 | 9–15 | 500 | .010–.015 | 5–8 | 220–360 | .008–.010 | 2–3 1/2 |
| 3/4 | 500 | .018–.030 | 9–15 | 400 | .010–.020 | 4–7 | 200–300 | .010–.012 | 1 3/4–3 1/2 |
| 7/8 | 440 | .018–.030 | 8–13 | 350 | .010–.020 | 3–7 | 180–260 | .010–.012 | 1 3/4–3 1/4 |
| 1 | 380 | .018–.030 | 7–11 | 300 | .010–.020 | 3–6 | 150–225 | .010–.015 | 1 1/2–3 |
| 1 1/8 | 340 | .020–.030 | 7–11 | 270 | .015–.020 | 4–5 1/2 | 140–200 | .012–.020 | 1 1/2–4 |
| 1 1/4 | 300 | .020–.030 | 6–9 | 240 | .015–.020 | 3 1/2–5 | 120–180 | .012–.020 | 1 1/2–3 1/2 |
| 1 3/8 | 275 | .020–.030 | 5 1/2–8 | 220 | .015–.020 | 3–4 1/2 | 110–165 | .012–.020 | 1 1/4–3 1/4 |
| 1 1/2 | 250 | .020–.035 | 5–8 | 200 | .015–.020 | 3–4 | 100–150 | .012–.020 | 1 1/4–3 |
| 1 5/8 | 235 | .020–.035 | 5–8 | 185 | .015–.020 | 2 3/4–3 3/4 | 95–140 | .012–.020 | 1–2 3/4 |
| 1 3/4 | 220 | .020–.035 | 4 1/2–7 1/2 | 170 | .015–.020 | 2 1/2–3 1/2 | 90–130 | .012–.020 | 1–2 1/2 |
| 1 7/8 | 205 | .020–.035 | 4–7 | 160 | .015–.020 | 2 1/2–3 | 80–120 | .012–.020 | 1–2 1/4 |
| 2 | 190 | .020–.035 | 4–7 | 150 | .015–.020 | 2 1/4–3 | 75–114 | .012–.020 | 1–2 1/4 |
| 2 1/8 | 180 | .025–.040 | 4 3/4–7 | 140 | .020–.030 | 2 3/4–4 | 70–108 | .012–.020 | 3/4–2 |
| 2 1/4 | 170 | .025–.040 | 4 3/4–7 | 130 | .020–.030 | 2 1/2–4 | 65–102 | .012–.020 | 3/4–2 |
| 2 3/8 | 165 | .025–.040 | 4–6 1/2 | 120 | .020–.035 | 2 1/2–4 | 60–95 | .012–.020 | 3/4–2 |
| 2 1/2 | 155 | .025–.050 | 3 3/4–7 1/2 | 115 | .020–.035 | 2 1/4–4 | 60–90 | .012–.020 | 3/4–1 3/4 |
| 2 5/8 | 145 | .030–.050 | 4–7 1/2 | 110 | .020–.035 | 2 1/4–3 3/4 | 55–85 | .012–.020 | 3/4–1 3/4 |
| 2 3/4 | 140 | .030–.050 | 4 1/2–7 | 108 | .020–.035 | 2–3 3/4 | 55–80 | .012–.020 | 3/4–1 3/4 |
| 2 7/8 | 130 | .030–.050 | 3 3/4–6 1/4 | 104 | .020–.035 | 2–3 1/2 | 52–77 | .012–.020 | 3/4–1 1/2 |
| 3 | 125 | .030–.050 | 3 3/4–6 1/4 | 100 | .020–.035 | 2–3 1/2 | 50–75 | .012–.020 | 3/4–1 1/2 |

0.025 in. per revolution for drills larger than 1 in. These figures apply to medium steel. Alloy steel and hard steel usually require lighter feeds than the foregoing while cast iron, brass and aluminum may be drilled with heavier feeds. See Table 3.

**DRILLING LUBRICANTS.**—To obtain the maximum performance from drills operating in any material except cast iron, a copious supply of cutting lubricant at the cutting edges is necessary. Cast iron should be drilled dry, or with a jet of compressed air directed into the hole to act as a cooling medium.* The following lubricants are recommended by the Cleveland Twist Drill Co. for cutting other materials, the preference being in the order given:

**Hard and Refractory Steel.**—Turpentine, kerosene, and soluble oil.

**Soft Steel.**—Lard oil, soluble oil.

**Wrought Iron.**—Lard oil, soluble oil.

**Malleable Iron.**—Soluble oil.

**Brass.**—Paraffin oil, or dry.

**Aluminum and Soft Alloys.**—Kerosene, soluble oil.

See also Boston and Oxford's experiments on cutting fluids, p. 21–51.

**PROPORTIONS OF TWIST DRILLS.**—Table 4 shows the dimensions of twist drills as given by standard B 5.0–1936 of the A.S.A. Fig. 1 illustrates nomenclature adopted by the A.S.A. in the same standard for twist drills. This standard covers only straight shank drills. The standards of various manufacturers differ in regard to taper shank drills. For dimensions of the various drill tapers in common use, see pp. 12–17 to 12–23.

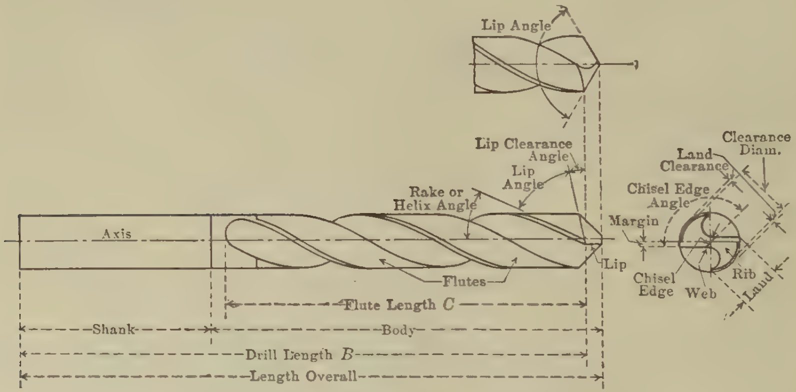

FIG. 1. Standard Straight Shank Twist Drill

For maximum efficiency it is essential that the point of the drill be properly ground. For iron and steel, experience indicates that a lip angle of 108 deg., *i.e.*, each lip making an angle of 59 deg. with the axis of the drill, gives the most efficient performance. The best lip clearance angle seems to be 12 deg. at the periphery of the drill, increasing gradually as the center of the drill is approached. The chisel-edge angle should be approximately 135 deg. Insufficient lip clearance angle at the center of the drill is the principal cause of drills splitting in the web. It also is essential that the cutting edges be of equal length.

**TWIST DRILL AND STEEL WIRE GAGES.**—Three standards of gages for twist drills and steel wire are in use—the Manufacturers' Standard, used by the Morse Twist Drill Co., Brown & Sharpe, and other manufacturers, the Stubs gage, and that of the Standard Tool Co. The Stubs and Manufacturers' gages are given in the table on page 6–25. The Standard Tool Co. gage agrees with the Manufacturers' gage for sizes from Nos. 1 to 60, inclusive, and with the Stubs gage for sizes from Nos. 61 to 80. In addition it has additional $^1/_2$ sizes interpolated at Nos. 60 $^1/_2$, 68 $^1/_2$, 69 $^1/_2$, 71 $^1/_2$, 73 $^1/_2$, 74 $^1/_2$, 78 $^1/_2$, and 79 $^1/_2$.

**DRILL GRINDING.**—Improper performance of drills, drill breakage and excessive power required for drilling usually can be traced to improper grinding of the drill. Correct grinding of drills by hand is almost impossible, and machine grinding will pay for itself even in the smallest of machine shops. It is essential not only that the lips be ground at the proper angle, but also that they be of equal length and that clearance should be properly ground. The web of the drill should be thinned down as the drill is worn away in order to provide the proper thickness of web and the correct angle and thickness of the chisel point. Drill grinding and point thinning machines are made by Wm. Sellers & Co., Philadelphia.

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* Boston and Oxford's experiments indicate that a cutting fluid will improve performance in drilling cast iron. See p. 21–51.

**POWER REQUIRED TO DRIVE DRILLS.**—The Cincinnati-Bickford Tool Co. gives the following formulas for the horsepower required for drilling cast iron and steel with high-speed steel drills. In these formulas $D$ = diam. of drill, in.; $F$ = feed, in. per revolution; $N$ = rev. per min.

Cast iron Hp. = $0.404\,D^2FN$ . . . . . . . . . . . [1]

Mild steel Hp. = $0.98\,D^2FN$ . . . . . . . . . . [2]

These formulas are based on the use of properly-ground drills, and on the use of a soda-water coolant when drilling steel. Under other conditions the horsepower will considerably exceed the values given by the formulas.

Table 5 gives the results of tests made by the American Tool Works Co., Cincinnati,

**Table 4.—Standard Sizes of Straight Shank Twist Drills**

A.S.A. Std, B 5.0 – 1936. See Fig. 1

| | Regular Series | | | Regular Series | | Long Series | | | Regular Series | | Long Series | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diameter, *A*, in. | Drill Length, *B*, in. | Flute Length, *C*, in. | Diameter, *A*, in. | Drill Length, *B*, in. | Flute Length, *C*, in. | Drill Length, *B*, in. | Flute Length, *C*, in. | Diameter, *A*, in. | Drill Length, *B*, in. | Flute Length, *C*, in. | Drill Length, *B*, in. | Flute Length, *C*, in. |
| 0.0156 | 7/8 | 3/16 | 0.0995 | 2 7/16 | 1 3/8 | ...... | ...... | 0.2344 | 3 7/8 | 2 5/8 | 6 1/8 | 3 3/4 |
| .0180 | 15/16 | 3/16 | .1024 | 2 7/16 | 1 7/16 | ...... | ...... | .2402 | 3 7/8 | 2 11/16 | ...... | ...... |
| .0200 | 1 | 1/4 | .1040 | 2 1/2 | 1 3/8 | ...... | ...... | .2460 | 4 | 2 3/4 | ...... | ...... |
| .0225 | 1 1/8 | 1/4 | .1063 | 2 9/16 | 1 1/2 | ...... | ...... | .2500 | 4 1/2 | 3 1/8 | 6 1/2 | 4 5/16 |
| .0240 | 1 3/16 | 5/16 | .1094 | 2 7/8 | 1 5/8 | ...... | ...... | .2520 | 4 1/2 | 3 1/8 | ...... | ...... |
| .0260 | 1 5/16 | 7/16 | .1130 | 2 11/16 | 1 5/8 | ...... | ...... | .2570 | 4 1/2 | 3 1/8 | ...... | ...... |
| .0280 | 1 5/16 | 7/16 | .1160 | 2 11/16 | 1 5/8 | ...... | ...... | .2610 | 4 1/2 | 3 1/8 | ...... | ...... |
| .0295 | 1 3/8 | 7/16 | .1200 | 2 3/4 | 1 5/8 | ...... | ...... | .2656 | 4 1/2 | 3 1/8 | 6 1/2 | 4 5/16 |
| .0312 | 1 1/2 | 9/16 | .1220 | 2 3/4 | 1 5/8 | ...... | ...... | .2720 | 4 1/2 | 3 1/8 | ...... | ...... |
| .0330 | 1 1/2 | 5/8 | .1250 | 3 | 1 3/4 | 5 1/8 | 3 | .2770 | 4 1/2 | 3 1/8 | ...... | ...... |
| .0350 | 1 1/2 | 5/8 | .1285 | 2 13/16 | 1 3/4 | ...... | ...... | .2812 | 4 3/4 | 3 3/8 | 6 3/4 | 4 1/2 |
| .0370 | 1 1/2 | 5/8 | .1299 | 2 13/16 | 1 11/16 | ...... | ...... | .2854 | 4 3/4 | 3 3/8 | ...... | ...... |
| .0390 | 1 1/2 | 5/8 | .1339 | 2 13/16 | 1 11/16 | ...... | ...... | .2913 | 4 3/4 | 3 3/8 | ...... | ...... |
| .0410 | 1 9/16 | 11/16 | .1360 | 2 13/16 | 1 3/4 | ...... | ...... | .2969 | 4 3/4 | 3 3/8 | 6 3/4 | 4 1/2 |
| .0430 | 1 11/16 | 3/4 | .1378 | 2 13/16 | 1 11/16 | ...... | ...... | .3020 | 4 3/4 | 3 3/8 | ...... | ...... |
| .0453 | 1 11/16 | 3/4 | .1406 | 3 1/8 | 1 7/8 | 5 3/8 | 3 1/4 | .3071 | 4 3/4 | 3 3/8 | ...... | ...... |
| .0469 | 1 3/4 | 3/4 | .1440 | 2 15/16 | 1 7/8 | ...... | ...... | .3125 | 5 | 3 5/8 | 7 | 4 11/16 |
| .0492 | 1 11/16 | 3/4 | .1470 | 2 15/16 | 1 7/8 | ...... | ...... | .3160 | 5 | 3 5/8 | ...... | ...... |
| .0512 | 1 3/4 | 3/4 | .1520 | 3 1/16 | 1 7/8 | ...... | ...... | .3230 | 5 | 3 5/8 | ...... | ...... |
| .0531 | 1 3/4 | 7/8 | .1562 | 3 1/4 | 2 | 5 3/8 | 3 1/4 | .3281 | 5 | 3 5/8 | 7 | 4 11/16 |
| .0550 | 1 13/16 | 7/8 | .1610 | 3 13/16 | 2 | ...... | ...... | .3320 | 5 | 3 5/8 | ...... | ...... |
| .0571 | 1 13/16 | 7/8 | .1660 | 3 1/4 | 2 | ...... | ...... | .3390 | 5 | 3 5/8 | ...... | ...... |
| .0591 | 1 13/16 | 7/8 | .1695 | 3 5/16 | 2 1/8 | ...... | ...... | .3437 | 5 1/4 | 3 7/8 | 7 1/4 | 4 15/16 |
| .0610 | 1 13/16 | 7/8 | .1719 | 3 3/8 | 2 1/8 | 5 3/4 | 3 1/2 | .3480 | 5 1/4 | 3 7/8 | ...... | ...... |
| .0615 | 2 1/2 | 1 1/4 | .1730 | 3 5/16 | 2 1/8 | ...... | ...... | .3543 | 5 1/4 | 3 7/8 | ...... | ...... |
| .0629 | 1 13/16 | 7/8 | .1770 | 3 3/8 | 2 1/8 | ...... | ...... | .3594 | 5 1/4 | 3 7/8 | 7 1/4 | 4 15/16 |
| .0650 | 1 13/16 | 15/16 | .1800 | 3 7/16 | 2 1/4 | ...... | ...... | .3680 | 5 1/4 | 3 7/8 | ...... | ...... |
| .0670 | 1 15/16 | 15/16 | .1850 | 3 1/2 | 2 1/4 | ...... | ...... | .3750 | 5 5/8 | 4 1/8 | 7 5/8 | 5 1/8 |
| .0700 | 1 15/16 | 15/16 | .1875 | 3 1/2 | 2 1/4 | 5 3/4 | 3 1/2 | .3860 | 5 5/8 | 4 1/8 | ...... | ...... |
| .0730 | 2 | 15/16 | .1910 | 3 9/16 | 2 1/4 | ...... | ...... | .3906 | 5 5/8 | 4 1/8 | 7 5/8 | 5 1/8 |
| .0760 | 2 1/16 | 1 1/16 | .1935 | 3 5/8 | 2 3/8 | ...... | ...... | .3970 | 5 5/8 | 4 1/8 | ...... | ...... |
| .0781 | 2 5/8 | 1 3/8 | .1960 | 3 11/16 | 2 3/8 | ...... | ...... | .4062 | 5 5/8 | 4 1/8 | 7 5/8 | 5 1/8 |
| .0810 | 2 1/8 | 1 1/8 | .1990 | 3 11/16 | 2 3/8 | ...... | ...... | .4219 | 5 5/8 | 4 1/8 | 7 5/8 | 5 1/8 |
| .0827 | 2 3/16 | 1 3/16 | .2031 | 3 5/8 | 2 3/8 | 6 | 3 5/8 | .4375 | 6 1/8 | 4 7/16 | 8 1/8 | 5 7/16 |
| .0860 | 2 3/16 | 1 1/8 | .2090 | 3 7/8 | 2 1/2 | ...... | ...... | .4531 | 6 1/8 | 4 7/16 | 8 1/8 | 5 7/16 |
| .0890 | 2 1/4 | 1 1/8 | .2130 | 3 15/16 | 2 5/8 | ...... | ...... | .4687 | 6 1/8 | 4 7/16 | 8 1/8 | 5 7/16 |
| .0906 | 2 1/4 | 1 1/4 | .2187 | 3 3/4 | 2 1/2 | 6 | 3 5/8 | .4844 | 6 1/8 | 4 7/16 | 8 1/8 | 5 7/16 |
| .0937 | 2 3/4 | 1 1/2 | .2244 | 3 7/8 | 2 11/16 | ...... | ...... | .5000 | 6 1/8 | 4 7/16 | 8 1/8 | 5 7/16 |
| .0960 | 2 5/16 | 1 1/4 | .2280 | 4 | 2 3/4 | ...... | ...... | ...... | ...... | ...... | ...... | ...... |

**Table 5.—Power Required to Drive Drills**

American Tool Works Co., Cincinnati

| Drill Diam., in. | R.p.m. | Cutting Speed, ft. per min. | Feed per Rev., in. | Penetration per Minute, in. | Horsepower | R.p.m. | Cutting Speed, ft. per min. | Feed per Rev., in. | Penetration per Minute, in. | Horsepower |
|---|---|---|---|---|---|---|---|---|---|---|
| | Drilling Cast Iron | | | | | Drilling Cast Steel | | | | |
| 3/4 | 550 | 108 | 0.016 | 8.800 | 10 1/2 | 380 | 75 | 0.029 | 11.020 | 12 1/2 |
| 3/4 | 550 | 108 | .022 | 12.100 | 11 1/2 | 380 | 75 | .022 | 8.360 | 11 |
| 3/4 | 550 | 108 | .029 | 15.950 | 12 1/2 | 380 | 75 | .016 | 6.080 | 7 |
| 3/4 | 550 | 108 | .040 | 22.000 | 13 | | | | | |
| 1 | 550 | 144 | .022 | 12.100 | 11 1/2 | 280 | 73 | .022 | 6.160 | 8 1/2 |
| 1 | 550 | 144 | .029 | 15.950 | 12 1/2 | 280 | 73 | .029 | 8.120 | 12 |
| 1 | 550 | 144 | .040 | 22.000 | 14 | | | | | |
| 1 1/4 | 380 | 125 | .022 | 7.600 | 9 1/2 | 240 | 79 | .022 | 5.280 | 10 1/2 |
| 1 1/4 | 380 | 125 | .029 | 11.020 | 10 1/2 | 240 | 79 | .029 | 6.960 | 12 1/2 |
| 1 1/4 | 380 | 125 | .040 | 15.200 | 13 | 240 | 79 | .040 | 9.600 | 18 1/2 |
| 1 1/2 | 280 | 110 | .022 | 6.160 | 8 1/2 | 192 | 75 | .022 | 4.224 | 14 |
| 1 1/2 | 280 | 110 | .029 | 8.120 | 10 | 192 | 75 | .029 | 5.568 | 17 1/2 |
| 1 1/2 | 280 | 110 | .040 | 11.200 | 12 1/2 | 192 | 75 | .040 | 7.680 | 23 |
| 2 | 240 | 125 | .022 | 5.280 | 10 | 133 | 70 | .022 | 2.926 | 10 1/2 |
| 2 | 240 | 125 | .029 | 6.960 | 11 1/2 | 133 | 70 | .029 | 3.857 | 14 |
| 2 | 240 | 125 | .040 | 9.600 | 14 | 133 | 70 | .040 | 5.320 | 18 1/2 |
| 2 1/2 | 227 | 148 | .022 | 4.994 | 16 | 116 | 76 | .022 | 2.552 | 14 1/2 |
| 2 1/2 | 227 | 148 | .029 | 6.583 | 18 1/2 | 116 | 76 | .029 | 3.364 | 20 |
| 2 1/2 | 227 | 148 | .040 | 9.080 | 22 | 116 | 76 | .040 | 4.640 | 28 |
| 3 | 158 | 124 | .022 | 3.476 | 14 | 99 | 78 | .016 | 1.584 | 15 1/2 |
| 3 | 158 | 124 | .029 | 4.582 | 16 | 99 | 78 | .022 | 2.178 | 21 |
| 3 | 158 | 124 | .040 | 6.320 | 20 | 99 | 78 | .029 | 2.871 | 27 |

on drilling cast iron and steel, using a plain radial drill with a 6-ft. arm. The American Tool Works Co. comments on these tests as follows:

> The tabulated cutting speeds for cast iron are in most instances somewhat higher than would be recommended for present day high-speed steel drills. In most cases, especially when drilling through scale, the use of cutting speeds above 100 ft. to 110 ft. per min. are excessive, and necessitate frequent regrinding of the drill. However, when drilling finished surfaces in the better grades of iron, cutting speeds up to 150 ft. per min. often are practicable.
>
> For drilling the average run of commercial cast steel, we have found that cutting speeds in excess of 80 ft. per min. quickly dull the cutting edge of the tool, especially if the material has hard spots. For this class of material, cutting speeds for 70 ft. to 80 ft. per min. will cover average conditions.

## 2. EXPERIMENTS ON DRILLING

**FORCES ACTING ON TWIST DRILLS.**—An extensive series of experiments on the forces acting on twist drills of high-speed steel when operating on cast iron and steel is reported by Dempster Smith and A. Poliakoff, in *Proc.* Inst. M. E., 1909. Abstracted in *Am. Mach.*, May, 1909. Approximate equations derived from the first set of experiments are as follows:

Torque, lb.-ft., $T = (1800\,t + 9)d^2$ for medium cast iron . . . . . . . . . . [3]

$T = (3200\,t + 20)d^2$, for medium steel . . . . . . . . . . . [4]

End thrust, lb. $P = 115{,}000\,t - 200$ for medium cast iron . . . . . . . . . . [5]

$P = 160{,}000\,(d - 0.5)t - 1000$ for medium steel . . . . . . . . [6]

$d$ = diam., $t$ = feed per revolution of drill, both in inches. The steel was of medium hardness, 0.29 C, 0.625 Mn.

A second series of experiments with soft cast iron of combined C., 0.2; graphitic C., 2.9; Si, 1.41; Mn, 0.68; S, 0.035; P, 1.48, and medium steel of C, 0.31; Si, 0.07; Mn, 0.50; S, 0.018; P, 0.033; tensile strength, 72,600 lb. per sq. in., gave results from which were derived the following approximate equations:

Torque, lb.-ft., $T = 740\,d^{1.8}\,t^{0.7}$, or $10\,d^2 + 100\,t(14\,d^2 + 3)$ for cast iron . . . [7]

$T = 1640\,d^{1.8}\,t^{0.7}$, or $28\,d^2(1 + 100\,t)$ for medium steel . . . . . [8]

End thrust, lb., $P = 35{,}500\,d^{0.7}\,t^{0.75}$, or $200\,d + 10{,}000\,t$ for cast iron . . . . . . [9]

$P = 35{,}500\,d^{0.7}\,t^{0.6}$, or $750\,d + 1000\,t(75\,d + 50)$ for medium steel, [10]

and for different sizes of drill the equations given in Table 6.

The tests were made without lubricants. When lubricants were used in drilling steel the average torque varied from 72%, with 1/400 in. feed, to 92%, with 1/35 in. feed, of that obtained when operating dry. The thrust for soft, medium and hard steel is 26%, 37% and 12%, respectively, less than when operating dry, no marked difference being found, as in the torque, with different feeds. The horsepower varies as $t^{0.7}$ and as $d^{0.8}$ for a given drill and speed. The torque and horsepower when drilling medium steel is about 2.1 times that required for cast iron with the same drill speed and feed. The horsepower per cu. in. of metal removed is inversely proportional to $d^{0.2}\,t^{0.3}$, and is independent of the revolutions.

While the chisel point of the drill scarcely affects the torque it is accountable for about 20% of the thrust. Tests made with a preliminary hole drilled before the main drill was used to enlarge the hole showed that the work required to drill a hole where only one drill is used is greater than that required to drill the hole in two operations, with drills of different diameter.

**Table 6.—Equations for Various Sizes of Drill**

| Drill | 3/4 | 1 | 1 1/2 |
|---|---|---|---|
| Cast iron $T$ = | 5 + 1,100 $t$ | 10 + 1.750 $t$ | 25 + 3,700 $t$ |
| Cast iron $P$ = | 125 + 82,000 $t$ | 200 + 89,000 $t$ | 350 + 103,000 $t$ |
| Steel $T$ = | 7.5 + 3,350 $t$ | 17.5 + 4,400 $t$ | 40 + 9,000 $t$ |
| Steel $P$ = | 550 + 109,000 $t$ | 750 + 131,000 $t$ | 1,250 + 162,000 $t$ |
| Drill | 2 | 2 1/2 | 3 |
| Cast iron $T$ = | 40 + 5,900 $t$ | 60 + 8,800 $t$ | 90 + 12,900 $t$ |
| Cast iron $P$ = | 500 + 110,000 $t$ | 600 + 126,000 $t$ | 850 + 140,000 $t$ |
| Steel $T$ = | 75 + 12,500 $t$ | 112.5 + 19,050 $t$ | 175 + 26,250 $t$ |
| Steel $P$ = | 1,500 + 181,250 $t$ | 1,725 + 224,375 $t$ | 2,350 + 280,000 $t$ |

**TORQUE AND THRUST IN RELATION TO CUTTING FLUID.**—A research to determine the performance of cutting fluids in drilling is reported by O. W. Boston and C. J. Oxford in *Trans.* A.S.M.E., RP 55–1, 1933 and RP 58–2, 1936. The former relates to drills ranging from 1/2 in. to 1 1/2 in. diameter, and the latter drills from 1/8 in. to 1/2 in. diameter. The materials on which the research was made were as follows: Chilled cast aluminum alloy S.A.E. 33, Brinell hardness No. 70; leaded free-cutting brass, extruded and drawn, of analysis Cu, 61.75, Zn, 35.0, Pb, 3.21, Fe, 0.04, Brinell hardness No. 179; cast iron containing 3.44 C, 2.62 Si, 0.986 Ni, and 0.478 Mn, Brinell hardness No. 179; malleable cast iron containing 2.46 C, 0.26 Mn, 0.163 P, 0.071 S, and 1.00 Si, Brinell hardness No. 137; S.A.E. 3150 steel, annealed, Brinell hardness No. 196; S.A.E. 1020 steel, annealed, Brinell hardness No. 131; S.A.E. 1035 steel, annealed, Brinell hardness No. 156; carbon tool steel, annealed, Brinell hardness No. 152; S.A.E. 1112 steel, cold drawn, Brinell hardness No. 217; The characteristics of the cutting fluids are given in Table 7.

The characteristics of the drills used in the first series of tests are given in Table 8. The smaller drills, from 1/8 to 1/2 in. diameter, all had helix angles of 30 deg. and ratios of web thickness to diameter of 0.185. Point angle was 120 deg., peripheral-edge angle, 130 deg., and clearance angle at periphery was 6 deg.

**Torque and Thrust Formulas.**—The experiments indicated that the torque required for drilling all metals could be expressed by an equation of the form

$$T = Cf^m d^n \quad \ldots\ldots\ldots\ldots \quad [11]$$

where $T$ = torque developed by drill, ft.-lb.; $f$ = feed of drill, in. per revolution; $d$ = diam. of drill, in.; $C$ = a constant depending on material drilled, cutting fluid and the drills. The values of the exponents $m$ and $n$ vary with the material cut and are given in Table 9. Values of $C$ are given in Table 10.

The research established that the values of exponents $m$ and $n$ were identical for all of the steels investigated, and the torque equation for steels, therefore, becomes

$$T = Cf^{0.78}\,d^{1.8} \quad \ldots\ldots\ldots\ldots \quad [12]$$

The thrust equation takes two forms depending on the material drilled. For aluminum and brass it takes the form

$$B = Kf^p\,d^r \quad \ldots\ldots\ldots\ldots \quad [13]$$

For steel and iron it takes the form

$$B = Kf^p\,\{(d/5) + (w/d)\}^r \quad \ldots\ldots\ldots\ldots \quad [14]$$

$B$ = thrust, lb. $K$ = a constant corresponding to $C$ in the torque equation; $w$ = web thickness, in. Values of $p$ and $r$ are given in Table 9. Values of $K$ are given in Table 10.

### Table 7.—Properties of Cutting Fluids Used in Tests

| No. | Composition | Base of Mineral Oil | Percent Compounding by Analysis | Viscosity at deg. F. | | | A.P.I.† Gravity at 60 deg. F. | Specific Gravity ‡ |
|---|---|---|---|---|---|---|---|---|
| | | | | 100° | 130° | 210° | | |
| 1 | Dry cutting.......... | ...... | .......... | ..... | ..... | ..... | ....... | ...... |
| 2 | 1 1/2% by weight of borax in water...... | ...... | 1.19 borax | ..... | ..... | ..... | ....... | ...... |
| 3 | 1 part soluble oil and 50 parts water...... | ...... | 2.3 soluble oil | 31 | ..... | ..... | ....... | ...... |
| 4 | 1 part soluble oil and 10 parts water...... | ...... | 8.37 soluble oil | 32 | ..... | ..... | ....... | ...... |
| 5 | No. 1 lard oil......... | ...... | 12.23–13.6 F.F.A.*<br>86.4 fat | 200 | 110 | ..... | 23.7 | 0.9117 |
| 6 | Light mineral oil...... | Mixed | .......... | 111 | 69 | 40.5 | 23.5 | .9129 |
| 7 | Heavy mineral oil..... | Mixed | .......... | 209 | 104 | 45.5 | 23.0 | .9159 |
| 8 | 10% No. 1 lard oil in mineral oil......... | Paraffin | 10.12 lard oil | 101 | 67 | 41 | 29.4 | .8794 |
| 9 | 5% oleic acid in mineral oil............ | Paraffin | 4.9 oleic acid | 97 | 65 | 40.5 | 29.2 | .8805 |
| 10 | Sulphurized mineral oil | Asphaltic | 4.38 sulphur | 105 | 67 | 40 | 21.6 | .9242 |
| 11 | Sulphurized lard oil with 5 parts mineral oil................ | Mixed | 2.34 sulphur<br>4.443 F.F.A.* sulphurized<br>13.94 fat sulphurized | 102 | 67 | 41.5 | 25.4 | .9108 |

* F.F.A. = free fatty acids. † American Petroleum Institute. ‡ National Petroleum Association.

### Table 8.—Characteristics of Drills Used in Tests

| Drill Diam., in. | Web Thickness, in. | Helix Angle, deg. | Point Angle, deg. | Chisel-edge Angle, deg. | Clearance at Periphery, deg. | Actual r.p.m. | Ft. per min. | Feed,* in. |
|---|---|---|---|---|---|---|---|---|
| 1/2 | 0.081 | 29 | 121 | 129 | 5–20 | 449 | 58.8 | 0.009 |
| 3/4 | .104 | 31 | 121 | 134 | 5 | 300 | 59.0 | .012 |
| 1 | .141 | 31 | 121 | 136 | 5 | 231 | 60.5 | .013 |
| 1 1/4 | .180 | 30 | 121 | 138 | 5 | 174 | 57.1 | .015 |
| 1 1/2 | .213 | 31–20 | 121 | 138 | 5 | 151 | 59.4 | .015 |

* Feeds used for drilling aluminum and free cutting brass were double these values, in order to obtain torque and thrust values large enough to measure without appreciable error.

### Table 9.—Values of Exponents in Torque and Thrust Formulas for Drills 1/2 to 1 1/2 in.

| Exponent | Al Alloy S.A.E. 33 | Leaded Brass | Cast Iron | Malleable Cast Iron | All Steels |
|---|---|---|---|---|---|
| $m$.............. | 0.83 | 0.73 | 0.60 | 0.66 | 0.78 |
| $n$.............. | 1.9 | 1.9 | 1.7 | 1.8 | 1.8 |
| $p$.............. | 1.1 | 0.6 | 0.73 | 0.75 | 0.87 |
| $r$.............. | 1.2 | 1.0 | 1.9 | 1.75 | 2.12 |

### Table 10.—Values of Constants in Torque and Thrust Formulas

| Material | Constant | Cutting Fluids (See Table 7.) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
| Aluminum Alloy S.A.E. 33 | $C$ | 646 | 601 | 589 | 553 | 482 | 471 | 471 | 443 | 453 | 495 | 463 |
| | $K$ | 75,030 | 56,910 | 53,230 | 51,070 | 45,970 | 51,400 | 49,270 | 44,380 | 45,300 | 47,000 | 45,270 |
| Leaded Brass | $C$ | 418 | 418 | 418 | 418 | 418 | 418 | 418 | 418 | 418 | 418 | 418 |
| | $K$ | 6938 | 6636 | 6850 | 6636 | 6636 | 6400 | 6850 | 6636 | 6400 | 6636 | 6530 |
| Cast Iron | $C$ | 370 | 340 | 334 | 344 | 336 | 334 | 334 | 324 | 306 | 314 | 330 |
| | $K$ | 160,000 | 146,500 | 150,000 | 148,000 | 153,000 | 143,000 | 161,000 | 144,000 | 143,000 | 154,000 | 158,000 |
| Malleable Cast Iron | $C$ | 544 | 544 | 544 | 544 | 485 | 485 | 485 | 485 | 485 | 485 | 485 |
| | $K$ | 152,000 | 152,000 | 152,000 | 152,000 | 129,700 | 129,700 | 129,700 | 129,700 | 129,700 | 129,700 | 129,700 |
| Steel, S.A.E. 3150 | $C$ | 2215 | 1937 | 1995 | 2025 | 2100 | 2124 | 2068 | 2164 | 2294 | 1696 | 1779 |
| | $K$ | 912,500 | 760,500 | 780,200 | 773,000 | 864,700 | 796,000 | 810,200 | 892,200 | 927,000 | 673,200 | 683,000 |
| Steel, S.A.E. 1020 | $C$ | 2186 | 1815 | 1786 | 1758 | 1623 | 1711 | 1670 | 1645 | 1707 | 1457 | 1457 |
| | $K$ | 906,000 | 772,000 | 785,000 | 785,000 | 711,000 | 751,000 | 738,000 | 738,000 | 751,000 | 651,000 | 651,000 |
| Steel, S.A.E. 1035 | $C$ | 1843 | 1622 | 1622 | 1582 | 1498 | 1560 | 1560 | 1536 | 1536 | 1363 | 1363 |
| | $K$ | 795,000 | 725,000 | 702,000 | 711,000 | 629,000 | 651,000 | 651,000 | 641,000 | 663,000 | 651,000 | 641,000 |
| Steel, Carbon Tool, Annealed | $C$ | 2089 | 1848 | 1918 | 1864 | 1840 | 1984 | 1938 | 1953 | 1937 | 1649 | 1715 |
| | $K$ | 933,000 | 901,000 | 927,000 | 921,000 | 962,000 | 942,000 | 927,000 | 965,000 | 902,000 | 759,000 | 758,000 |
| Steel, S.A.E. 1112 | $C$ | 1222 | 1309 | 1222 | 1222 | 1222 | 1222 | 1222 | 1222 | 1222 | 1162 | 1139 |
| | $K$ | 624,000 | 613,000 | 580,300 | 604,700 | 613,000 | 613,000 | 624,000 | 613,000 | 613,000 | 613,000 | 613,000 |

The research established that the values of exponents $p$ and $r$ were identical for all of the steels investigated, and the thrust equation for steels, therefore, becomes

$$B = Kf^{0.87}\{(d/5) + (w/d)\}^{2.12} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15]$$

**Tests on Small Drills.**—The tests on small drills showed that the torque equation for all steels was the same as for large drills, that is, it is identical with the equation [12]. The torque equation for other materials with small drills is identical in form with equation [11], but the value of exponents $m$ and $n$ vary as follows:

| | Al Alloy S.A.E. 33 | Leaded Brass | Cast Iron | Malleable Cast Iron |
|---|---|---|---|---|
| $m =$ | 0.83 | 0.83 | 0.69 | 0.78 |
| $n =$ | 1.9 | 1.9 | 1.7 | 1.8 |

The equation for thrust, with small drills, is of the same form as equation [13], the exponent $r$ in all cases being 1, with the exception of aluminum alloy, in which case $r = 1.2$. The value of the exponent $p$ for all of the steels is 0.87. Its value for the other materials is as follows:

| | Al Alloy, S.A.E. 33 | Leaded Brass | Cast Iron | Malleable Cast Iron |
|---|---|---|---|---|
| $p =$ | 1.0 | 0.6 | 0.73 | 0.75 |

Values for constants $C$ and $K$ in the torque and thrust equations are as follows:

| | Al Alloy S.A.E. 33 | Leaded Brass | Cast Iron | Mall. Cast Iron | Steel | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | S.A.E. 3150 | S.A.E. 1020 | S.A.E. 1035 | Carbon Tool | S.A.E. 1112 |
| $C =$ | 556 | 512 | 347 | 689 | 1,500 | 1,740 | 1,300 | 1,842 | 1,000 |
| $K =$ | 90,400 | 6160 | 25,800 | 24,400 | 104,300 | 96,000 | 133,100 | 108,500 | 74,100 |

The effect of thinning the point of drills was to materially decrease the thrust. Values of thrust may be lowered as much as 50% by point thinning. Both torque and thrust decrease with an increase of helix angle, both being reduced a total of 26% as helix angle is increased from 21° 26′ to 40° 23′. This probably is due to relatively easier escape of chips from the flutes. However, the life of the drill with the largest helix angle is considerably below the life with the smaller helix angle. The influence of web thickness on torque and thrust is negligible with feeds up to 0.004 in. For heavier feeds, torque increases rapidly, probably because of the decreased cross-sectional areas of the flutes, causing increased frictional resistance to the flow of chips. Experiments with drills 1 in. diameter, however, show constant torque for drills with different web thicknesses.

**Effect of Cutting Fluids.**—The research showed that savings were possible with cutting fluids, as compared with dry drilling. Taking the net power output for drilling dry, as 100%, savings were shown as follows: Aluminum alloy S.A.E. 33, 90% mineral oil and 10% lard oil, 31.7%; leaded brass, none; cast iron, mineral oil, with 5% oleic acid, 17.2%; malleable cast iron, lard, mineral, compounded or sulphurized oil, 10.5%; S.A.E. 3150 steel, sulphurized mineral oil, 23.0%; S.A.E. 1020 steel, sulphurized oil, 33.3%; S.A.E. 1035 steel, sulphurized lard-mineral oil, 26.0%; carbon tool steel, sulphurized mineral oil, 21%; S.A.E. 1112 steel, sulphurized lard-mineral oil, 6.8%.

Discussing the paper, Alvan Davis, *Trans.* A.S.M.E. RP 55–1, p. 25, 1933, stated that there is no gain in using cutting fluid with free-cutting brass at the speeds used in the Boston-Oxford experiments. However, in automatic machine operations, speeds of about five times those reported in the paper are used, and a copious supply of light paraffin oil is necessary to prevent the drills from burning up.

# PRESSURE WORKING OF METALS

By H. B. Morse

## 1. PLASTIC PROPERTIES OF METAL

**THE PLASTIC WORKING** of metal requires the use of powerful machines to change the shape, contour or cross-section of the initial shape by force as compared with change of shape by the whittling operations of machine tools. The theory involved applies not only to mechanical and hydraulic press operations but also to hammer work, rolling operations, bench and wire drawing, spinning and shearing.

**HOT AND COLD OPERATIONS** are to be found under almost all of these headings. The obvious distinction is whether the metal is to be heated preparatory to working or whether it is to be worked at room temperature. Another worth-while distinction is the relation of working temperature to the recrystallization temperature, or range of temperatures.

**Above the Recrystallization Range** the metal movement seems to be largely inter-crystalline. Any (transcrystalline) slip-plane distortion through the crystal is corrected at once by what has been termed *spontaneous annealing*, and the crystal structure is unstrained upon completion of the operation. At elevated temperatures the amount of work performed on the metal is practically unlimited except by the capacity of the part to "hold the heat." Working loads increase with speed of working in the lower speed ranges, but most production operations seem to be above this range and are affected very little by changes of speed. It should be noted that tin and lead are above their recrystallization range at normal room temperatures and work at such temperatures in the same way that steel and brass work at elevated temperatures. Zinc is in its hot working range at about 300° F.

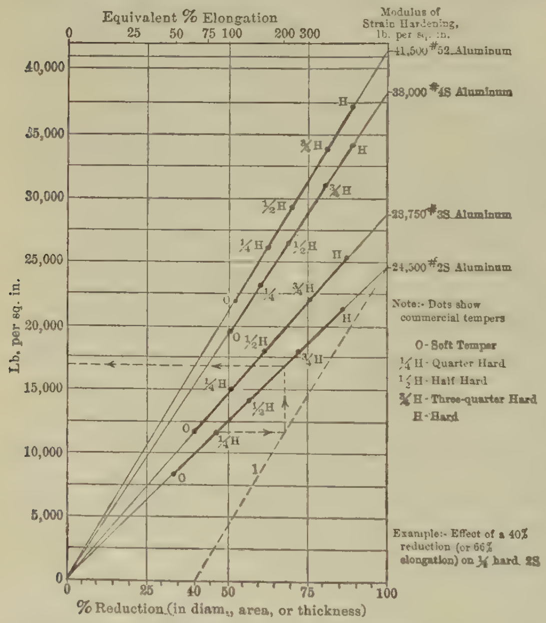

Fig. 1. Plastic Range Chart for Four Grades of Aluminum at Room Temperature

**COLD WORKING** of metals (below their recrystallization temperatures) causes cumulative strain-hardening with a progressively increasing yield point, due to slip-plane movements to the individual crystal. After severe strain-hardening in one or a series of operations the metal must be annealed to restore an unstrained crystal structure before further cold working can be undertaken.

**ANNEALING** requires sufficient temperature to enable the atoms of the distorted structure to gain sufficient kinetic energy to rotate themselves into their normal space lattice relation, in the formation of new and unstrained crystals. This is recrystallization, and occurs at somewhat lower temperatures in the more severely worked metals. The new small crystals (or grains) increase in size with increasing temperature or continued application of heat until they reach the desired size. Plasticity improves and yield point becomes lower as grain size increases, but excessive grain size results in dangerous non-uniformity. In annealing low carbon steels (0.02 C–0.12 C) it is necessary to heat quickly to a point above the *germination temperature*. Germination is an excessive and sudden grain growth occurring at suitable combinations of strain and temperature. A non-uniformly strained stamping is likely to include some local area of favorable strain, and if germination occurs the local large grains will result in an area of weakness.

**THE PLASTIC RANGE** (See Figs. 1 and 2) for the cold working of metals lies between the initial yield point in the commercially annealed state, usually designated as soft temper, and the severely strain-hardened state in which all conveniently oriented slip-planes have been used up so that internal fractures are likely.

**THE PLASTIC CYCLE** includes strain-hardening by cold working and then restoration to the unstrained state by annealing. This cycle may be repeated many times in fabricating the common metals.

**THE APPLICATION OF STRAIN-HARDENING THEORY,** Fig. 1, permits: Adding the effect of a series of cold-working operations; judging the need for annealing; finding the yield point or resistance developed in an operation, for figuring equipment required. An example in dotted lines is arranged with arrows to show the method of using the chart. The plastic range for each metal is indicated by a heavy line and the standard aluminum tempers (listed beside Fig. 1) are indicated in order by dots. There is still a shortage of consistent test data relative to the plastic range properties of metals, but the selection of tentative values given in Table 1 and Figs. 1 and 2, has been made with considerable care.

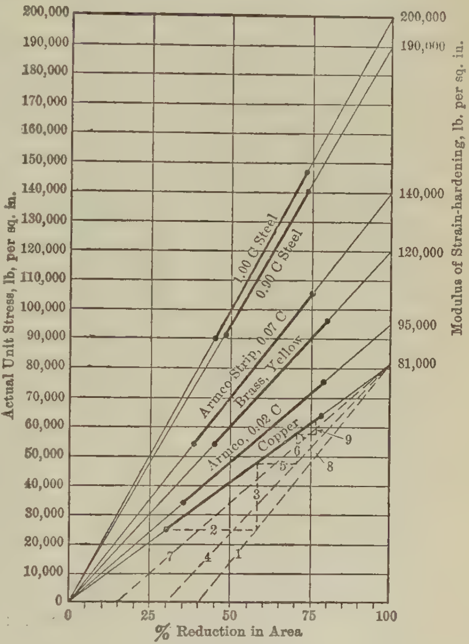

FIG. 2. Strain-Hardening Curves

EXAMPLE 1.—A reduction of 40% is to be made on 2S Aluminum 1/4 H. On Fig. 1, first draw a line from the modulus of elasticity of 2S Aluminum (24,500), to 40% on the base line. Then draw a line horizontally from the initial state (1/4 H) across to line 1. Then draw a line vertically upward to a point representing completion of the work on the plastic curve. A horizontal line drawn from this point back to the unit stress line shows about 16,500 lb. per sq. in. yield point reduction.

EXAMPLE 2.—Determine the cumulative strain-hardening effect of a series of operations on copper, and check approximately whether or not an annealing operation is required. It is assumed that copper is received in the soft temper or commercially annealed state and that the operations to be performed, involve successive reductions of 40, 30 and 50%. (See Fig. 2.) The construction lines are numbered in order, and indicate successive yield points as follows: Before the first operation, about 25,000 lb. per sq. in.; after the first operation, about 42,000 lb. per sq. in.; after the second operation, about 51,000 lb. per sq. in.; after the third operation, about 62,000 lb. per sq. in. The final point is still within the plastic range of the material so that an annealing operation has not yet been required. It is obvious however that the material is now severely strain-hardened and must be annealed before any further working is possible.

## 2. EQUIPMENT

**METAL WORKING EQUIPMENT** is most easily discussed by comparing several groups according to their structural characteristics rather than their applications. This particularly is true because applications often overlap.

**THE HAMMER GROUP** (See also pp. 19–03 to 19–22) is the oldest, tracing directly to the swinging sledge in the hands of the early blacksmith. In each case the principle is essentially the same; the job absorbs at each blow the whole energy of the hammer head. One, several, or many blows may be required to complete the part. As most types of hammers

## Table 1.—Plastic Physical Properties of Metals

| Material | Theoretical Yield, Point, lb. per sq. in.: Minimum, Commercially Annealed | Theoretical Yield, Point, lb. per sq. in.: Maximum, Severely Cold-worked | Reduction in Area, Max., Annealed, % | Shearing Resistance and Penetration for Clean Fracture: Annealed to Soft or "O" Temper, lb. per sq. in. | Annealed to Soft or "O" Temper, per cent | Cold-worked to Hard Temper, lb. per sq. in. | Cold-worked to Hard Temper, per cent | Tensile Strength, (nominal) Annealed, lb. per sq. in. | Elongation in 2 in., Average, Annealed, percent | Young's Modulus of Elasticity, lb. per sq. in. | Modulus of Strain Hardening, Tentative, lb. per sq. in. | Bulk Modulus, (Volume), lb. per sq. in. | Annealing Temperature, Approx., Commercial, deg. F. | Forging Temperature, Approx. deg. F. | Thermal Coefficient of Expansion, in. per in. per deg. F. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Aluminum, 2S. comm'l, pure | (4000)* –8,000 | H- 21,000 | 80 | 8,000 | 60 | H- 13,000 | 30 | 13,000 | 35–45 | 10,300,000 | 25,000 | 10,200,000 | 650 | ........ | 0.000 010 |
| " 3S. manganese | 11,500 | H- 25,000 | 80 | 11,000 | ...... | H- 16,000 | ..... | 16,000 | 30–40 | 10,300,000 | 29,000 | .......... | 750 | ........ | .......... |
| " 17S, for heat-treat | 30,000 | ......... | ...... | ...... | ...... | ......... | ..... | ....... | ...... | ......... | ........ | .......... | 630–650 | 400 | .......... |
| " 52S, chromium | 21,000 | H- 37,000 | 80 | 14,000 | ...... | H- 36,000 | ..... | 29,000 | 25–30 | 10,300,000 | 38,000 | .......... | 650 | ........ | .......... |
| Brass, yellow, for cold working | (10,000)*–40,000 | 95,000 | 75 | 32,000 | 50 | 52,000 | 20 | 50,000 | 65 | 13,400,000 | 120,000 | 8,800,000 | 1100 | ........ | .......... |
| " B, for forging | ............ | ......... | ...... | ...... | ...... | ......... | ..... | ....... | ...... | ......... | ........ | .......... | ........ | 1200–1400 | .......... |
| Bronze, alloy X | 25,000 | 120,000 | 53 | 36,000 | 25 | 1/4H-42,000 | 17 | 60,000 | 46 | ......... | 250,000 | .......... | ........ | ........ | .......... |
| Copper | (10,000)*–25,000 | 62,000 | 65 | 22,000 | 55 | ......... | ..... | 33,000 | 68 | 14,500,000 | 81,000 | 17,400,000 | ........ | 1600–1800 | 0.000 009 |
| Gold | ............ | ......... | ...... | ...... | ...... | ......... | ..... | ....... | ...... | 11,600,000 | ........ | 23,200,000 | 600–800 | ........ | 0.000 008 |
| Iron, Cast gray | 8,000†<br>30,000‡ | ......... | ...... | ...... | ...... | ......... | ..... | 23,000 | ...... | 12,000,000 | ........ | 13,900,000 | ........ | ........ | 0.000 005 5 |
| Iron, High test | 15,000†<br>40,000‡ | ......... | ...... | ...... | ...... | ......... | ..... | 45,000 | ...... | 20,000,000 | ........ | .......... | ........ | ........ | .......... |
| Lead | (1000)*– 3,000 | 4,000 | ...... | 3,500 | 50 | ......... | ..... | ....... | ...... | 2,470,000 | ........ | 1,100,000 | § | ........ | 0.000 017 |
| Monel Metal | 28,000 | H-112,000 | 63 | ...... | ...... | ......... | ..... | 70,000 | 30–50 | ......... | | .......... | 1100–1200 | ........ | .......... |
| Nickel | 21,000 | H-115,000 | 75 | 35,000 | 55 | ......... | ..... | 70,000 | 47 | 32,000,000 | ........ | 24,600,000 | 1200 1300 | ........ | 0.000 007 |
| Silver | ............ | ......... | ...... | ...... | ...... | ......... | ..... | ....... | ...... | 10,900,000 | ........ | 14,500,000 | 600–800 | ........ | 0.000 010 |
| Steel, 0.03 C Armco | (20,000)*–35,000 | 70,000 | 76 | 34,000 | 60 | ......... | ..... | 42–48,000 | 48 | 30,000,000 | 100,000 | .......... | 1300–1400 | ........ | .......... |
| Steel, .15 C | (32,000)*–55,000 | 110,000 | 60 | 48,000 | 38 | 61 000 | 25 | 57,000 | 35 | 30,000,000 | 150,000 | 23,200,000 | ........ | ........ | 0.000 009–<br>0.000 016 |
| Steel, .50 C | (55,000)*–70,000 | ......... | 45 | 71,000 | 24 | 90,000 | 14 | 85,000 | 25 | 30,000,000 | ........ | .......... | 1300–1400 | 1800–2100 | .......... |
| Steel, 1.00 C | 90,000 | 145,000 | 20–40 | 115,000 | 10 | 150,000 | 2 | 120,000 | 10 | 30,000,000 | 200,000 | .......... | ........ | ........ | .......... |
| Steel, Silicon high | ............ | ......... | ...... | 65,000 | 30 | ......... | ..... | ....... | ...... | ......... | ........ | .......... | ........ | ........ | .......... |
| Steel, Stainless Enduro AA | 50,000 | 120,000 | 60 | ...... | ...... | ......... | ..... | 80,000 | 25 | 30,000,000 | ........ | .......... | ........ | ........ | .......... |
| Steel, Low Carbon (18-8) | (35,000)*–50,000 | 165,000 | 71 | 57,000 | 39 | ......... | ..... | 95,000 | 57 | ......... | ........ | .......... | 1300–1900 | 2100 | .......... |
| Tin | (2000)*– 4,000 | 4,500 | ...... | 5,000 | 40 | ......... | ..... | ....... | ...... | 7,260,000 | ........ | 7,260,000 | § | ........ | 0 000 015 |
| Zinc | (12,000)*–28,000 | ......... | 43 | 14,000 | 50 | 19,000 | 25 | 30–37,000 | 27 | 13,100,000 | ........ | 5,100,000 | ¶ | 200–300 | 0.000 0145 |

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* Figures in parenthesis are misleadingly low test values.

† Tensile.

‡ Compression. § Below room temperature. ¶ Room temperature.

employ a gravity fall, the energy delivered to do work at each stroke is the product of the weight of the hammer head multiplied by the distance through which it falls. In some cases the velocity of the head is increased mechanically and its energy must be derived from its weight and its velocity as it reaches the work.

**Helve Hammers, Steam Hammers and Air Hammers** may fall in either of the above classes. They usually are used for simple forging and taper drawing operations with plain-faced hammer heads and anvils.

**Board-lift Drop Hammers** are still widely used for forging operations which require dies with several developed shapes in them to preform and finish-form the required part. The hammer usually is rated by the weight of the head in pounds.

**Belt-lift Hammers and Rope-lift Hammers** usually are used for single die work. The belt-lift hammers developed for the jewelry trade, had a belt attached to the slide and carried up over a constantly revolving drum. Pulling on this belt gave it enough traction on the pulley to lift the slide. The rope-lift drop hammers followed a similar principle except that the head was very heavy and large in area and was lifted by double ropes wrapped two or three times around the drum for better lifting traction when tightened.

**ROTARY SWAGING MACHINES** for tapering, pointing and reducing tubes and rods, resemble helve hammers in accomplishing their results by many quick blows. The blows are delivered, however, by a group of cam-actuated rolls revolving around the work.

**PERCUSSION PRESSES,** or power screw presses, are an intermediate development between the drop-hammer and power-press. The slide is driven up and down by a large steep-pitch screw, traveling up through a fixed nut in the frame to a light fly-wheel on top. Friction contact of the fly-wheel rim with a constantly revolving disc on one side drives the slide down at increasing velocity until the die comes in contact with the work and all the energy of the fly-wheel is absorbed. Then another revolving disc makes contact with the other edge of the wheel and lifts the slide. Foot treadle control, similar to that of a drop hammer, is used.

**HYDRAULIC PRESSES** come after hammers in development, but occupy a rather different field. Essentially slow-moving machines, due to fluid friction and inertia of long columns of liquid and of accumulators, they were limited to hot-forming operations on parts heavy enough to hold heat over a considerable time, and to cold operations on limited production. Subsequent developments have increased their speed materially at the expense of some complication by introducing variable-delivery motorized pumps, gravity fill systems and foot valve controls. Fast-acting hydraulic presses are adapted to some production metal-working operations, both hot and cold, particularly where it is important to limit the working loads. The operations include particularly heavy forging, hot extrusion of shells and shapes, heavy bending and cold flattening.

**MECHANICAL POWER PRESSES** and such modifications as headers, upsetting machines, bulldozers, bending brakes and (boiler) punches, comprise essentially: A substantial frame, a reciprocating slide with actuating crank-shaft, a clutch for control and a fly-wheel. This fly-wheel is the important distinction from previous types of machines, in that it permits large energy storage during the idle period of each cycle and practically instant delivery of the energy required during the brief working period. Many variations of press design are possible in size, shape and position of frame, type of clutch and use of gearing. Accessories include knockouts, blank-holding devices and other clamping means, and mechanical feeds of many types.

**PRESS POSITION** is one of the most important design modifications. The upright position is the most common, with the slide descending vertically to perform the work. A considerable proportion of the smaller sizes are made *inclinable*, so that they may be used in the upright position or inclined as much as 30 or 40 deg. at will, to facilitate gravity discharge of the work. This same consideration is responsible for many specialized and single-purpose machines being inclined at greater angles (45, 60, 75 deg.) and an even larger group being mounted horizontally. The latter group includes nail machines, forging machines, special purpose headers, some redrawing presses, bulldozers, etc.

**OTHER DESIGN FEATURES.**—Pressure may be delivered from crank- or eccentric-shafts to the work slides; through one central connection (as in single-crank presses), or through two connections (double-crank presses), or through four connections at the corners of the slide. For extremely short strokes and high pressures, the crank-shaft is moved to one side and used to straighten a pair of knuckles or toggles to obtain high mechanical advantage. The energy-storing fly-wheel usually is mounted directly on the crank-shaft for relatively fast and easy jobs. Gearing is introduced between the fly-wheel and the crank-shaft, as the energy demand and stroke increase, to speed up the wheel and to keep down the tool operating speed. Single gear reductions, up to about 7 $^1/_2$ : 1, suffice on small and moderate size presses. The larger presses usually require double gearing. Twin driving gears at each end of the crank-shaft, to divide the torsional driving strain, become advisable for extreme strokes (say 2 or 3 times the shaft diameter). Positive engagement clutches between the crank-shaft and driving wheel, are widely used on the smaller presses which may be turned over by hand for die setting purposes, with fair ease. Friction clutches on the back-shaft are preferred on large presses to facilitate die setting.

**DRAWING OPERATIONS** in power presses require added motions for blank-holding purposes. The common single-action presses may be fitted for the easier drawing operations by adding spring or rubber pressure attachments or by using pneumatic die cushions which give better results and are more convenient. For the more severe draws, shallow stretching, and for power economy, double-action presses are used. These have the crank-actuated drawing slide or plunger, and around it an outside slide which is forced down rigidly to hold the blank, by means of cams in the early models, and now by a system of toggle linkages. A further modification known as "triple-action" equipment permits combined drawing and redrawing at one handling.

**SHEARS.**—Squaring shears or "gate" shears are essentially a single-purpose modification of the gap frame double-crank type of power press, arranged with a pair of shear blades, a hold-down to clamp the metal while cutting it and a set of gages to assist in locating the cut. Circle-cutting shears, rotary slitting shears and gang slitting machines for two or more parallel cuts, all belong to the next group.

**ROLLING ACTION** characterizes a large group of metal-working machinery and distinguishes it from all of the foregoing *reciprocating action* machines. Working pressures in all cases must exceed the yield point of the metal in whatever state received; hot or cold, annealed or strain-hardened. In the rolling machines the working pressure is localized on a very small area at the point or line of contact between the rolls and the work. In most cases, except oscillating forging rolls, the work is progressive and a continuous and more or less uniform supply of power must be maintained.

**ROLLING MILLS** are used for the initial reduction of a hot mass of cast metal, and for subsequent hot and cold operations in the production of plate, sheet, strip and various rolled shapes. For shapes (rails, beams, etc.) a series of grooves in the rolls must be worked out to develop the desired shape from an initially rectangular section. For light-gage cold rolling, small diameter work rolls may be used to reduce the contact area and load relative to the reduction per pass. These work rolls should be backed up by larger diameter rolls to minimize deflection and maintain uniformity of gage. For large production, trains of several stands of mills in series are used to perform consecutive rolling operations at one pass.

**PROGRESSIVE FORGE ROLLING** is a modification of shape rolling, to compete with large-quantity drop forging. In this process matching impressions of the desired forging are cut at intervals in the surfaces of a pair of mill rolls of suitable diameter. The forgings are trimmed out of each bar of flash at the mill. Related to this process by antedating it, *oscillating forging rolls* with a single pair of impressions have been used particularly for suitable shapes in the production of agricultural implements.

**PROGRESSIVE ROLL FORMING MACHINES** use several pairs of suitably shaped rolls to convert metal from flat strip or coil form to a variety of shapes such as molding, channel, lock-seamed tubing, etc.

**SPINNING LATHES** are used for surface burnishing, trimming and curling of some cooking utensils and for spinning wrinkles out of some taper jobs. They also are used to spin flat blanks or plain drawn shells to various shapes on wooden or metal chucks. The working area on a spinning operation usually is little more than a point contact against the spinning roll or hand tool.

**BEADING, FLANGING, THREAD ROLLING, KNURLING, CURLING, DOUBLE SEAMING** and other rotary cold-forming operations are performed in a variety of ma-

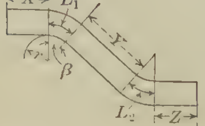

FIG. 3. Length of Strip for Bending

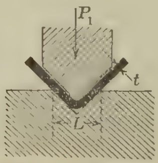

FIG. 4. Bending in V-Dies

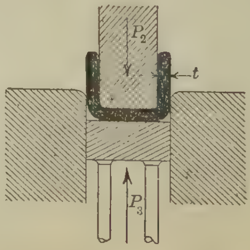

FIG. 5. Bending in U-Dies

chines from simple parallel-shaft, hand-served types, to multiple-spindle, chute-feed, high-speed automatics.

**PROFILE DIES,** to suit the cross-section of the required product, characterize a group of machines including wire blocks, draw benches and hydraulic extrusion presses.

**WIRE DRAWING** employs power-driven capstans or drums to pull the wire through reducing dies, which have much in common with dies for ironing cartridge case walls and other drawn shells.

**DRAW BENCHES** employ a capstan, chain and clamp to pull cold strip metal through proper die slots to change its shape to that of a molding or other progressively bent job. Horizontal hydraulic extrusion presses may be arranged for rather similar work, except that they achieve their results by pushing hot material through a suitably shaped die opening.

## 3. SELECTION OF EQUIPMENT

**FORMULAS** for use in the selection of the foregoing equipment to suit particular job requirements are given below. They are kept as simple as possible, and necessarily omit some more or less intangible factors for which a factor of personal judgment should be added in practically all cases. It would be foolish to try to use more complicated formulas to avoid some small error when the materials to be worked are subject to as wide variations in physical properties as is found to be unavoidable.

**HAMMERS.**—For data governing the selection of hammers, see pp. 19–03 to 19–22.

**TRIMMING PRESS** sizes which have been found suitable for use with drop hammers under average conditions are given in Table 2.

**Table 2.—Trimming Press Sizes**

(E. W. Bliss Co., Brooklyn, N. Y.)

| Drop-hammer ram weight, lb | 600 | 800 | 1000 | 1200 | 1500 | 2000 | 2500 | 3000 | 5000 | 8000 |
|---|---|---|---|---|---|---|---|---|---|---|
| Trimming press, shaft diameter, in | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 6 1/2 | 7 | 8 | 9 |

**ROLLING OPERATIONS.**—The working or contact area between the rolls may be taken as:

$$A = w\sqrt{R \times (t_1 - t_2)} \qquad [1]$$

where $t_1$, $t_2$ = respectively thickness of strip entering and leaving the rolls, in.; $R$ = radius of roll, in.; $w$ = width of strip, in.

The working or screw-down pressure then will be

$$P = C \times S \times A \qquad [2]$$

where $S$ = yield point of the metal at that particular stage in the rolling process; refer to the strain-hardening curves, Figs. 1 and 2. $C$ = 1.3 may be taken for soft temper and relatively heavy gage material up to 3.8 for light-gage and strain-hardened work, to cover both the elastic deformation of the rolls and the pyramiding of the pressure due to constriction of flow in the working area. For further information also see bulletin entitled Rolling-Mill Data for Calculation and Design, issued by S. K. F. Industries, Inc., Philadelphia, Pa.

**BENDING.**—The length of strip or bar, $L$ in Fig. 3, to be allowed for making bends, when cutting blanks, depends on the inside radius $r$, the angle of the bend $\beta$, and the thickness of the metal $t$. The allowance $L$ is not a true arc at a distance $^1/_2\ t$ from the die radius. A true arc is close enough for calculations, however, if it is taken at a radius of $(r + 0.4\,t)$. Then

$$L = (r + 0.4\,t) \times (2\pi\beta/360) \qquad [3]$$

$$\text{Total length of blank} = X + L_1 + Y + L_2 + Z \qquad [4]$$

**BENDING PRESSURE FOR V-DIES.**—See Fig. 4. Let $S$ = ultimate tensile strength of the metal, lb. per sq. in.; $L$ = span of female die, in.; $w$ = width of strip, in.; $t$ = thickness of strip, in. Then, bending pressure $P_1$, in pounds, is

$$P_1 = (C \times S \times w \times t^2)/L. \qquad [5]$$

$C$ = 1.33 when $L = 8t$; = 1.24 when $L = 12t$; = 1.2 when $L = 16t$. If the punch is set down hard to squeeze the surface metal and cause balancing strains in order to minimize "spring-back," the load will be higher than for bending only. In such a case, the job should be considered as coining, and the contact area should be reduced to suit.

**BENDING IN U-DIES.**—See Fig. 5. The bending pressure $P_2$ on the punch will be

$$P_2 = {}^2/_3\, S \times w \times t \times (1 + t/L) \qquad [6]$$

and the pressure $P_3$ required on the pad to hold the bottom from bowing will be

$$P_3 = {}^2/_3\, (S \times w \times t^2)/L \qquad [7]$$

Notation for equations [6] and [7] is the same as for equation [5].

**BENDING IN ROLLS.**—The screw-down pressure $P$, in pounds, may be approximated from

$$P = 12 \times \Delta \times E_m \times w \times t^3, \quad \ldots\ldots \quad [8]$$

where $\Delta$ = deflection, in.; $E_m$ = modulus of elasticity; other notation as before.

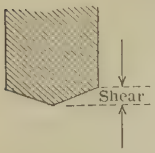

Fig. 6. Shear on Punches

**BLANKING, PIERCING OR SHEARING** with flat tools. Pressure $P_1$, in pounds, for rectangular work is given by

$$P_1 = L \times t \times S \quad \ldots\ldots\ldots\ldots \quad [9]$$

For round work, the equation becomes

$$P_1 = \pi \times D \times t \times S \quad \ldots\ldots\ldots \quad [10]$$

where $S$ = shearing strength of the metal (Table 1); $L$ = total length or periphery of the shearing cut, in.; $t$ = thickness of metal, in.

**BLANKING WITH SHEARED TOOLS.**—Pressure required $P_2$, in pounds, is given by

$$P_2 = (P_1 \times t \times p)/s \quad \ldots\ldots\ldots\ldots \quad [11]$$

where $p$ = penetration, percent; $s$ = shear, in. (see Fig. 6); $t$ = thickness of metal, in. $P_1$ is obtained by equation [10] and values of $p$ may be taken from Table 1.

**WORK IN BLANKING.**—Let $W_e$ = working energy = (working pressure × working distance), usually in inch-tons. Then

$$W_e = (t \times p \times P_1)/2000 \quad \ldots\ldots\ldots\ldots \quad [12]$$

Notation is the same as in equation [11]. Sufficient clearance for clean fracture is assumed and no allowance is made for friction losses.

**BLANK DIAMETERS FOR DRAWING SHELLS.**—Let $D$ = blank diameter, in.; $d$ = mean shell diam., in.; $h$ = height of shell, in. Then

$$D = \sqrt{d^2 + 4\,d\,h} \quad \ldots\ldots\ldots\ldots \quad [13]$$

Refer also to the wide range given in Table 3.

**LOADS IN LIMIT DRAWS.**—The load in a limit draw is the force $P$, pounds, necessary to push the bottom out of the shell being drawn. If $L$ = length of perimeter, in.; $t$ = thickness of metal, in.; $S$ = ultimate tensile strength of the metal (Table 1),

$$P = L \times t \times S \quad \ldots\ldots\ldots\ldots \quad [14]$$

For round work this equation becomes

$$P = \pi \times d \times t \times S \quad \ldots\ldots\ldots\ldots \quad [15]$$

where $d$ = mean diameter of shell, in.

**LOADS IN SHALLOW DRAWS.**—The load in a shallow draw (blank diam. = $D$) is

$$P = \pi \times d \times t \times S \times \{(D/d) - 1 + C\} \quad \ldots\ldots\ldots \quad [16]$$

in which notation is the same as for equation [15], and the constant $C$ = 0.3.

**LOADS IN DRAWING RECTANGULAR SHAPES.**—

$$P = t \times S \times \{2\,\pi\,r\,C_1 + (L \times C_2)\} \quad \ldots\ldots\ldots \quad [17]$$

where $r$ = corner radius of rectangular shell, in.; $t$ = thickness of metal, in.; $S$ = nominal ultimate tensile strength of the metal (see Table 1); $L$ = total length of the straight sides of the shell, in.; $C_1$ = 0.5 for a very low shell, up to, say, 2 for a shell of a depth of five or six times the corner radius $r$; $C_2$ = 0.2 for easy draw radius, ample clearance and no holding pressure, or, say, 0.3 for similar free flow and a normal blank holding pressure of about $^1/_3$ $P$, or a maximum of 1 with the metal clamped too tightly to flow.

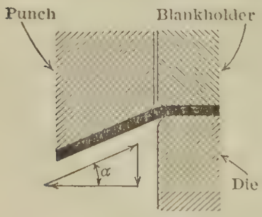

Fig. 7. Loads in Shallow Stretching

**LOADS IN STRETCHING** or shallow forming operations, in which the metal is gripped tightly around the edge, are determined by

$$P = L\,t\,S_e \sin\alpha \quad \ldots\ldots\ldots \quad [18]$$

where $P$ = punch pressure, lb.; $L$ = punch periphery, in.; $S_e$ = elastic limit or yield point of metal, lb. per sq. in. (see Table 1). $\alpha$ = angle of incidence (see Fig. 7).

**REDUCTIONS PER DRAW.**—In drawing round shells, the following figures can be used for first reductions:

| | | | | | |
|---|---|---|---|---|---|
| Percent reduction in first draw, $(D - d)/d$...... | 30 | 35 | 40 | 45 | 47.5 |
| Approx. min. blank thickness, percent of its diameter, $(t/D)$ | | | | | |
| Double-action.......................... | 0.15 | 0.2 | 0.3 | 0.4 | 0.5 |
| Single-action.......................... | .. | 1.5 | 1.7 | 2.0 | 2.5 |

The reductions in succeeding operations are as follows: For each succeeding double-action operation, a 30% reduction is the common limit. For each succeeding single-action operation, a 25% reduction is the normal maximum, although reductions as high as 37% have been recorded with heavy metal and two-step dies.

**RULE FOR DEPTH OF DRAW FOR RECTANGULAR SHELLS.**—In drawing rectangular shells, the limit depth per operation is about four to six times the corner radius, because of distribution of surplus metal from the corner area into the adjacent wall area.

**BLANK-HOLDING PRESSURES.—Rectangular Shells.**—For large rectangular work the drawing pressure is relatively lower than for round work, but the allowance for blank-holding may run as high as 100% of the drawing pressure.

**Shallow Stretching Work.**—In stretching shallow shapes the metal must be gripped tightly around the edge, and blank-holding pressure may be as high as two or three times the relatively low drawing pressure.

**Round Shells.**—The pressure required to hold the blank flat while deep drawing round work varies from zero in case of relatively thick blanks, up to about $^1/_3$ or $^1/_2$ of the drawing pressure. When the blank-holding pressure exceeds $^1/_3$ of the drawing pressure it is best to use draw beads to permit the blank-holding pressure to be decreased. Tapered or domed shapes must be classed with shallow stretching above.

**BLANK-HOLDING DEVICES.**—Mechanical grip on the edges of the blank, as obtained in toggle and cam presses, is, for practical purposes, positive and non-yielding. It is particularly desirable for domed and tapered shallow shapes and for shallow stretching, especially where draw beads must be stamped to help grip the metal.

Rubber bumper pressures may be approximated at 10 lb. per sq. in. of area, per 1% of compression. Actually, by test, the pressure rises on a slightly accelerating curve, and varies with the composition, temperature and age of the rubber. (See pp. 5–27 to 5–31.) Spring pressures may be figured from spring tables elsewhere in this book. (See pp. 10–03 to 10–22.) More uniform pressures and more convenient control of pressure, are obtained with die cushions. Pneumatic die cushions may be selected on a basis of 100 lb. per sq. in. of piston area, maximum; and hydro-pneumatic cushions for drawing purposes, on a basis of about 500 lb. per sq. in.

**REDUCTION PER ANNEAL.**—The total percent reduction accomplished between annealing operations, whether in one or more operations, should be kept within the "percent reduction in area" found in tensile tests of the annealed material (Table 1).

**WORK DONE IN DRAWING OPERATIONS.**—Let $W_e$ = working energy, in.-tons = (working pressure × working distance); $P$ = drawing pressure, tons, plus blank-holding pressure in cases of spring, rubber or air drawing attachments; $h$ = shell height, in.; $C$ = constant = 0.60 to 0.80 to allow for the gradual rise of pressure; Then

$$W_e = P \times h \times C \qquad [19]$$

No allowance is made for friction losses.

**SQUEEZING OPERATIONS.**—The load $P$, lb., in squeezing operations is

$$P = A \times S \qquad [20]$$

where $A$ = projected area, sq. in., of the surface of a job and flash which is in contact at the completion of the squeeze; $S$ = compressive stress, lb. per sq. in. (see Table 1).

**Work Done in Squeezing** is

$$W_e = P \times l \qquad [21]$$

where $P$ = average pressure, tons; $l$ = working distance, in.

In figuring work in squeezing operations the distance the punch travels, from its first contact with the slug or blank to bottom of stroke, includes the amount the metal is moved and the amount of deflection in the press and tools. The working pressure initially is the yield point of the metal *times* the initial area of contact. It will rise gradually as the material strain-hardens and the contact area *increases*. In the case of closed dies there is likely to be a further sharp rise when the die fills, its duration and extent being determined by the amount necessary to stretch the press to get over bottom center. The metal practically is incompressible.

**UPSETTING RULES.**—1. The limit of length of unsupported stock that can be gathered or upset in one blow, without injurious buckling, is not more than three times the diameter of the bar. 2. Lengths of stock more than three times the diameter of the bar can be upset successfully in one blow (in a closed die), provided the diameter of the upset made is not more than one and one-half times the diameter of the bar. 3. In an upset which requires more than three diameters of stock in length, and in which the diameter of the upset is one and one-half times the diameter of the bar, the amount of unsupported stock beyond the face of the die must not exceed one diameter of the stock.

**BULK MODULUS** $B$ in Table 1 is of use in determining the elastic change of volume under a stress $S$, lb. per sq. in. If $V_1$, $V_2$ = respectively, larger and smaller volumes,

$$(V_1 - V_2) = (S \times V_2)/B \quad \text{. . . . . . . . . .} \quad [22]$$

### Table 3 —Approximate Blank Diameters for Cylindrical Shells

(E. W. Bliss Co., Brooklyn, N. Y.)

| Diam. of Shell, in. | Height of Shell, inches | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/4 | 2 1/2 | 2 3/4 | 3 | 3 1/4 | 3 1/2 | 3 3/4 | 4 |
| 1/4 | 0.75 | 0.83 | 0.90 | 0.97 | 1.03 | 1.09 | 1.15 | 1.25 | 1.34 | 1.43 | 1.52 | 1.59 | 1.67 | 1.75 | 1.82 | 1.89 | 1.95 | 2.02 |
| 3/8 | 0.94 | 1.04 | 1.13 | 1.21 | 1.28 | 1.34 | 1.42 | 1.55 | 1.67 | 1.78 | 1.88 | 1.98 | 2.07 | 2.16 | 2.24 | 2.32 | 2.40 | 2.48 |
| 1/2 | 1.12 | 1.22 | 1.32 | 1.41 | 1.50 | 1.58 | 1.66 | 1.80 | 1.93 | 2.07 | 2.18 | 2.29 | 2.39 | 2.50 | 2.60 | 2.69 | 2.78 | 2.88 |
| 5/8 | 1.28 | 1.40 | 1.50 | 1.60 | 1.70 | 1.79 | 1.88 | 2.04 | 2.19 | 2.32 | 2.46 | 2.58 | 2.69 | 2.80 | 2.92 | 3.02 | 3.12 | 3.23 |
| 3/4 | 1.44 | 1.56 | 1.67 | 1.78 | 1.89 | 1.98 | 2.08 | 2.25 | 2.41 | 2.56 | 2.70 | 2.84 | 2.96 | 3.09 | 3.21 | 3.32 | 3.43 | 3.54 |
| 7/8 | 1.59 | 1.72 | 1.84 | 1.95 | 2.06 | 2.16 | 2.26 | 2.45 | 2.62 | 2.78 | 2.94 | 3.08 | 3.22 | 3.38 | 3.50 | 3.63 | 3.75 | 3.84 |
| 1 | 1.73 | 1.87 | 2.00 | 2.12 | 2.23 | 2.34 | 2.45 | 2.64 | 2.82 | 3.00 | 3.16 | 3.31 | 3.46 | 3.61 | 3.74 | 3.87 | 4.00 | 4.13 |
| 1 1/8 | 1.87 | 2.02 | 2.15 | 2.28 | 2.40 | 2.51 | 2.62 | 2.83 | 3.02 | 3.21 | 3.37 | 3.53 | 3.69 | 3.84 | 3.98 | 4.12 | 4.26 | 4.39 |
| 1 1/4 | 2.01 | 2.16 | 2.30 | 2.43 | 2.56 | 2.68 | 2.80 | 3.01 | 3.22 | 3.40 | 3.58 | 3.75 | 3.91 | 4.07 | 4.22 | 4.37 | 4.51 | 4.65 |
| 1 3/8 | 2.16 | 2.31 | 2.45 | 2.59 | 2.72 | 2.84 | 2.96 | 3.18 | 3.39 | 3.59 | 3.77 | 3.95 | 4.12 | 4.28 | 4.44 | 4.59 | 4.74 | 4.88 |
| 1 1/2 | 2.29 | 2.45 | 2.60 | 2.74 | 2.87 | 3.00 | 3.12 | 3.36 | 3.58 | 3.77 | 3.97 | 4.15 | 4.33 | 4.50 | 4.66 | 4.82 | 4.97 | 5.12 |
| 1 5/8 | 2.42 | 2.59 | 2.74 | 2.89 | 3.02 | 3.15 | 3.28 | 3.52 | 3.74 | 3.95 | 4.15 | 4.34 | 4.53 | 4.70 | 4.87 | 5.03 | 5.20 | 5.35 |
| 1 3/4 | 2.56 | 2.72 | 2.88 | 3.03 | 3.17 | 3.30 | 3.43 | 3.68 | 3.91 | 4.14 | 4.34 | 4.54 | 4.73 | 4.90 | 5.08 | 5.25 | 5.42 | 5.57 |
| 1 7/8 | 2.70 | 2.86 | 3.02 | 3.18 | 3.32 | 3.46 | 3.59 | 3.84 | 4.08 | 4.30 | 4.52 | 4.71 | 4.91 | 5.10 | 5.28 | 5.45 | 5.62 | 5.79 |
| 2 | 2.83 | 3.00 | 3.16 | 3.31 | 3.46 | 3.61 | 3.75 | 4.00 | 4.24 | 4.47 | 4.69 | 4.90 | 5.10 | 5.29 | 5.48 | 5.65 | 5.83 | 6.00 |
| 2 1/8 | 2.96 | 3.13 | 3.30 | 3.46 | 3.61 | 3.75 | 3 89 | 4.16 | 4.40 | 4.64 | 4.86 | 5.08 | 5.28 | 5.48 | 5.67 | 5.85 | 6.03 | 6.21 |
| 2 1/4 | 3.09 | 3.27 | 3.44 | 3.60 | 3.75 | 3.90 | 4.04 | 4.31 | 4.56 | 4.80 | 5.03 | 5.25 | 5.46 | 5.66 | 5.86 | 6.05 | 6.23 | 6.41 |
| 2 3/8 | 3.22 | 3.40 | 3.57 | 3.74 | 3.89 | 4.04 | 4.18 | 4.46 | 4.72 | 4.96 | 5.20 | 5.42 | 5.64 | 5 84 | 6.04 | 6.24 | 6.42 | 6.61 |
| 2 1/2 | 3.35 | 3.54 | 3.71 | 3.87 | 4.03 | 4.18 | 4.33 | 4.61 | 4.87 | 5.12 | 5.36 | 5.59 | 5.81 | 6.02 | 6.22 | 6.42 | 6.61 | 6.80 |
| 2 5/8 | 3.48 | 3.67 | 3.84 | 4.01 | 4.17 | 4.32 | 4.47 | 4.76 | 5.03 | 5.28 | 5.52 | 5.76 | 5.98 | 6.20 | 6.40 | 6.61 | 6.80 | 6.99 |
| 2 3/4 | 3.61 | 3.80 | 3.98 | 4.15 | 4.31 | 4.47 | 4.62 | 4.90 | 5.18 | 5.44 | 5.68 | 5.92 | 6.15 | 6.37 | 6.58 | 6.79 | 6.99 | 7.18 |
| 2 7/8 | 3.75 | 3.93 | 4.11 | 4.28 | 4.45 | 4.60 | 4.76 | 5.05 | 5.33 | 5.59 | 5.84 | 6.08 | 6.32 | 6.54 | 6.76 | 6.96 | 7.17 | 7.37 |
| 3 | 3.87 | 4.06 | 4.24 | 4.42 | 4.58 | 4.74 | 4.90 | 5.20 | 5.48 | 5.74 | 6.00 | 6.25 | 6.48 | 6.71 | 6.93 | 7.14 | 7.35 | 7.55 |
| 3 1/8 | 4.00 | 4.19 | 4.38 | 4.55 | 4.72 | 4.88 | 5.04 | 5.34 | 5.63 | 5.90 | 6.16 | 6.40 | 6.64 | 6.87 | 7.10 | 7.32 | 7.52 | 7.73 |
| 3 1/4 | 4.13 | 4.32 | 4.51 | 4.68 | 4.85 | 5.02 | 5.18 | 5.48 | 5.77 | 6.05 | 6.31 | 6.56 | 6.80 | 7.04 | 7.26 | 7.48 | 7.70 | 7.91 |
| 3 3/8 | 4.26 | 4.45 | 4.64 | 4.82 | 4.99 | 5.15 | 5.31 | 5.62 | 5.92 | 6.19 | 6.47 | 6.72 | 6.96 | 7.20 | 7.43 | 7.66 | 7.87 | 8.08 |
| 3 1/2 | 4.39 | 4.58 | 4.77 | 4.95 | 5.12 | 5.29 | 5.45 | 5.77 | 6.06 | 6.35 | 6.61 | 6.87 | 7.12 | 7.37 | 7.60 | 7.83 | 8.05 | 8.26 |
| 3 5/8 | 4.51 | 4.71 | 4.90 | 5.08 | 5.26 | 5.43 | 5.59 | 5 90 | 6.21 | 6.49 | 6.76 | 7.03 | 7.28 | 7.52 | 7.76 | 7.99 | 8.21 | 8.43 |
| 3 3/4 | 4.64 | 4.84 | 5.03 | 5.21 | 5.39 | 5.56 | 5.73 | 6.04 | 6.35 | 6.63 | 6.91 | 7.18 | 7.43 | 7.68 | 7.92 | 8.16 | 8.38 | 8.60 |
| 3 7/8 | 4.77 | 4.97 | 5.16 | 5.34 | 5.52 | 5.69 | 5.86 | 6.18 | 6.49 | 6.78 | 7.06 | 7.33 | 7.59 | 7.84 | 8.08 | 8.32 | 8.55 | 8.77 |
| 4 | 4.90 | 5.10 | 5.29 | 5.48 | 5.66 | 5.83 | 6.00 | 6.32 | 6.63 | 6.92 | 7.21 | 7.48 | 7.74 | 8.00 | 8.24 | 8.48 | 8.71 | 8.94 |
| 4 1/8 | 5.02 | 5.22 | 5.42 | 5 61 | 5.79 | 5.96 | 6.13 | 6.46 | 6.77 | 7.07 | 7.36 | 7.63 | 7.89 | 8.15 | 8.40 | 8.64 | 8.88 | 9.11 |
| 4 1/4 | 5.15 | 5.35 | 5.55 | 5.74 | 5.92 | 6.09 | 6.27 | 6.60 | 6.91 | 7.21 | 7.50 | 7.78 | 8.05 | 8.31 | 8.56 | 8.80 | 9.04 | 9.27 |
| 4 3/8 | 5.28 | 5.48 | 5.68 | 5.87 | 6.05 | 6.23 | 6.40 | 6.73 | 7.05 | 7.35 | 7.64 | 7.93 | 8.20 | 8.46 | 8.71 | 8.96 | 9.20 | 9.44 |
| 4 1/2 | 5.40 | 5.61 | 5.81 | 6.00 | 6.18 | 6.36 | 6.53 | 6.87 | 7.19 | 7.50 | 7.79 | 8.07 | 8.35 | 8.61 | 8 87 | 9.12 | 9.36 | 9.60 |
| 4 5/8 | 5.53 | 5.74 | 5.93 | 6.13 | 6.31 | 6.49 | 6.67 | 7.01 | 7.33 | 7.64 | 7.93 | 8.22 | 8.50 | 8.76 | 9.02 | 9.28 | 9.52 | 9.76 |
| 4 3/4 | 5.66 | 5.86 | 6.06 | 6.26 | 6.44 | 6.62 | 6.80 | 7.14 | 7.47 | 7.78 | 8.08 | 8.37 | 8.64 | 8.91 | 9.18 | 9.43 | 9.68 | 9.92 |
| 4 7/8 | 5.78 | 5.99 | 6.19 | 6.39 | 6.57 | 6.76 | 6.93 | 7.28 | 7.60 | 7.92 | 8.22 | 8.51 | 8.79 | 9.06 | 9.33 | 9.59 | 9.84 | 10.08 |
| 5 | 5.91 | 6.12 | 6.32 | 6.52 | 6.70 | 6.89 | 7.07 | 7.41 | 7.74 | 8.06 | 8.36 | 8.66 | 8.94 | 9.21 | 9.48 | 9.74 | 10.00 | 10.24 |
| 5 1/4 | 6.17 | 6.37 | 6.58 | 6.77 | 6.96 | 7.15 | 7.33 | 7.68 | 8.02 | 8.34 | 8.65 | 8.94 | 9.23 | 9.51 | 9.78 | 10.05 | 10.31 | 10.56 |
| 5 1/2 | 6.42 | 6.63 | 6.83 | 7.03 | 7.22 | 7.41 | 7.60 | 7.95 | 8.29 | 8.61 | 8.93 | 9.23 | 9.52 | 9.81 | 10.08 | 10.35 | 10.62 | 10.87 |
| 5 3/4 | 6.67 | 6.88 | 7.09 | 7.29 | 7.48 | 7.67 | 7.86 | 8.22 | 8.56 | 8.89 | 9.20 | 9.51 | 9.81 | 10.10 | 10.38 | 10.65 | 10.92 | 11.18 |
| 6 | 6 92 | 7.14 | 7.34 | 7.55 | 7.74 | 7.93 | 8.12 | 8.48 | 8.83 | 9.16 | 9.48 | 9.79 | 10.09 | 10.39 | 10.67 | 10.95 | 11.22 | 11.48 |
| 6 1/4 | 7.18 | 7.39 | 7.60 | 7.80 | 8.00 | 8.19 | 8.38 | 8.75 | 9.10 | 9.44 | 9.76 | 10.07 | 10.38 | 10.68 | 10.96 | 11.25 | 11.52 | 11.79 |
| 6 1/2 | 7.43 | 7.64 | 7.85 | 8.06 | 8.26 | 8.45 | 8.64 | 9.01 | 9.36 | 9.71 | 10.03 | 10.35 | 10.66 | 10.96 | 11.25 | 11.54 | 11.82 | 12.09 |
| 6 3/4 | 7.68 | 7.90 | 8.11 | 8.31 | 8.51 | 8.71 | 8.90 | 9.27 | 9.63 | 9.97 | 10.31 | 10.63 | 10.94 | 11.25 | 11.54 | 11.83 | 12.11 | 12.39 |
| 7 | 7.93 | 8.15 | 8.36 | 8.57 | 8.77 | 8.97 | 9.16 | 9.53 | 9.90 | 10.24 | 10.58 | 10.90 | 11.22 | 11.53 | 11.83 | 12.12 | 12.40 | 12.68 |
| 7 1/4 | 8.18 | 8.40 | 8.62 | 8 82 | 9.03 | 9.22 | 9.42 | 9.80 | 10.16 | 10.51 | 10.85 | 11.18 | 11.50 | 11.81 | 12.11 | 12.41 | 12.70 | 12.98 |
| 7 1/2 | 8.44 | 8.66 | 8.87 | 9.08 | 9.28 | 9.48 | 9.68 | 10.06 | 10.42 | 10.78 | 11.12 | 11.45 | 11.77 | 12.09 | 12.39 | 12.69 | 12.99 | 13.27 |
| 7 3/4 | 8.69 | 8.91 | 9.12 | 9.33 | 9.54 | 9.74 | 9.94 | 10.32 | 10.69 | 11.04 | 11.39 | 11.72 | 12.05 | 12.37 | 12.68 | 12.98 | 13.27 | 13.56 |
| 8 | 8.94 | 9.16 | 9.38 | 9.59 | 9.79 | 10.00 | 10.19 | 10.58 | 10.95 | 11.31 | 11.66 | 11.99 | 12.32 | 12.64 | 12.96 | 13.26 | 13.56 | 13.85 |
| 8 1/4 | 9.19 | 9.41 | 9 63 | 9 84 | 10.05 | 10.25 | 10.45 | 10.84 | 11.21 | 11.57 | 11.92 | 12.27 | 12.60 | 12.92 | 13.23 | 13.54 | 13.84 | 14.14 |
| 8 1/2 | 9.44 | 9.66 | 9.88 | 10.10 | 10 30 | 10.51 | 10.71 | 11.10 | 11.47 | 11.84 | 12.19 | 12.53 | 12.87 | 13.20 | 13.51 | 13.82 | 14.13 | 14.43 |
| 8 3/4 | 9.69 | 9.92 | 10.13 | 10.35 | 10.56 | 10.76 | 10.96 | 11.36 | 11.73 | 12.10 | 12.46 | 12.80 | 13.14 | 13.47 | 13.79 | 14.10 | 14.41 | 14.71 |
| 9 | 9.95 | 10.17 | 10.39 | 10.60 | 10.81 | 11.02 | 11.21 | 11.61 | 11.99 | 12.36 | 12.72 | 13.07 | 13.41 | 13.74 | 14.07 | 14.38 | 14.69 | 14.99 |
| 9 1/4 | 10.20 | 10.42 | 10.64 | 10.85 | 11.07 | 11.27 | 11.48 | 11.87 | 12.26 | 12.63 | 12.99 | 13.34 | 13.68 | 14.02 | 14.34 | 14.66 | 14.97 | 15.28 |
| 9 1/2 | 10.45 | 10.67 | 10.89 | 11.11 | 11.32 | 11.53 | 11.73 | 12.13 | 12 52 | 12.89 | 13.25 | 13.61 | 13.95 | 14.29 | 14.61 | 14.94 | 15.25 | 15.56 |
| 9 3/4 | 10.70 | 10.92 | 11.14 | 11.36 | 11.57 | 11.78 | 11.99 | 12.39 | 12.77 | 13.15 | 13.52 | 13.87 | 14.22 | 14.56 | 14.89 | 15.21 | 15.53 | 15.84 |
| 10 | 10.95 | 11.18 | 11.40 | 11.61 | 11.83 | 12.04 | 12.24 | 12.64 | 13.03 | 13.41 | 13.78 | 14.14 | 14.49 | 14.83 | 15.16 | 15.49 | 15.81 | 16.12 |
| 10 1/4 | 11.20 | 11.43 | 11.65 | 11.87 | 12.08 | 12.29 | 12.50 | 12 90 | 13.29 | 13.67 | 14.04 | 14.40 | 14.75 | 15.10 | 15.43 | 15.76 | 16.08 | 16.40 |
| 10 1/2 | 11.45 | 11.68 | 11.90 | 12.12 | 12.33 | 12.54 | 12.75 | 13.16 | 13.55 | 13.93 | 14.30 | 14.66 | 15.02 | 15.36 | 15.70 | 16.03 | 16.36 | 16.68 |
| 10 3/4 | 11.70 | 11.93 | 12.15 | 12.37 | 12.59 | 12.80 | 13.01 | 13.41 | 13.81 | 14.19 | 14.56 | 14.93 | 15.29 | 15.63 | 15.97 | 16.31 | 16.63 | 16.95 |
| 11 | 11.95 | 12.18 | 12.40 | 12.62 | 12.84 | 13.05 | 13.26 | 13.67 | 14.07 | 14.45 | 14.83 | 15.19 | 15.55 | 15.90 | 16.24 | 16.58 | 16.91 | 17.22 |
| 11 1/4 | 12.20 | 12.43 | 12.67 | 12.91 | 13.09 | 13.30 | 13 52 | 13.93 | 14.33 | 14.71 | 15.08 | 15.46 | 15.82 | 16.17 | 16.52 | 16.84 | 17.17 | 17.50 |
| 11 1/2 | 12.45 | 12.65 | 12.90 | 13.11 | 13.35 | 13.56 | 13.77 | 14.18 | 14.58 | 14.97 | 15.35 | 15.72 | 16.08 | 16.43 | 16.78 | 17.11 | 17.45 | 17.78 |
| 11 3/4 | 12.66 | 12.92 | 13.16 | 13.39 | 13.59 | 13.82 | 14.03 | 14.44 | 14.84 | 15.22 | 15.60 | 15.98 | 16.34 | 16.70 | 17.05 | 17.39 | 17.73 | 18.05 |
| 12 | 12 96 | 13.20 | 13.42 | 13.63 | 13.86 | 14.07 | 14.28 | 14.70 | 15.10 | 15.50 | 15.88 | 16.24 | 16.61 | 16.96 | 17.32 | 17.66 | 18.00 | 18.33 |

**THERMAL COEFFICIENT OF EXPANSION** $f$, in Table 1, is used to estimate changes in lineal dimensions with changes in the temperature.

$$L_2 = L_1 + \{L_1 \times f \times (T_2 - T_1)\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

where $L_1$, $L_2$ = respectively, initial and final lengths, in.; $T_1$, $T_2$ = respectively, initial and final temperatures, deg. F.

**Table 3.—Approximate Blank Diameters for Cylindrical Shells—***Continued*

| Diam. of Shell, in. | Height of Shell, inches | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 4 1/4 | 4 1/2 | 4 3/4 | 5 | 5 1/2 | 6 | 6 1/2 | 7 | 7 1/2 | 8 | 8 1/2 | 9 | 9 1/2 | 10 | 10 1/2 | 11 | 11 1/2 | 12 |
| 1/4 | 2.08 | 2.13 | 2.20 | 2.25 | 2.36 | 2.46 | 2.56 | 2.66 | 2.75 | 2.83 | 2.93 | 3.01 | 3.09 | 3.17 | 3.25 | 3.33 | 3.40 | 3.48 |
| 3/8 | 2.55 | 2.63 | 2.70 | 2.76 | 2.90 | 3.03 | 3.15 | 3.27 | 3.38 | 3.48 | 3.59 | 3.70 | 3.80 | 3 90 | 3.98 | 4.08 | 4.17 | 4.26 |
| 1/2 | 2.96 | 3.04 | 3.12 | 3.20 | 3.35 | 3.50 | 3.64 | 3.77 | 3.90 | 4.03 | 4.15 | 4.27 | 4.39 | 4.50 | 4.61 | 4.71 | 4.82 | 4.93 |
| 5/8 | 3.32 | 3.42 | 3.50 | 3.59 | 3.76 | 3.92 | 4.08 | 4.23 | 4.38 | 4.52 | 4.65 | 4.78 | 4.91 | 5.04 | 5.16 | 5.28 | 5.39 | 5.51 |
| 3/4 | 3.65 | 3.75 | 3.85 | 3.94 | 4.13 | 4.30 | 4.48 | 4.64 | 4.80 | 4.96 | 5.10 | 5.25 | 5.39 | 5.53 | 5.66 | 5.79 | 5.92 | 6.05 |
| 7/8 | 3.95 | 4.06 | 4.17 | 4.27 | 4.47 | 4.66 | 4.84 | 5.03 | 5.19 | 5.36 | 5.52 | 5.68 | 5.83 | 5.98 | 6.12 | 6.26 | 6.40 | 6.53 |
| 1 | 4.24 | 4.35 | 4.46 | 4.58 | 4.79 | 5 00 | 5.19 | 5.39 | 5.57 | 5.74 | 5.92 | 6.08 | 6.24 | 6.40 | 6.56 | 6.71 | 6.86 | 7.00 |
| 1 1/8 | 4.51 | 4.63 | 4.75 | 4.87 | 5.10 | 5.31 | 5.52 | 5.72 | 5.92 | 6.10 | 6.28 | 6.46 | 6.63 | 6.80 | 6.96 | 7.12 | 7.28 | 7.43 |
| 1 1/4 | 4.78 | 4.90 | 5.03 | 5.15 | 5.39 | 5.61 | 5.84 | 6.04 | 6.25 | 6.45 | 6.64 | 6.82 | 7.00 | 7.18 | 7.35 | 7.52 | 7.68 | 7.84 |
| 1 3/8 | 5.02 | 5.16 | 5.29 | 5.42 | 5.67 | 5.90 | 6.13 | 6.35 | 6.57 | 6.78 | 6.97 | 7.16 | 7.35 | 7.54 | 7.73 | 7.89 | 8.07 | 8.24 |
| 1 1/2 | 5.26 | 5.40 | 5.54 | 5 68 | 5 93 | 6.19 | 6.42 | 6 65 | 6.87 | 7.09 | 7.30 | 7.50 | 7.70 | 7.89 | 8.08 | 8.26 | 8.44 | 8.62 |
| 1 3/8 | 5.50 | 5.64 | 5.78 | 5.92 | 6.20 | 6.45 | 6.70 | 6.94 | 7.17 | 7.39 | 7.61 | 7.82 | 8.02 | 8.22 | 8.42 | 8.61 | 8.80 | 8.98 |
| 1 3/4 | 5.72 | 5.88 | 6.02 | 6.17 | 6.45 | 6.71 | 6.97 | 7.22 | 7 46 | 7.69 | 7.91 | 8.13 | 8.34 | 8.55 | 8.75 | 8.95 | 9.14 | 9.33 |
| 1 7/8 | 5.95 | 6.10 | 6.25 | 6.41 | 6.69 | 6.96 | 7.23 | 7.49 | 7.73 | 7.97 | 8.20 | 8.43 | 8.64 | 8.86 | 9.07 | 9.27 | 9.48 | 9.67 |
| 2 | 6.16 | 6.32 | 6.48 | 6.63 | 6.93 | 7.21 | 7.49 | 7.75 | 8.00 | 8.25 | 8.48 | 8.72 | 8.94 | 9.17 | 9.38 | 9.59 | 9.80 | 10.00 |
| 2 1/8 | 6 38 | 6.54 | 6.70 | 6.86 | 7.16 | 7.45 | 7.73 | 8.00 | 8.26 | 8.52 | 8.76 | 9.00 | 9.23 | 9.46 | 9.68 | 9.90 | 10.11 | 10.32 |
| 2 1/4 | 6.58 | 6.75 | 6.91 | 7.08 | 7.39 | 7.68 | 7.97 | 8.25 | 8.52 | 8.78 | 9.03 | 9.28 | 9.52 | 9.75 | 9.98 | 10.21 | 10.42 | 10.63 |
| 2 3/8 | 6.78 | 6.96 | 7.13 | 7.29 | 7.61 | 7.91 | 8.21 | 8.49 | 8.77 | 9.03 | 9.29 | 9.55 | 9.79 | 10.03 | 10.26 | 10.49 | 10.72 | 10.94 |
| 2 1/2 | 6.98 | 7.16 | 7.33 | 7.50 | 7.82 | 8.14 | 8.44 | 8.73 | 9.01 | 9.29 | 9.55 | 9.81 | 10.06 | 10.31 | 10.55 | 10.78 | 11.01 | 11.24 |
| 2 5/8 | 7.18 | 7.36 | 7.53 | 7.71 | 8.04 | 8.36 | 8.67 | 8.97 | 9.25 | 9.53 | 9.80 | 10.07 | 10.33 | 10.58 | 10.82 | 11.06 | 11.30 | 11.53 |
| 2 3/4 | 7.37 | 7.55 | 7.73 | 7.91 | 8.25 | 8.58 | 8.89 | 9.20 | 9.49 | 9.78 | 10.05 | 10.32 | 10.59 | 10.84 | 11.09 | 11.33 | 11.58 | 11.81 |
| 2 7/8 | 7.56 | 7.75 | 7.93 | 8.11 | 8.46 | 8.79 | 9.11 | 9.42 | 9.72 | 10.01 | 10.29 | 10.57 | 10.84 | 11.11 | 11.35 | 11.61 | 11.85 | 12.09 |
| 3 | 7.75 | 7.94 | 8.12 | 8.30 | 8.66 | 9.00 | 9.33 | 9.64 | 9.95 | 10.25 | 10.53 | 10.82 | 11.09 | 11.36 | 11.62 | 11.87 | 12.12 | 12.37 |
| 3 1/8 | 7.93 | 8.12 | 8.31 | 8.50 | 8.86 | 9.20 | 9.54 | 9.86 | 10.17 | 10.48 | 10.77 | 11.06 | 11.34 | 11.61 | 11.88 | 12.13 | 12.39 | 12.64 |
| 3 1/4 | 8 11 | 8.31 | 8.50 | 8.69 | 9.06 | 9.41 | 9.75 | 10.08 | 10.40 | 10.70 | 11.00 | 11.29 | 11.58 | 11.86 | 12.13 | 12.39 | 12.65 | 12.90 |
| 3 3/8 | 8.29 | 8.49 | 8.69 | 8.88 | 9.25 | 9.61 | 9.96 | 10.29 | 10.61 | 10.93 | 11.23 | 11.53 | 11.82 | 12.10 | 12.37 | 12.64 | 12.91 | 13.17 |
| 3 1/2 | 8.47 | 8.68 | 8.87 | 9.06 | 9.45 | 9.81 | 10.16 | 10.50 | 10.83 | 11.15 | 11.46 | 11.76 | 12.05 | 12.34 | 12.62 | 12.89 | 13.16 | 13.43 |
| 3 5/8 | 8.65 | 8.85 | 9.06 | 9.25 | 9.64 | 10 00 | 10.36 | 10.71 | 11.04 | 11.36 | 11.68 | 11.98 | 12.28 | 12.57 | 12.86 | 13.14 | 13.41 | 13.68 |
| 3 3/4 | 8 82 | 9.03 | 9.23 | 9.44 | 9.82 | 10.20 | 10.56 | 10.91 | 11.25 | 11.58 | 11.90 | 12.21 | 12.51 | 12.80 | 13.09 | 13.37 | 13.65 | 13.93 |
| 3 7/8 | 8.99 | 9.20 | 9.41 | 9.61 | 10.01 | 10.39 | 10.76 | 11.11 | 11.45 | 11.79 | 12.11 | 12.43 | 12.74 | 13.04 | 13.33 | 13.62 | 13.90 | 14.18 |
| 4 | 9.16 | 9.38 | 9.59 | 9.79 | 10.20 | 10.58 | 10.95 | 11.31 | 11.66 | 12.00 | 12.33 | 12.65 | 12.96 | 13.26 | 13.56 | 13.85 | 14.14 | 14.42 |
| 4 1/8 | 9.33 | 9.55 | 9.76 | 9.97 | 10.38 | 10.76 | 11.14 | 11.51 | 11.86 | 12.20 | 12.54 | 12.86 | 13.18 | 13.49 | 13.79 | 14.09 | 14.38 | 14.66 |
| 4 1/4 | 9.50 | 9.72 | 9.94 | 10.15 | 10.56 | 10.95 | 11.33 | 11.71 | 12.06 | 12.41 | 12.75 | 13.07 | 13.40 | 13.72 | 14.02 | 14.32 | 14.61 | 14.90 |
| 4 3/8 | 9.67 | 9.89 | 10.11 | 10.32 | 10.74 | 11.14 | 11.52 | 11.90 | 12.26 | 12.61 | 12.95 | 13.29 | 13.61 | 13.93 | 14.24 | 14.54 | 14.84 | 15.13 |
| 4 1/2 | 9.83 | 10.06 | 10.28 | 10.50 | 10.92 | 11.32 | 11.71 | 12.09 | 12.45 | 12.81 | 13.16 | 13.50 | 13.82 | 14.15 | 14.45 | 14.77 | 15.07 | 15.37 |
| 4 5/8 | 10.00 | 10.22 | 10.45 | 10.67 | 11.09 | 11.50 | 11.90 | 12.28 | 12.65 | 13.01 | 13.36 | 13.70 | 14.04 | 14.36 | 14.68 | 14.99 | 15.30 | 15.60 |
| 4 3/4 | 10.16 | 10.39 | 10.62 | 10.84 | 11.27 | 11.68 | 12.08 | 12.47 | 12.84 | 13.21 | 13.56 | 13.91 | 14.25 | 14.57 | 14.90 | 15.22 | 15.52 | 15.83 |
| 4 7/8 | 10.32 | 10.55 | 10.78 | 11.01 | 11.44 | 11.86 | 12.26 | 12.65 | 13.03 | 13.41 | 13.76 | 14.11 | 14.45 | 14.79 | 15.11 | 15.43 | 15.74 | 16.05 |
| 5 | 10.48 | 10.72 | 10.95 | 11.18 | 11.61 | 12.04 | 12.45 | 12.84 | 13.23 | 13.60 | 13.96 | 14.32 | 14.66 | 15.00 | 15.33 | 15.65 | 15.97 | 16.28 |
| 5 1/4 | 10.80 | 11.04 | 11.28 | 11.51 | 11.96 | 12.39 | 12.81 | 13.21 | 13.60 | 13.98 | 14.35 | 14.71 | 15 07 | 15.41 | 15.75 | 16.08 | 16.40 | 16.72 |
| 5 1/2 | 11.12 | 11.36 | 11.60 | 11.84 | 12.29 | 12.73 | 13.16 | 13.57 | 13.97 | 14.36 | 14.73 | 15.11 | 15.47 | 15.82 | 16.16 | 16.50 | 16.83 | 17.15 |
| 5 3/4 | 11.43 | 11.68 | 11.92 | 12.16 | 12.63 | 13.07 | 13.51 | 13.93 | 14.33 | 14.73 | 15.11 | 15.49 | 15.86 | 16.21 | 16.57 | 16.91 | 17.25 | 17.58 |
| 6 | 11.74 | 12.00 | 12.24 | 12.48 | 12.96 | 13.41 | 13.85 | 14.28 | 14.69 | 15.10 | 15.49 | 15.87 | 16.24 | 16.61 | 16.97 | 17.32 | 17.66 | 18 00 |
| 6 1/4 | 12.05 | 12.31 | 12.56 | 12.80 | 13.28 | 13.74 | 14.19 | 14.63 | 15.05 | 15.46 | 15.86 | 16.25 | 16.63 | 17.00 | 17.36 | 17.72 | 18.07 | 18.41 |
| 6 1/2 | 12.36 | 12.62 | 12.87 | 13.12 | 13.61 | 14.08 | 14.53 | 14.97 | 15.40 | 15.82 | 16.22 | 16.62 | 17.00 | 17.38 | 17.75 | 18.11 | 18.47 | 18.82 |
| 6 3/4 | 12.66 | 12.92 | 13.18 | 13.43 | 13.93 | 14.40 | 14.86 | 15.31 | 15.75 | 16.17 | 16.58 | 16.98 | 17.38 | 17.76 | 18.14 | 18.50 | 18.87 | 19.22 |
| 7 | 12.96 | 13.23 | 13.49 | 13.74 | 14.24 | 14.73 | 15.19 | 15.65 | 16.09 | 16.52 | 16.94 | 17.34 | 17.74 | 18.13 | 18.52 | 18.89 | 19.26 | 19.62 |
| 7 1/4 | 13.25 | 13.52 | 13.79 | 14.05 | 14.56 | 15.05 | 15.52 | 15.98 | 16.43 | 16.86 | 17.29 | 17.70 | 18.11 | 18.51 | 18.89 | 19.27 | 19.64 | 20.01 |
| 7 1/2 | 13.55 | 13.82 | 14.09 | 14.36 | 14.87 | 15.37 | 15.85 | 16.31 | 16.77 | 17.21 | 17.64 | 18.06 | 18.47 | 18.87 | 19.26 | 19.65 | 20.03 | 20.40 |
| 7 3/4 | 13.84 | 14.12 | 14.39 | 14.66 | 15.18 | 15.68 | 16.17 | 16.64 | 17.10 | 17.55 | 17.98 | 18.41 | 18.82 | 19.23 | 19.63 | 20.02 | 20.40 | 20.78 |
| 8 | 14.14 | 14.42 | 14.69 | 14.96 | 15.49 | 15.99 | 16.49 | 16.97 | 17.43 | 17.88 | 18.33 | 18.76 | 19.18 | 19.59 | 19.99 | 20.39 | 20.78 | 21 16 |
| 8 1/4 | 14.43 | 14.71 | 14.99 | 15.26 | 15.79 | 16.31 | 16.80 | 17.29 | 17.76 | 18.22 | 18.66 | 19.10 | 19.53 | 19.95 | 20.36 | 20.76 | 21.15 | 21.54 |
| 8 1/2 | 14.72 | 15.00 | 15.28 | 15.56 | 16.10 | 16.62 | 17.12 | 17.61 | 18.09 | 18.55 | 19.00 | 19.44 | 19.88 | 20.30 | 20.71 | 21.11 | 21.51 | 21.91 |
| 8 3/4 | 15.00 | 15.29 | 15.58 | 15.86 | 16.40 | 16.92 | 17.43 | 17.93 | 18.41 | 18.88 | 19.34 | 19.78 | 20.22 | 20.65 | 21.07 | 21.48 | 21.88 | 22.28 |
| 9 | 15.29 | 15.58 | 15.87 | 16.15 | 16.70 | 17.23 | 17.74 | 18.24 | 18.73 | 19.20 | 19.67 | 20.12 | 20.56 | 21.00 | 21.42 | 21.84 | 22.24 | 22.64 |
| 9 1/4 | 15.58 | 15.87 | 16.16 | 16.44 | 17.00 | 17.53 | 18.05 | 18.56 | 19.05 | 19.53 | 20.00 | 20.45 | 20.90 | 21.34 | 21.77 | 22.19 | 22.60 | 23.00 |
| 9 1/2 | 15.86 | 16.16 | 16.45 | 16.74 | 17.29 | 17.83 | 18.36 | 18.87 | 19.37 | 19.85 | 20.32 | 20.79 | 21.24 | 21.68 | 22.11 | 22.54 | 22 96 | 23.37 |
| 9 3/4 | 16.14 | 16.44 | 16.74 | 17.03 | 17.59 | 18.14 | 18.66 | 19.18 | 19.68 | 20.17 | 20.65 | 21.12 | 21.57 | 22.02 | 22.46 | 22.89 | 23.31 | 23.72 |
| 10 | 16.43 | 16.73 | 17.02 | 17.31 | 17.88 | 18.43 | 18.97 | 19.49 | 20.00 | 20.49 | 20.97 | 21.44 | 21.90 | 22.36 | 22.80 | 23.24 | 23 66 | 24.08 |
| 10 1/4 | 16.71 | 17.01 | 17.31 | 17.60 | 18.18 | 18.73 | 19.27 | 19.80 | 20.31 | 20.81 | 21.29 | 21.77 | 22.23 | 22.69 | 23.14 | 23.58 | 24.01 | 24 43 |
| 10 1/2 | 16.99 | 17.29 | 17.59 | 17.89 | 18.47 | 19.03 | 19.57 | 20.10 | 20.62 | 21.12 | 21.61 | 22.09 | 22.56 | 23.02 | 23.47 | 23.92 | 24.35 | 24.78 |
| 10 3/4 | 17.27 | 17.58 | 17.88 | 18.18 | 18.76 | 19.32 | 19.87 | 20.40 | 20.93 | 21.43 | 21.93 | 22.41 | 22.89 | 23.35 | 23.81 | 24.25 | 24.69 | 25.13 |
| 11 | 17.55 | 17.86 | 18.16 | 18.46 | 19.05 | 19.62 | 20.17 | 20.71 | 21.23 | 21.74 | 22.24 | 22.73 | 23.21 | 23.68 | 24.14 | 24.59 | 25.04 | 25.47 |
| 11 1/4 | 17.81 | 18.12 | 18.44 | 18.74 | 19.33 | 19.90 | 20.46 | 21.01 | 21.54 | 22.05 | 22.55 | 23.06 | 23.54 | 24.00 | 24.47 | 24.92 | 25.37 | 25.82 |
| 11 1/2 | 18.09 | 18.41 | 18.71 | 19.02 | 19.63 | 20.20 | 20.76 | 21.30 | 21.84 | 22.36 | 22.87 | 23.36 | 23.85 | 24.33 | 24.80 | 25.25 | 25.70 | 26.15 |
| 11 3/4 | 18.38 | 18.70 | 19.01 | 19.32 | 19.90 | 20.50 | 21.05 | 21.61 | 22.14 | 22.67 | 23.18 | 23.68 | 24.17 | 24.66 | 25.13 | 25.58 | 26.04 | 26.49 |
| 12 | 18.64 | 18.97 | 19.28 | 19.59 | 20.20 | 20.77 | 21.35 | 21.90 | 22.45 | 22.97 | 23.50 | 23.99 | 24.48 | 24.97 | 25.45 | 25.93 | 26.37 | 26.84 |

**YOUNG'S MODULUS OF ELASTICITY** $E_m$ (Table 1) permits computations of the elastic elongation $e$, in inches per inch of length, for any metal under a stress of $S$ lb. per sq. in., by means of the equation $e = S/E_m$.

**SPRING-BACK.**—Two ways of overcoming spring-back are: 1. *Overbend* to get the required angle. The amount of overbend usually is determined by trial and error. 2. *Hit-home* or bottom at the end of the bend to "set" the metal. In effect, this operation is a coining or squeezing operation, and the loads should be figured as such. The squeeze sets up compressive stress sufficient to overcome remaining internal tensile stresses.

## 4. MACHINE SELECTION

**FACTORS GOVERNING MACHINE SELECTION.**—Several factors are involved in machine selection. First, obviously, stroke, space for the dies, position and type of frame relative to handling and discharge of work, *i.e.*, vertical, inclined or horizontal, straight-sided or gap frame. Next, 1, capacity of frame, shaft, gearing (tons pressure); 2, fly-wheel (energy, to prevent stalling); and 3, motor.

**TONNAGE RATINGS** vary considerably with details of design and the conservativeness of the builder. Tables 4 and 5 give general values which take into consideration, the type of shaft, the stroke, the gearing and the frame, based upon well-developed designs.

**FLY-WHEEL ENERGY.**—The fly-wheel must store up and deliver at the proper instant, the energy required to maintain the working pressure throughout the working distance. In the case of long working strokes the motor may be able to take a material portion of this load. Otherwise it just restores energy to the fly-wheel during the balance of the cycle by bringing it back to speed.

**Fly-wheel Slowdown** is arbitrarily figured up to 10% for continuous running equipment; up to 20% for intermittent operating conditions; and occasionally up to 40% which approaches stalling. The working energy $W_e$, inch-tons, delivered by the average design of fly-wheel at 10% slowdown may be taken as

$$W_e = N^2 \times D_f^2 \times w_f/5{,}250{,}000{,}000. \quad \ldots\ldots\ldots \quad [24]$$

where = $N$ rev. per min.; $D_f$ = diameter of fly-wheel, in.; $w_f$ = weight of fly-wheel, lb. For 20% slowdown multiply $W_e$ by 1.9.

**Table 4.—Approximate Capacities of Straight-side, Single-crank Presses**

(E. W. Bliss Co., Brooklyn, N. Y.)

| Stroke of Press, in. | | | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Approximate Capacity in Tons | | | | | | | | | | | | |
| Shaft Diam., in. | Type of Drive | At Bottom | At Mid-stroke | | | | | | | | | | | |
| 1 | Single | 3 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 1 1/2 | Single | 6 3/4 | 3 | 2 1/4 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2 | Single | 12 | 7 1/2 | 5 1/2 | 4 1/2 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 2 1/2 | Single | 19 | 14 | 11 | 8 1/2 | 7 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 | Single | 32 | 20 | 15 | 12 | 9 | 8 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 1/2 | Single | 44 | 31 1/2 | 23 1/2 | 19 | 16 | 14 | 11 1/2 | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 | Single | 58 | 44 | 36 | 30 | 24 | 18 | 13 | 9 | ..... | ..... | ..... | ..... | ..... |
| 4 1/2 | Single | 74 | 60 | 50 | 40 | 35 | 26 | 19 | 12 | ..... | ..... | ..... | ..... | ..... |
| 5 | Single | 93 | 73 | 64 | 56 | 47 | 35 | 28 | 22 | 18 | 14 | ..... | ..... | ..... |
| 6 | Single | 135 | ..... | 108 | 94 | 80 | 60 | 48 | 39 | 35 | 32 | 30 | 27 | ..... |
| 7 | Single | 190 | ..... | ..... | 135 | 122 | 98 | 78 | 69 | 62 | 46 | 36 | 31 | ..... |
| 8 | Single | 255 | ..... | ..... | ..... | 174 | 147 | 117 | 98 | 82 | 70 | 55 | 45 | 37 |
| 8 | Twin | 255 | ..... | ..... | ..... | ..... | 225 | 196 | 162 | 142 | 121 | 110 | 95 | 85 |
| 9 | Single | 345 | ..... | ..... | 265 | 245 | 205 | 163 | 135 | 125 | 115 | 91 | 75 | 63 |
| 9 | Twin | 345 | ..... | ..... | ..... | ..... | ..... | 274 | 227 | 193 | 169 | 152 | 135 | 120 |
| 10 | Single | 440 | ..... | ..... | ..... | 340 | 282 | 227 | 185 | 150 | 123 | 117 | 111 | 105 |
| 10 | Twin | 440 | ..... | ..... | ..... | ..... | ..... | 354 | 310 | 267 | 233 | 206 | 185 | 166 |
| 11 | Single | 545 | ..... | ..... | ..... | 450 | 376 | 300 | 252 | 210 | 180 | 170 | 159 | 145 |
| 11 | Twin | 545 | ..... | ..... | ..... | ..... | ..... | 450 | 410 | 362 | 332 | 278 | 250 | 225 |
| 12 | Single | 665 | ..... | ..... | ..... | ..... | 485 | 390 | 320 | 280 | 240 | 210 | 190 | 170 |
| 12 | Twin | 665 | ..... | ..... | ..... | ..... | ..... | 550 | 510 | 460 | 405 | 360 | 325 | 285 |
| 13 | Single | 790 | ..... | ..... | ..... | ..... | 595 | 500 | 415 | 350 | 300 | 250 | 220 | 195 |
| 13 | Twin | 790 | ..... | ..... | ..... | ..... | ..... | ..... | 605 | 570 | 515 | 461 | 427 | 376 |
| 14 | Single | 920 | ..... | ..... | ..... | ..... | 707 | 615 | 510 | 460 | 380 | 310 | 280 | 250 |
| 14 | Twin | 920 | ..... | ..... | ..... | ..... | ..... | ..... | 737 | 700 | 630 | 565 | 510 | 465 |

NOTE.—The above tonnage or pressure ratings are entirely independent of work or energy ratings. A long stroke press, if loaded to maximum pressure capacity through its maximum working stroke, will undoubtedly require additional fly-wheel and motor capacity.

**Table 5.—Approximate Capacities of Straight-side, Double-crank Presses**
(E. W. Bliss Co., Brooklyn, N. Y.)

| Stroke of Press, in. | | | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Approximate Capacity in Tons | | | | | | | | | | | | |
| Shaft Diam., in. | Type of Drive | At Bottom | At Mid-stroke | | | | | | | | | | | |
| 2 1/2 | Single | 19 | 16 | 12 | 8 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 | Single | 32 | 20 | 15 | 13 | 10 | 5 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3 1/2 | Single | 44 | 31 | 23 | 19 | 16 | 9 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 | Single | 58 | 41 | 33 | 29 | 22 | 19 | 12 | ..... | ..... | ..... | ..... | ..... | ..... |
| 4 1/2 | Single | 74 | 59 | 51 | 40 | 34 | 25 | 19 | ..... | ..... | ..... | ..... | ..... | ..... |
| 5 | Single | 93 | 73 | 64 | 56 | 47 | 36 | 28 | 21 | ..... | ..... | ..... | ..... | ..... |
| 6 | Single | 135 | ..... | 101 | 91 | 82 | 60 | 48 | 39 | ..... | ..... | ..... | ..... | ..... |
| 7 | Single | 200 | ..... | ..... | 135 | 122 | 102 | 76 | 66 | 54 | ..... | ..... | ..... | ..... |
| 8 | Single | 290 | ..... | ..... | ..... | 180 | 145 | 117 | 99 | 72 | 68 | ..... | ..... | ..... |
| 8 | Twin | 290 | ..... | ..... | ..... | 260 | 230 | 196 | 160 | 140 | 120 | 108 | 96 | ..... |
| 9 | Single | 400 | ..... | ..... | ..... | 250 | 205 | 165 | 137 | 110 | 98 | 90 | 85 | ..... |
| 9 | Twin | 400 | ..... | ..... | ..... | 333 | 310 | 273 | 227 | 195 | 170 | 150 | 133 | ..... |
| 10 | Single | 525 | ..... | ..... | ..... | 340 | 285 | 227 | 187 | 155 | 144 | 128 | 112 | ..... |
| 10 | Twin | 525 | ..... | ..... | ..... | ..... | ..... | 375 | 315 | 265 | 233 | 205 | 185 | 170 |
| 11 | Single | 700 | ..... | ..... | ..... | 450 | 375 | 305 | 250 | 200 | 190 | 175 | 152 | 135 |
| 11 | Twin | 700 | ..... | ..... | ..... | ..... | ..... | 470 | 420 | 365 | 315 | 275 | 250 | 225 |
| 12 | Single | 900 | ..... | ..... | ..... | 580 | 485 | 390 | 325 | 270 | 240 | 223 | 192 | 185 |
| 12 | Twin | 900 | ..... | ..... | ..... | ..... | ..... | 610 | 540 | 470 | 410 | 360 | 320 | 290 |
| 13 | Single | 1150 | ..... | ..... | ..... | 740 | 621 | 498 | 415 | 355 | 315 | 285 | 250 | 235 |
| 13 | Twin | 1150 | ..... | ..... | ..... | ..... | ..... | 800 | 700 | 630 | 520 | 470 | 430 | 385 |
| 14 | Single | 1400 | ..... | ..... | ..... | 900 | 770 | 610 | 510 | 425 | 380 | 350 | 310 | 275 |
| 14 | Twin | 1400 | ..... | ..... | ..... | ..... | ..... | 958 | 850 | 730 | 640 | 565 | 510 | 455 |

NOTE.—The above tonnage or pressure ratings are entirely independent of work or energy ratings. A long stroke press, if loaded to maximum pressure capacity through its maximum working stroke, will undoubtedly require additional fly-wheel and motor capacity.

**MOTOR REQUIREMENTS.**—Motor sizes specified for punch presses usually are blanket recommendations to cover the wide range of work which may be put in the all-purpose press. They frequently are high for blanking, forging or coining operations, and low for deep drawing operations. In checking, the following approximations may be used:

$$\text{Hp.} = 0.005 \times W_e \times n, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

where Hp. = horsepower of motor; $W_e$ = working energy, inch-tons; $n$ = number strokes per minute. Equation [25] holds good in cases where the fly-wheel has ample energy to meet the total requirements. Otherwise a check should be made with equation [26] and the higher value used in selecting the motor.

$$\text{Hp.} = 1.82 \times n \times (W_{e1} - W_{e2}) \div \theta \; . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [26]$$

in which $(W_{e1} - W_{e2})$ is the difference between the job requirements and the available fly-wheel energy. $\theta$ is the working portion of the cycle in degrees.

In equation [25] the motor may use the entire cycle to build up fly-wheel energy. In equation [26] it is selected to carry a portion of the working load. The motor lags some at the beginning of the demand, but then builds up to something over its rating to compensate. An allowance always must be made for friction losses and starting inertia losses, though these are small compared with average working loads.

Punch press motors must be capable of taking a slowdown up to about 10% on continuous running jobs, and up to about 20% (and higher on some jobs) for intermittently operating presses. Squirrel-cage motors are suitable for most work, but high-slip motors often are required on deep draws. Instantaneous peak loads may amount to 100% overload.

## 5. LUBRICANTS FOR WORKING METALS

**OIL.**—Many jobs may be done dry but better results and longer life of dies are obtainable by the use of lubricants. Oil lends itself to automatic lubrication by spray, drip, hollow punches, and wiper, and is often good for more than one operation. Usually an ordinary mineral oil is satisfactory.

As drawing conditions become more severe, a compounded oil of, say, 15% lard oil and 5% of wool fat in a mineral oil base is about as efficient as pure lard oil, and much less expensive. Occasionally, an unusually heavy job is considered to required heavy, crude oils which are very sticky and messy.

Electrical lamination steel often is lubricated with paraffin applied warm to the sheets. A mixture of paraffin oil and kerosene applied sparingly, to evaporate in the process, also is used.

**WATER-SOLUBLE LUBRICANTS.**—Oil lubricants are common, but ordinarily have to be cleaned off before annealing to prevent the oil burning into the steel. Again, they must be carefully cleaned off prior to soldering or to applying many finishes. For such reasons, water-soluble lubricants are widely used. These include soap solutions, flour, sulphunated oils, talc, amorphous chalk, zinc oxides, etc., in various mixtures which are more or less stable.

Fully automatic roll feed presses frequently are fitted with drip valves and felt rolls or wipers to spread the lubricant. Sometimes the drawing punches are drilled to carry oil or other lubricants to the work.

# ABRASIVE PROCESSES

**References.**—*Am. Mach.* Grinding Book; *Grits and Grinds*, Norton Co.; Points about Grinding Wheels and Their Selection, Brown & Sharpe Mfg. Co.; Treatise on Tool Room Grinding, Carborundum Co.; Table of Causes of Grinding Wheel Accidents, Independence Inspection Bureau; Safety Code for Use, Care and Protection of Abrasive Wheels, Am. Standards Assoc.; Operation of Grinding Wheels in Machine Grinding, Geo. I. Alden, *Trans.* A.S.M.E., xxxvi, p. 451, 1914.

## 1. ABRASIVE WHEELS

**SELECTION OF ABRASIVE WHEELS.**—The essential features governing the selection of a grinding wheel are: Grain, grade, bond, and structure, and the conditions of grinding that cause them to vary; methods of balancing and mounting; truing and dressing; effect on grain and grade of machine vibration and arc of contact; relation of work speed and wheel speed on production and finish; safeguards and dust removal systems.

Grinding wheels are made of abrasive grains of various sizes held together by a binder called the bond. Two types of manufactured abrasives and five types of bond are used. A few wheels are made of natural abrasive, as emery or corundum.

**ABRASIVES.**—The manufactured abrasives may be generally classified as aluminum oxide abrasives and silicon carbide abrasives. Both are products of the electric furnace, and are sold under various trade names, as Alundum, Aloxite, Borolon, Carborundum, Crystolon, etc.

A typical aluminum oxide abrasive is Alundum made by the Norton Co. by fusing bauxite in the electric furnace. Crystolon is a typical silicon carbide abrasive, made by the Norton Co., is produced by the electric furnace reaction of pure silica and coke. The aluminum oxide abrasives generally are used for grinding hard, tough and strong materials, are used for heavy duty work, such as snagging steel castings, and for cylindrical grinding on all but the hardest steels. Silicon carbide abrasives are hard and sharp, but not as tough as the aluminum oxide abrasives. They are used for snagging iron and brass castings, and for grinding materials of hard, dense grain structures. They also can be used to grind cemented carbide tools.

**GRAIN SIZE.**—Abrasive grains are designated by numbers corresponding to the number of meshes per lineal inch of the screen through which they are graded. Standard grain sizes are 8, 10, 12, 14, 16, 20, 24, 30, 36, 46, 60, 70, 80, 90, 100, 120, 150, 180, 220, and 240. The fine flours are classified hydraulically to the sizes 280, 320, 400, 500, and 600.

**BOND.**—Five types of bond are used in grinding wheels, as follows: 1. Vitrified bond, consisting of a clay mixture, subjected to a sufficiently high temperature to fuse to a molten glass condition. Vitrified wheels are strong, and can be used for very heavy work, and for the rapid removal of material. 2. Silicate bond, consisting of a mixture of silicate of soda and minerals, which at comparatively low temperatures form a hard cement. Silicate bond releases the grain more readily than does vitrified bond. Wheels with silicate bond are used on work where the heat generated by grinding must be kept at a minimum, as edge tools. 3. Shellac bond, comprising organic materials which melt at low temperatures, but attain a permanent set at slightly higher temperatures. Shellac bond wheels will impart high finishes to such work as cam-shafts and steel mill rolls. They are used for light cuts and for gumming and sharpening saws, and for cut-off wheels. 4. Rubber bond consists of rubber mixed with certain minerals in definite proportions, which on vulcanization, produces the desired hardness and toughness. Rubber bond wheels are used for high-speed grinding, and for rapid stock removal. Cut-off wheels with rubber bond can be made thin enough to reduce to a minimum the stock lost in making a cut. 5. Synthetic resin bond, as resinoid. Wheels with this type of bond are used for high-speed work in foundries and billet shops. They will operate at 9500 surface feet per minute in swing frame grinders for snagging castings and removing scale from billets. Cut-off wheels with resinoid bond will operate at 16,000 surface feet per minute.

**GRADE** is a term denoting the hardness or strength of a grinding wheel. It does not express the hardness of the abrasive itself. Hardness of abrasive grain and the grade of the wheel are quite different. A grinding wheel may be made of a very hard abrasive and still be a soft wheel. A wheel is hard when the retentive powers of the bond are great. If grains of abrasive easily are broken out of the bond, the wheel is soft. A wheel is of the proper grade when the cutting grains automatically are replaced when dulled. Wheels that are too hard will glaze and must be resharpened by dressing. The points of the dresser break out and break off the cutting grains by percussion.

While the ideal way to measure grade is to grind on each wheel, the best known practicable method is to measure the resistance offered by the wheel to the penetration of a steel cutting tool. Under a uniform pressure the tool will penetrate deeper into a softer wheel than a hard one. The resistance against turning the tool is less in a softer wheel than in a hard one. Grades are designated so that there is a gradual change in the grading from the extremely hard wheel to the extremely soft one. The practice of the various grinding wheel manufacturers in grading differs.

Most manufacturers use a letter following the grain symbol to designate the grade of the wheel. Unfortunately these letters and their relation to each other do not always mean the same thing. In the Carborundum Co. scale of grades the letters at the beginning of the alphabet indicate the hardest grades and those at the end of the alphabet the softest. Norton Co. and most of the other manufacturers use the alphabet scale in the reverse order, while a few manufacturers use special code arrangements of the letters.

For rubber, synthetic resin and shellac bonded wheels, the grade is sometimes designated by letters, sometimes by numbers. Norton Co. formerly used numbers but now uses letters to unify the system with that used for vitrified and silicate wheels. Carborundum Co. uses numbers from 1 to 8, No. 1 being the hardest and No. 8 the softest, while many others use the number system in the reverse order.

**STRUCTURE** designates the spacing of the abrasive grains in the wheel. The same total volume of voids in two different wheels may be made up of a relatively small number of voids of large volume, or of a large number of small volume. The cutting action of the two wheels will be different, although they may be of the same grain and grade. In general, wheels with the wider grain spacing will have longer life than those with close grain spacing.

Structure is denoted by numbers. The system of numbers used by the Norton Co. assigns the numbers 0, 1, 2, 3 to close spacings; the numbers 4, 5, 6 to medium spacings, and the numbers 7, 8, 9, 10, 11 and 12 to the wide spacings.

**Table 1.—Grinding Wheel Symbol Components**

| Abrasive | | Grain | Grade | | Structure | | Bond | |
|---|---|---|---|---|---|---|---|---|
| Kind | Designation | | Character | Designation | Spacing | Designation | Kind | Designation |
| Alundum | Blank | See paragraph "Grain size" p. 21–66 | Very soft | E, F, G | Close | 0, 1, 2, 3 | Vitrified | Blank |
| 38 Alundum | 38 | | Soft | H, I, J, K | Medium | 4, 5, 6 | "B" Vitrified | B |
| 19 Alundum | 19 | | Medium | L, M, N, O | Wide | 7, 8, 9, 10, 11, 12 | Silicate | S |
| 15 Alundum | 15 | | Hard | P, Q, R, S | | | Resinoid | T |
| 35 Alundum | 35 | | Very hard | T, U, W, Z | | | Rubber | R |
| Crystolon | 37 | | | | | | Shellac | L |
| | | | | | | | "V" Shellac | V |

**WHEEL MARKING.**—Grinding wheels are marked with a symbol that completely describes and identifies the wheel. The practice of the several manufacturers varies. The system used by the Norton Co. is given below.

The symbol consists of numerals and letters, which, by their position, indicate a particular characteristic of the wheel as follows:

| Position No. ........ | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Represents......... | Abrasive | Grain | Grade | Structure | Bond |
| Example........... | 38 | 46 | K | 5 | B |

The complete symbol is 3846–K5B, which represents a wheel of 38 brand Alundum abrasive, 46 grain size, grade K, structure 5, vitrified bond B. The significance of the various numbers and letters used in the symbols is given in Table 1.

**DIAMOND WHEELS.**—Wheels in which the abrasive surface is formed of diamond dust are used extensively for the grinding and lapping of cemented carbide tools and inserts, and for cutting up cemented carbide shapes. These wheels are made in a variety of sizes and shapes for use on the several types of tool and cutter grinders. In the smaller sizes, up to, say, 1 $^3/_{16}$ in. diameter by $^1/_4$ in. face, they are made solid; in the larger sizes, up to, say, 12 in. diameter, they are made with a relatively thin section of diamond on the periphery or other working surface. The sizes of abrasive used are Nos. 100, 220, and 320. For grinding, No. 100 abrasive should be used. No. 220 grain is used in wheels for lapping operations to produce keen cutting edges, and the No. 320 grain wheel is used to lap highly finished surfaces and to produce microscopically keen cutting edges.

The usual tool grinding speeds are satisfactory for diamond wheels. Water is a satisfactory coolant, if the flow can be maintained constant. In the absence of a cooling system, the surface should be lubricated freely with light machine oil. Cooling with oil or water will prolong the life of the wheel.

## 2. FACTORS AFFECTING GRINDING WHEEL SELECTION

Selection of a grinding wheel for a given operation involves consideration of the more or less fixed conditions into which the wheel must fit, and also of the factors that may be modified to suit the fixed conditions.

**FIXED CONDITIONS. 1. Material to be Ground.**—The physical properties of the material to be ground govern the type of abrasive to be used. Materials of high tensile strength, from hard, tough alloy steels to the tougher varieties of bronze, some of the aluminum alloys, and materials that are neither brittle nor easily penetrated, require aluminum oxide wheels. Different degrees of hardness, toughness and ductility in metals are handled by aluminum oxides of varying degrees of temper. See Tables 6 and 7.

Materials of low tensile strength, and which are penetrated easily, as soft brasses and bronzes, cast and chilled iron, aluminum, copper, marble, stone, glass, wood, leather, etc., should be ground with silicon carbide wheels.

In general, hard, dense materials require a relatively soft wheel. A large number of cutting points is necessary in the wheel, since the penetration of each point is slight. These points may be obtained by the use of a fine-grain wheel or a dense structure, if the use of fine grain is inadvisable. Exceptionally hard materials, as tungsten carbides, are best ground with open grain wheels. Soft, tough and ductile materials require an open grain wheel.

**2. Amount of Material to be Removed.**—For relatively large stock reduction, coarse grain wheels with open structure should be used, particularly for off-hand snagging of castings. Where finish is of importance, a closer structure should be used. Finish depends largely on grain size, except in machine grinding. Here, proper truing facilities enable a fine finish to be obtained with coarse grain wheels.

**3. Area of Contact.**—Proper grain size, grade and structure depend on the area of contact between wheel and work, because of the relation between area of contact and unit working pressure. Cylindrical grinding is an example of high unit pressure and small area of contact. Such conditions call for a wheel of fine grain, closely spaced structure to distribute the pressure over a large number of cutting points, none of which is subjected to excessive pressure. The grade should be medium hard, to prevent excessively fast wheel wear, since the forces tending to tear the grains from the wheel are great.

Surface grinding, with the rim of a cup-shaped wheel in contact with the surface of the work, is an example of high total pressure of wheel on the work, but of low unit pressure. The wheel characteristics under such conditions would be coarse grain, with widely spaced structure, to distribute the pressure per unit of area over a small number of cutting points, each receiving sufficient pressure to cut efficiently. The grade should be soft, in order that the dull grains may be released promptly, as the forces tending to tear them from the wheel are light.

**4. Type of Grinding Machine.**—The grinding machine usually is already installed, and imposes conditions that must be considered in wheel selection. Heavy, rigid machines require wheels that are softer and of more open structure than wheels for light, flexible machines. Machines that vibrate require finer grain wheels, of harder and denser structure, than the wheels for more rigid machines. Feed and speed combinations on precision grinding machines will affect the wheel characteristics. Surface grinding machines using cup wheels require softer wheels of more open structure than do similar machines using straight or disc wheels.

**VARIABLE FACTORS.—A. Wheel Speed.**—Too slow a speed wastes abrasive without corresponding useful work. Excessive speed may result in hard grinding action and danger of breakage. It is best to operate a grinding wheel close to the speed recommended by the maker. The grits, grades and structures recommended for given grinding operations are based on the recommended speeds. If these speeds cannot be used, the grade at least should be changed to meet this changed condition. Table 2 gives the peripheral speeds recommended for various classes of work.

**B. Work Speed.**—In machine grinding, high work speed tends to wear the wheel faster than slow work speed. Wheel wear depends on the ratio of wheel speed to work speed, both in surface feet per minute. The higher the ratio, the less work the wheel is required to do in a given time, and the wheel wears at a lower rate. A decreased ratio, resulting from increased work speed, requires the wheel to do more work in a given time and to wear faster. In general, the longer the arc or the larger the area of contact in precision

**Table 2.—Recommended Wheel Speeds for Various Classes of Grinding**

| Type of Grinding | Wheel Speed, Surface Feet per min. |
|---|---|
| Cylindrical grinding | 5,500– 6,500 |
| Cutter and tool grinding | 4,500– 6,000 |
| Internal grinding | 2,000– 6,000 |
| Snagging, off-hand grinding (vitrified wheels) | 5,000– 6,000 |
| Snagging (rubber and Resinoid wheels) | 7,000– 9,500 |
| Surface grinding | 4,000– 6,000 |
| Knife grinding | 3,500– 4,500 |
| Hemming cylinders | 2,100– 5,000* |
| Wet tool grinding | 5,000– 6,000 |
| Cutlery wheels | 4,000– 5,000 |
| Rubber, shellac and Resinoid cutting-off wheel | 9,000–16,000* |

* Higher speed recommended only if bearings, protection devices and machine rigidity are adequate.

**Table 3.—Revolutions per Minute for Various Surface Speeds**

| Diameter, in. | Surface Speeds, Feet per Minute | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1,000 | 2,000 | 3,000 | 4,000 | 5,000 | 6,000 | 7,000 | 8,000 | 9,000 | 10,000 | 12,000 | 14,000 | 16,000 |
| | Revolutions per Minute | | | | | | | | | | | | |
| 1 | 3,820 | 7,639 | 11,459 | 15,279 | 19,099 | 22,918 | 26,738 | 30,558 | 34,377 | 38,196 | 45,836 | 53,474 | 61,116 |
| 2 | 1,910 | 3,820 | 5,730 | 7,639 | 9,549 | 11,459 | 13,369 | 15,279 | 17,188 | 19,098 | 22,918 | 26,737 | 30,558 |
| 3 | 1,273 | 2,546 | 3,820 | 5,093 | 6,366 | 7,639 | 8,913 | 10,186 | 11,459 | 12,732 | 15,278 | 17,826 | 20,372 |
| 4 | 955 | 1,910 | 2,865 | 3,820 | 4,775 | 5,729 | 6,684 | 7,639 | 8,595 | 9,549 | 11,459 | 13,368 | 15,278 |
| 5 | 764 | 1,528 | 2,292 | 3,056 | 3,820 | 4,584 | 5,347 | 6,111 | 6,876 | 7,640 | 9,168 | 10,696 | 12,224 |
| 6 | 637 | 1,273 | 1,910 | 2,546 | 3,183 | 3,820 | 4,456 | 5,093 | 5,729 | 6,366 | 7,639 | 8,913 | 10,186 |
| 7 | 546 | 1,091 | 1,637 | 2,183 | 2,728 | 3,274 | 3,820 | 4,365 | 4,911 | 5,456 | 6,548 | 7,640 | 8,732 |
| 8 | 477 | 955 | 1,432 | 1,910 | 2,387 | 2,865 | 3,342 | 3,820 | 4,297 | 4,775 | 5,729 | 6,685 | 7,640 |
| 10 | 382 | 764 | 1,146 | 1,528 | 1,910 | 2,292 | 2,674 | 3,056 | 3,438 | 3,820 | 4,584 | 5,348 | 6,112 |
| 12 | 318 | 637 | 955 | 1,273 | 1,591 | 1,910 | 2,228 | 2,546 | 2,864 | 3,183 | 3,820 | 4,456 | 5,092 |
| 14 | 273 | 546 | 818 | 1,091 | 1,364 | 1,637 | 1,910 | 2,183 | 2,455 | 2,728 | 3,274 | 3,820 | 4,366 |
| 16 | 239 | 477 | 716 | 955 | 1,194 | 1,432 | 1,671 | 1,910 | 2,149 | 2,387 | 2,865 | 3,342 | 3,820 |
| 18 | 212 | 424 | 637 | 849 | 1,061 | 1,273 | 1,485 | 1,698 | 1,910 | 2,122 | 2,546 | 2,970 | 3,396 |
| 20 | 191 | 382 | 573 | 764 | 955 | 1,146 | 1,337 | 1,528 | 1,719 | 1,910 | 2,292 | 2,674 | 3,056 |
| 22 | 174 | 347 | 521 | 694 | 868 | 1,042 | 1,215 | 1,389 | 1,562 | 1,736 | 2,084 | 2,430 | 2,776 |
| 24 | 159 | 318 | 477 | 637 | 796 | 955 | 1,114 | 1,273 | 1,433 | 1,591 | 1,910 | 2,228 | 2,546 |
| 30 | 127 | 255 | 382 | 509 | 637 | 764 | 891 | 1,018 | 1,146 | 1,274 | 1,528 | 1,782 | 2,036 |
| 36 | 106 | 212 | 318 | 424 | 530 | 637 | 743 | 849 | 954 | 1,061 | 1,273 | 1,484 | 1,698 |

**Table 4.—Circumferences in Feet of Grinding Wheels for Given Diameter in Inches**

| Diam., in. | Circum., ft. | Diam., in. | Circum., ft. | Diam., in. | Circum., ft. | Diam., in. | Circum., ft. | Diam., in. | Circum., ft. | Diam., in. | Circum., ft. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.262 | 13 | 3.403 | 25 | 6.546 | 37 | 9.687 | 49 | 12.828 | 61 | 15.970 |
| 2 | .524 | 14 | 3.665 | 26 | 6.807 | 38 | 9.948 | 50 | 13.090 | 62 | 16.232 |
| 3 | .785 | 15 | 3.927 | 27 | 7.069 | 39 | 10.210 | 51 | 13.352 | 63 | 16.493 |
| 4 | 1.047 | 16 | 4.189 | 28 | 7.330 | 40 | 10.472 | 52 | 13.613 | 64 | 16.755 |
| 5 | 1.309 | 17 | 4.451 | 29 | 7.592 | 41 | 10.734 | 53 | 13.875 | 65 | 17.017 |
| 6 | 1.571 | 18 | 4.712 | 30 | 7.854 | 42 | 10.996 | 54 | 14.137 | 66 | 17.279 |
| 7 | 1.833 | 19 | 4.974 | 31 | 8.116 | 43 | 11.257 | 55 | 14.499 | 67 | 17.541 |
| 8 | 2.094 | 20 | 5.236 | 32 | 8.377 | 44 | 11.519 | 56 | 14.661 | 68 | 17.802 |
| 9 | 2.356 | 21 | 5.498 | 33 | 8.639 | 45 | 11.781 | 57 | 14.923 | 69 | 18.064 |
| 10 | 2.618 | 22 | 5.760 | 34 | 8.901 | 46 | 12.043 | 58 | 15.184 | 70 | 18.326 |
| 11 | 2.880 | 23 | 6.021 | 35 | 9.163 | 47 | 12.305 | 59 | 15.446 | 71 | 18.588 |
| 12 | 3.142 | 24 | 6.283 | 36 | 9.425 | 48 | 12.566 | 60 | 15.708 | 72 | 18.850 |

grinding operations, the faster should be the work speed, in order to have the wheel cut properly. In off-hand grinding, the rate at which the work is forced against the wheel or the method of applying the work influences grade selection. The harder the grinding wheel is forced, the harder should be the wheel for abrasive economy. This relation also exists in semi-automatic and high-production precision grinding.

**C. Condition of Grinding Machine.**—The grinding wheel cannot work to good advantage on a machine in poor repair or improperly set up. If conditions are bad, allowances must be made in the selection of the wheel grade. Spindles loose in the bearings require much harder wheels than do normal conditions. Insecure or shaky foundations necessitate hard grades to overcome the tendency of the wheel to wear rapidly, cutting qualities being sacrificed thereby.

**D. Skill of Workmen.**—Grinding costs on off-hand grinding may vary 100% on some work with the same kind of machine in the same factory due to the difference in the method of applying the work to the wheel, and to the variation of skill and knowledge of the different workmen. Even in machine grinding, the operator can vary the results in grinding castings by his methods of handling the machine. The foregoing considerations are summarized in Table 5. Table 6 gives the type, grain and grade of wheel recommended for various classes of grinding.

**Table 5.—Factors Influencing the Selection of Abrasive Wheels.** (Norton Co.)

| Selection of | Influential Factors | Kind of Wheel | Material Ground, Result Desired, or Condition To Be Met |
|---|---|---|---|
| Abrasive | Physical Properties of Material Ground | Alundum | Carbon, alloy or high-speed steel; annealed malleable iron; wrought iron; tough bronzes; tungsten, etc. |
| | | Crystolon | Gray iron; chilled iron; brass; soft bronze; aluminum; copper; marble; granite; pearl; rubber; leather; very hard alloys as tungsten-carbide, etc. |
| Grain | Amount of Material to be Removed | Coarse grain | Fast removal of stock |
| | Finish Desired | Fine grain | Fine finish |
| | Physical Properties of Material to be Ground | Coarse grain | Soft, ductile materials |
| | | Fine grain | Dense, hard or brittle material |
| Grade | Physical Properties of Material to be Ground | Hard Wheel | Soft material |
| | | Soft Wheel | Hard material |
| | Area of Contact | . . . . . . . . . . . . . . | Hardness increases as area of contact decreases |
| | Wheel Speed | Hard Wheel | Low wheel speed |
| | | Soft Wheel | High wheel speed |
| | Work Speed | Hard Wheel | High work speed |
| | | Soft Wheel | Low work speed |
| | Condition of Grinding Machine | Hard Wheel | Machine in poor condition |
| | | Soft Wheel | Machine in good condition |
| | Skill of Operator | Harder Wheels | Unskilled operator; piece work |
| | | Softer Wheels | Skilled operator; day work; economical production |
| Structure | Physical Properties of Material to be Ground | Wide Spacing | Soft, tough, ductile materials |
| | | Close Spacing | Hard, brittle materials (except cemented carbides) |
| | Finish Desired | Wide Spacing | Medium or coarse finish |
| | | Close Spacing | Fine finish |
| | Nature of Operation | Wide Spacing | Snagging; operations with flexible application of pressure; surfacing |
| | | Medium Spacing | Cylindrical, centerless, tool and cutter grinding |
| | | Close Spacing | Heavy pressures tending to deform wheel |
| Bond | Wheel Dimensions | Vitrified | For general use; rapid removal of stock |
| | | Silicate | Solid wheels over 36 in. diameter |
| | | Synthetic Resin<br>Shellac<br>Rubber | Thin cutting-off wheels; wheels subject to deflection |
| | Operating Speed | Vitrified | Wheel speeds up to 6500 surface ft. per min. |
| | | Synthetic Resin<br>Shellac<br>Rubber | Wheel speeds above 6500 surface ft. per min. |
| | Finish Desired | Synthetic Resin<br>Shellac<br>Rubber | High finish where high production is not a factor |
| | | Silicate | To replace sandstone on cutlery, etc. |

## Table 6.—Grinding Wheel Recommendations

Based on the Norton Co. system of wheel marking. See p. 21–67.

| Work, Material or Operation | Abrasive * | Grade | Grain | Structure | Bond † |
|---|---|---|---|---|---|
| **Aluminum** | | | | | |
| Cylindrical | 37 | 30 | J | 8 | ..... |
| Surfacing (cups or cylinders) | 38 | 24 | K | 8 | B |
| Surfacing (straight wheels) | 38 | 46 | H | 8 | B |
| Internal | 37 | 46 | K | ..... | ..... |
| Snagging (low speed) | 37 | 24 | O | 6 | ..... |
| " (high speed) | ..... | 30 | O | 4 | T2 |
| Cutting-off | ..... | 24 | S | 8 | T2 |
| **Axles,** Automobile and Railway | | | | | |
| Centerless | ..... | 46 | M | 5 | BE |
| Cylindrical | ..... | 46 | L | 5 | BE |
| **Balls, Steel** | | | | | |
| Soft, large, rough | ..... | 46 | Z1 | 7 | ..... |
| " small, rough | ..... | 60 | Z2 | 8 | ..... |
| " large, semi-finish | ..... | 80 | Z2 | 9 | ..... |
| " small, semi-finish | ..... | 180 | Z4 | 10 | ..... |
| Hard, large, final finish | 37 | 320 | F–X | 9A | ..... |
| " small, final finish | 37 | XF | Z4 | 11 | ..... |
| **Ball Bearings** | | | | | |
| Surfacing cups and cones, soft | 38 | 46 | G | ..... | S |
| " cups and cones, hard | 38 | 80 | F | ..... | S |
| O.D. cups, rough and finish | ..... | 80 | M | 5 | BE |
| Outer race | ..... | 80 | R | 6 | R |
| Inner race | ..... | 120 | P | 2 | R |
| Internal bore | ..... | 60 | M | 4 | ..... |
| **Billets, Cleaning** | | | | | |
| High-carbon and high speed steels | | | | | |
| High speed swing frames | ..... | 14 | Q | 4 | T2H |
| Low speed swing frames | ..... | 16 | Q | 5 | B |
| Flexible shaft | ..... | 20 | Q | 5 | B |
| Stainless and alloy steels | | | | | |
| High speed swing frames | ..... | 12 | S | 4 | T2H |
| Low speed swing frames { | 15 | 14 | S | 6 | |
| | ..... | 14 | R | 7 | B |
| Flexible shaft | ..... | 20 | R | 7 | B |
| **Bits, Auger** | | | | | |
| Grinding throats | ..... | 36 | R | 0 | R |
| Fluting | ..... | 70 | R | 0 | R |
| **Brass** | | | | | |
| Centerless | 37 | 36 | N | 5 | ..... |
| Cylindrical | 37 | 36 | K | 5 | ..... |
| Internal | 37 | 36 | J | 6 | ..... |
| Surfacing (cups and cylinders) | 37 | 24 | H | 8 | ..... |
| Snagging (slow speed) | 37 | 24 | Q | 6 | ..... |
| " (high speed) | 37 | 20 | O | 2 | T2 |
| Cutting-off | ..... | 30 | W | 7 | T3 |
| **Broaches,** sharpening | 38 | 46 | K | 5 | BE |
| **Bronze,** soft, See Brass | | | | | |
| **Bronze,** hard | | | | | |
| Centerless | 37 | 50 | K | 5 | ..... |
| Cylindrical | ..... | 46 | K | 5 | BE |
| Internal | 38 | 60 | J | 5 | BE |
| Snagging | ..... | 20 | Q | 8 | B |
| Cutting-off | ..... | 30 | W | 7 | T3 |
| **Bushings,** hardened steel | | | | | |
| Cylindrical | ..... | 60 | L | 5 | BE |
| Internal | 38 | 60 | K | 5 | BE |
| Centerless (rough) | ..... | 60/2 | M | 5 | BE |
| " (finish) | ..... | 120 | N | 0 | R |

| Work, Material or Operation | Abrasive * | Grade | Grain | Structure | Bond † |
|---|---|---|---|---|---|
| **Cams,** hardened steel | | | | | |
| Roughing | ..... | 60 | L | 5 | BE |
| Finishing | ..... | 60 | L | 1 | R |
| **Cam Rollers,** hardened steel | | | | | |
| Cylindrical | 38 | 60 | N | 6 | BE |
| Internal | 38 | 60 | L | ..... | ..... |
| **Cam Shaft Bearings** | | | | | |
| Cylindrical | ..... | 46 | N | 5 | BE |
| **Carbon** | | | | | |
| Soft, cutting-off | 37 | 16 | R | 0 | R |
| Hard, round, centerless | 37 | 36 | N | 5 | ..... |
| " round, cutting-off | 37 | 46 | R | 6 | T2 |
| " plate, stripping | 37 | 30 | O | 6 | T2 |
| Metallic, cutting-off | 37 | 46 | Q | 6 | T2 |
| Surfacing | 37 | 30 | J | 8 | ..... |
| **Cast Iron** | | | | | |
| Centerless | 37 | 46 | L | 5 | ..... |
| Cutting-off | 37 | 36 | R | 8 | T2 |
| Cylindrical | 37 | 36 | J | ..... | ..... |
| Internal | 37 | 46 | J | ..... | ..... |
| Surfacing (cups and cylinders) | 37 | 16 | H | 7 | ..... |
| " (straight wheels) | 37 | 30 | J | 8 | ..... |
| Snagging (low speed) { | 37 | 16 | S | 5 | ..... |
| | ..... | 30 | R | 7 | B |
| " (low speed portable) | 37 | 16 | S | 5 | ..... |
| " (high speed) | 37 | 16 | S | 2 | T2H |
| **Chain Links,** Annealed Malleable Iron and Steel | | | | | |
| Snagging (low speed) | ..... | 20 | Q | 8 | B |
| " (high speed) | ..... | 16 | P | 4 | T2 |
| **Chilled Iron** | | | | | |
| Snagging (high speed) | 37 | 16 | Q | 2 | T2H |
| " (low speed) | 37 | 16 | R | 5 | ..... |
| Surfacing (cups and cylinders) | 38 | 24 | I | 8 | B |
| " (straight wheels) | 37 | 36 | I | 8 | ..... |
| **Chisels, Woodworking** | | | | | |
| Surfacing (cups and cylinders) | ..... | 46 | O | 3 | L |
| Edging (cups and cylinders) | ..... | 60 | M | 7 | B |
| Sharpening | ..... | 46 | M | 5 | BE |
| **Chromium Plating** | | | | | |
| Good commercial finish | 38 | 60 | L | 5 | BE |
| Excellent commercial finish | ..... | 80 | K | 5 | L |
| High finish | ..... | 500 | I | 9 | L |
| **Connecting Rods** | | | | | |
| Internal | 38 | 60 | K | ..... | ..... |
| Surfacing (cups and cylinders) | 38 | 24 | J | 8 | B |
| **Copper** | | | | | |
| Tubes (cylindrical) | 37 | 70 | J | 5 | L |
| Cylindrical | ..... | 60 | L | 4 | L |
| Surfacing (cups and cylinders) | 37 | 14 | I | 7 | ..... |
| Cutting-off, wet (low speed) | 37 | 36 | P | 0 | R |
| " (low speed) | 37 | 24 | R | 0 | R |
| **Crank-shafts** | | | | | |
| Airplane | 38 | 60 | K | 5 | BE |
| Diesel | ..... | 46 | L | 5 | BE |
| Automotive | | | | | |
| Snagging (slow speed) | ..... | 20 | O | 5 | B |
| " (high speed) | ..... | 20 | P | 4 | T2L |

*See footnotes at end of table.*

*Table continued on following page.*

### Table 6.—Grinding Wheel Recommendations—*Continued*

| Work, Material or Operation | Abrasive * | Grade | Grain | Structure | Bond † |
|---|---|---|---|---|---|
| **Crank-shaft Pins, Bearings** | | | | | |
| Rough, heavy side removal | ..... | 36 | R | 5 | A |
| “ light side removal | ..... | 46 | Q | 5 | A |
| Finish | ..... | 50 | O | 5 | A |
| **Cylinders, Cast Iron, Internal** | 37 | 36 | H | 9 | ..... |
| **Dies** | | | | | |
| Forging, cleaning | ..... | 60 | P | .... | ..... |
| Threading, chamfering | ..... | 80 | Q | 5 | B |
| Drawing, cleaning | ..... | 60 | L | 5 | BE |
| “ Surfacing (cups and cylinders) | 38 | 24 | J | 8 | B |
| “ “ (straight wheels) | 38 | 46 | I | 8 | BE |
| **Drills** | | | | | |
| Cutting-off (high speed) | ..... | 36 | R | 8 | T2 |
| “ (low speed) | ..... | 60 | P | 7 | T2 |
| Cylindrical | ..... | 50 | N | 5 | BE |
| Centerless (soft) through feed | ..... | 60 | M | 5 | BE |
| “ (hard) in feed | ..... | 80 | N | 5 | B |
| Fluting, small | ..... | 46 | R | 0 | R |
| “ large | ..... | 46 | P | 5 | BE |
| Off-hand sharpening, small | ..... | 60 | N | 5 | BE |
| “ “ large (wet) | ..... | 36 | P | 5 | BE |
| Precision sharpening | 38 | 46 | M | 5 | BE |
| Point thinning | ..... | 60 | N | 5 | BE |
| **Forgings** | | | | | |
| Centerless | ..... | 60 | M | 5 | BE |
| Cylindrical | ..... | 46 | M | 5 | BE |
| Snagging (high speed) | ..... | 16 | P | 4 | T2L |
| “ (low speed) | ..... | 20 | Q | 8 | B |
| **Gages** | | | | | |
| Plug, cylindrical | 38 | 80 | K | 5 | BE |
| Thread | | | | | |
| Threading, coarse pitch | 38 | 120 | K | .... | ..... |
| “ medium pitch | 38 | 220 | K | .... | ..... |
| “ fine pitch | 38 | 220 | M | .... | ..... |
| **Gears,** hardened steel | | | | | |
| Form precision grinding | 38 | 60 | K | 5 | BE |
| Generative precision grinding | 38 | 60 | J | 5 | BE |
| Internal | 38 | 46 | L | 5 | BE |
| | ..... | 46 | L | 5 | BE |
| Surfacing (cups and cylinders) | 38 | 30 | I | 8 | B |
| “ (straight wheels) | 38 | 46 | J | 5 | BE |
| **Glass, Tubing** | | | | | |
| Cutting-off (low speed), wet | 37 | 90 | K | 0 | R |
| Cylindrical | 37 | 36 | J | 6 | ..... |
| Internal | 37 | 60 | H | 9 | ..... |
| Nicking | 37 | 180 | S | .... | ..... |
| **Hammers, Claw** | | | | | |
| Grinding between claws | ..... | 46 | R | 0 | R |
| Neck grinding | ..... | 24 | Q | 5 | B |
| **Knives, Machine** | | | | | |
| Chipper and barker | ..... | 36 | K | 5 | BE |
| Moulding, off-hand sharpening | ..... | 36 | N | 5 | BE |
| Machine, sharpening | ..... | 36 | J | 5 | BE |
| Veneer, sharpening | 19 | 36 | J | .... | S |
| **Machine Shop** | | | | | |
| General off-hand grinding | ..... | 30 | P | 5 | BE |
| **Malleable Iron,** Annealed | | | | | |
| Snagging (slow speed) | ..... | 16 | R | 7 | B |
| “ (high speed) | ..... | 16 | Q | 4 | T2H |
| **Malleable Iron,** Unannealed | | | | | |
| Snagging (slow speed) | 37 | 16 | T | 5 | ..... |
| “ (high speed) | 37 | 16 | Q | 4 | T2 |
| **Monel Metal** | | | | | |
| Cutting-off | ..... | 36 | P | 8 | T2 |
| Snagging | ..... | 24 | Q | 5 | B |
| Cylindrical | ..... | 46 | L | 5 | B |
| **Nitralloy** | | | | | |
| Before nitriding | ..... | 36 | J | .... | ..... |
| After nitriding | 37 | 100 | I | .... | ..... |
| | 38 | 60 | H | 8 | B |
| **Pipe** | | | | | |
| Cast-iron, internal | 37 | 14 | T | 5 | ..... |
| | 37 | 16 | S | 4 | T2H |
| Soft steel, cutting-off | ..... | 36 | S | 8 | T2 |
| “ “ internal | ..... | 20 | P | .... | ..... |
| **Pistons** | | | | | |
| Aluminum, cylindrical | 37 | 36 | J | 7 | ..... |
| “ centerless | 37 | 46 | I | 5 | ..... |
| “ regrinding | 37 | 46 | K | 5 | ..... |
| Cast-iron, cylindrical | 37 | 36 | K | 7 | ..... |
| “ centerless | 37 | 46 | J | 6 | ..... |
| “ regrinding | 37 | 46 | K | 5 | ..... |
| **Piston Pins** | | | | | |
| Centerless, roughing | ..... | 60 | M | 5 | BE |
| “ semi-finishing | ..... | 80 | M | 5 | BE |
| “ finishing | 37 | 320 | N | 8 | L |
| **Piston Rings,** Cast-iron, Semi-steel | | | | | |
| Surfacing, rough (cylinders) | 38 | 30 | H | 8 | B |
| “ (straight wheels) | ..... | 50 | L | 5 | BE |
| Internal, snagging | 37 | 36 | R | .... | ..... |
| **Plows** | | | | | |
| Steel, surfacing | ..... | 20 | Q | 5 | B |
| “ fitting | ..... | 24 | R | 7 | B |
| “ edging, jointing | ..... | 16 | S | .... | ..... |
| Chilled iron, all operations, wet | ..... | 20 | R | 7 | B |
| “ iron, all operations, dry | ..... | 20 | N | 5 | B |
| **Pulleys** | | | | | |
| Cast-iron, cylindrical | 37 | 30 | J | 8 | ..... |
| **Reamers** | | | | | |
| Backing-off | 38 | 46 | K | 5 | BE |
| Cylindrical | 38 | 46 | M | 5 | BE |
| Fluting | 38 | 46 | L | 5 | BE |
| **Roller Bearings** | | | | | |
| Cups, centerless, O.D. | ..... | 80 | L | 7 | BE |
| “ internal | ..... | 70 | N | 5 | BE |
| Rollers, centerless, roughing | ..... | 80 | O | 7 | BE |
| “ centerless, finishing | ..... | 90 | R | .... | R |
| **Rolls** | | | | | |
| Brass or copper | | | | | |
| Cylindrical, roughing | 37 | 46 | L | 3 | L |
| “ finishing | 37 | 100 | I | 9 | ..... |
| Cast-iron | | | | | |
| Cylindrical, roughing | 37 | 30 | K | 5 | ..... |
| “ finishing | 37 | 80 | J | 7 | ..... |

*See footnotes at end of table.*

*Table continued on following page.*

**Table 6.—Grinding Wheel Recommendations—*Continued***

| Work, Material or Operation | Abrasive * | Grade | Grain | Structure | Bond † |
|---|---|---|---|---|---|
| **Rolls**—*Continued* | | | | | |
| Chilled iron | | | | | |
| Machine, roughing | 37 | 30 | L | 5 | L |
| | 37 | 36 | L | 5 | T2 |
| " finishing | 37 | 70 | J | 4 | L |
| Hard rubber | | | | | |
| Cylindrical, rough and finish | 37 | 36 | K | 5 | ..... |
| Soft rubber | | | | | |
| Cylindrical, rough | 37 | 24 | K | 5 | T2 |
| " finish | 37 | 60 | K | 5 | T2 |
| Steel, hardened | | | | | |
| Cylindrical | 38 | 46 | L | 5 | BE |
| | 38 | 100 | I | 6 | BE |
| " polishing | 37 | 500 | I | 9 | L |
| **Rubber** | | | | | |
| Soft, cylindrical | 37 | 24 | K | 5 | L |
| Hard, cutting-off | 37 | 36 | Q | 6 | T2 |
| " cylindrical | 37 | 30 | K | 5 | T2 |
| **Saws** | | | | | |
| Band, gumming | ..... | 46 | M | 5 | BE |
| | ..... | 36 | O | 7 | T |
| Circular, gumming | ..... | 46 | M | 5 | BE |
| Metal-cutting, gumming | ..... | 60 | P | .... | ..... |
| **Scissors and Shears** | | | | | |
| Cast-iron, surfacing blades | 37 | 100 | S | .... | ..... |
| Steel, surfacing blades | ..... | 150 | O | 5 | V |
| " grinding flash from bows | ..... | 46 | P | .... | ..... |
| " pointing and shaping | ..... | 50 | O | .... | ..... |
| " grinding neck | ..... | 120 | N | 5 | L |
| " strike cutting edges | ..... | 120 | M | .... | ..... |
| **Shear Blades** | | | | | |
| Segments | ..... | 30 | G | 8 | B |
| Cylinders | 38 | 30 | G | 8 | B |
| **Shovels**, edging | ..... | 30 | R | 7 | B |
| **Spline Shafts** | | | | | |
| Centerless | ..... | 50 | N | 5 | BE |
| Cylindrical | ..... | 50 | O | 5 | BE |
| Surfacing splines | ..... | 50 | L | 5 | BE |
| **Springs** | | | | | |
| Leaf, grinding eyes | ..... | 20 | P | 5 | B |
| " chamfering | ..... | 20 | S | .... | ..... |
| Coil, squaring ends, small | ..... | 30 | O | 5 | B |
| " " " large | ..... | 16 | N | 5 | B |
| **Steel Castings** | | | | | |
| Low carbon | | | | | |
| Swing frame (low speed) | ..... | 14 | Q | 5 | B |
| " " (high speed) | ..... | 14 | Q | 4 | T2H |
| Floor stand (low speed) | ..... | 20 | Q | 8 | B |
| " " (high speed) | ..... | 20 | Q | 4 | T2L |
| Manganese | | | | | |
| Snagging (low speed swing frame) | ..... | 14 | Q | 5 | B |
| " (high speed swing frame) | ..... | 14 | R | 4 | T2H |
| " (low speed floor stand) | ..... | 16 | P | 5 | B |
| " (high speed floor stand) | ..... | 20 | Q | 4 | T2L |
| **Steel, hardened** | | | | | |
| Centerless | 38 | 60 | M | 5 | BE |
| Cylindrical | 38 | 46 | L | 5 | BE |
| Surfacing (cups and cylinders) | 38 | 30 | G | 8 | B |
| " (straight wheels) | 38 | 36 | H | 8 | BE |
| Internal | 38 | 60 | J | 5 | GE |
| Cutting-off | ..... | 46 | Q | 8 | T2 |

| Work, Material or Operation | Abrasive * | Grade | Grain | Structure | Bond † |
|---|---|---|---|---|---|
| **Steel**, soft | | | | | |
| Centerless | ..... | 60 | M | 5 | BE |
| Cylindrical | ..... | 46 | N | 5 | BE |
| Surfacing (cups and cylinders) | 38 | 16 | K | 8 | B |
| " (straight wheels) | 38 | 36 | L | 8 | BE |
| Internal | 38 | 46 | L | 5 | BE |
| Cutting-off, dry | ..... | 24 | S | 8 | T2 |
| " " wet | ..... | 60 | R | 2 | RIT |
| **Steel**, high-speed | | | | | |
| Centerless | ..... | 60 | L | 5 | BE |
| Cylindrical | 38 | 46 | L | 5 | BE |
| Surfacing (cups and cylinders) | 38 | 46 | G | 8 | BE |
| " (straight wheels) | 38 | 46 | H | 8 | BE |
| Cutting-off | ..... | 46 | Q | 8 | T2 |
| Internal | 38 | 46 | J | 5 | BE |
| Snagging, annealed billets | ..... | 14 | Q | 5 | B |
| " unannealed billets | ..... | 16 | P | 5 | B |
| **Steel**, stainless | | | | | |
| Centerless | 37 | 50 | M | 9 | ..... |
| Cylindrical | 37 | 46 | M | 5 | ..... |
| Surfacing | 38 | 36 | H | 8 | BE |
| Cutting-off | ..... | 36 | Q | 8 | T2 |
| Snagging, billets (high speed) | ..... | 12 | T | 2 | T2H |
| " billets (low speed) | 15 | 14 | S | 6 | ..... |
| **Stellite** | | | | | |
| Cylindrical | ..... | 46 | M | 5 | BE |
| Off-hand (tools) | ..... | 46 | N | 5 | BE |
| Surfacing (cups and cylinders) | ..... | 50 | G | 8 | BE |
| " (straight wheels) | 38 | 46 | H | 8 | BE |
| Cutting-off (low speed) | ..... | 60 | P | 7 | T2 |
| Tool and cutter | 38 | 46 | J | 5 | BE |
| **Taps** | | | | | |
| Squaring ends | ..... | 60 | N | 5 | BE |
| Grinding relief | 38 | 80 | J | 5 | BE |
| Fluting, small taps | ..... | 60 | R | .... | R |
| " large taps | 38 | 46 | L | 5 | BE |
| Threading | 38 | 150 | L | .... | ..... |
| Shanks (cylindrical) | 38 | 80 | M | 6 | BE |
| **Tools, Lathe and Planer,** Carbon and High-speed Steel | | | | | |
| Light off-hand roughing | ..... | 36 | O | 5 | BE |
| " " " finishing | ..... | 60 | N | 5 | BE |
| Heavy, off-hand | ..... | 30 | O | 5 | B |
| Automatic | ..... | 24 | M | 5 | B |
| Fixture grinding | 38 | 46 | K | 5 | BE |
| **Tubes**, steel | | | | | |
| Centerless | ..... | 60 | N | 5 | BE |
| Cutting-off, thin wall | ..... | 60 | R | 2 | RIT |
| " " heavy wall | ..... | 46 | R | 2 | RIQ |
| **Tungsten- and Tantalum-carbide** | | | | | |
| Off-hand roughing | 37‡ | 60 | J | 7 | ..... |
| " " semi-finishing | 37‡ | 80 | I | 7 | ..... |
| " " finishing | 37‡ | 100 | I | 7 | ..... |
| Fixture grinding (semi-off-hand) | 37‡ | 60 | J | 7 | ..... |
| " grinding (semi-finish | 37‡ | 80 | J | 7 | ..... |

*See footnotes at end of table.*

*Table continued on following page.*

Table 6.—Grinding Wheel Recommendations—*Continued*

| Work, Material or Operation | Abrasive * | Grade | Grain | Structure | Bond † |
|---|---|---|---|---|---|
| **Tungsten- and Tantalum-carbide**—*Continued* | | | | | |
| Surface grinding | | | | | |
| Rough tungsten...... | 37‡<br>37‡ | 80<br>60 | H<br>I | 7<br>7 | .....<br>..... |
| Finish tungsten........ | 37‡ | 100 | H | 7 | ..... |
| Rough tantalum....... | 37‡ | 80 | H | 7 | ..... |
| Finish tantalum........ | 37‡ | 100 | G | 7 | ..... |
| Wet, rough............ | 37‡ | 60 | J | 7 | ..... |
| " finish............ | 37‡ | 100 | I | 7 | ..... |
| Form grinding......... | 37‡ | 100 | I | 7 | ..... |
| Sharp angles......... | 37‡ | 280 | I | 7 | ..... |
| Backing-off............ | 37‡ | 60 | I | 7 | ..... |
| Milling cutters (cups). | 37‡ | 80 | H | 7 | ..... |
| Centerless.............. | 37‡ | 80 | H | 7 | ..... |
| Milling cutters (straight wheels)........... | 37‡ | 60 | J | 7 | ..... |
| Cylindrical, rough...... | 37‡ | 60 | J | 7 | ..... |
| " finish...... | 37‡ | 100 | I | 7 | ..... |
| Internal.............. | 37‡ | 46 | L | 8 | ..... |
| Lapping (loose grain)..... | ..... | 280§ | .... | .... | ..... |
| " solid wheel, coarse.. | ..... | 100¶ | .... | .... | ..... |
| " " " medium | ..... | 220¶ | .... | .... | ..... |
| " " " fine.... | ..... | 320¶ | .... | .... | ..... |
| **Tungsten Rods** | | | | | |
| Cutting-off.............. | ..... | 150 | R | 7 | RID |
| Centerless............... | 37 | 60 | J | 9 | ..... |
| **Valves, Automobile** | | | | | |
| Seats..................... | ..... | 80 | O | 6 | BE |
| Stems.................... | ..... | 46 | N | 5 | BE |
| " cutting-off......... | ..... | 30 | W | 7 | T3 |
| " centerless grind..... | ..... | 60 | O | 5 | BE |
| **Valve Seat Inserts** | | | | | |
| Cast-iron, roughing....... | 37 | 46 | M | .... | ..... |
| Alloy steel, roughing...... | 38 | 60 | N | 6 | BE |
| Stellite, roughing......... | 38 | 80 | I | 8 | BE |
| All seats, finishing........ | 37 | 150 | L | .... | ..... |
| **Valve Tappets** | | | | | |
| Facing.................... | 38 | 60 | L | 5 | BE |
| Centerless............... | ..... | 80 | P | 7 | BE |
| Cylindrical............... | ..... | 46 | M | 5 | BE |
| **Worms,** grinding threads.. | 38 | 36 | L | 5 | BE |
| **Wrought iron,** snagging... | ..... | 16 | R | 7 | B |

* A blank space in the abrasive column signifies that the abrasive is alundum.

† A blank space in the bond column signifies vitrified bond.

‡ Green crystolon. § Boron carbide. ¶ Diamond resinoid.

Table 7.—Wheel Recommendations for Portable Grinding Equipment *

| Material | High-speed Grinder | Slow-speed Grinder |
|---|---|---|
| Aluminum Castings... | 30–N4T–2$^{a}$ | 3724–O$^{b}$ |
| Brass and Bronze..... | 3724–Q2T–2$^{c}$ | 3724–Q$^{b}$ |
| Gray Iron Casting.... | 3716–S2T–2$^{c}$ | 3724–S$^{b}$ |
| Forgings............. | 20–Q2T–2$^{a}$ | 24–Q$^{d}$ |
| Malleable Castings (annealed)......... | 16–Q2T–2$^{a}$ | 24–Q$^{d}$ |
| Malleable Castings (unannealed)....... | 3716–R2T–2$^{c}$ | 3724–S$^{b}$ |
| Steel Castings (low carbon........... | 20–U0R$^{e}$<br>16–Q2T–2$^{a}$ | 24–Q$^{d}$ |
| Manganese Steel Castings......... | 20–Q2T–2$^{a}$ | 24–Q$^{d}$ |
| Stainless Steel...... | 20–U0R$^{e}$<br>20–Q2T–2$^{a}$ | 24–Q$^{d}$ |
| Welds, Straight Wheels.......... | 20–U0R$^{e}$<br>20–R4T–2H$^{a}$ | 24–Q$^{d}$ |
| Welds, Cup Wheels . | 20–Q4T–2$^{a}$<br>20–S0R$^{e}$ | ...... |

* For significance of the wheel symbols, see p. 21–67.

[a] Alundum resinoid. [b] Crystolon vitrified. [c] Crystolon resinoid. [d] Alundum vitrified. [e] Alundum rubber.

**GRINDING WITH PORTABLE EQUIPMENT.**—Portable grinders have come into wide use in foundries and steel mills for snagging, cleaning and smoothing of castings and billets, and in industries such as steel furniture and automobile body, and in tool rooms and die shops where they are used as production tools. The wheels used generally are smaller than those used in stationary or swing frame grinders, and are driven at higher speeds by electric motor or air turbine. The grain, grade and other properties also are different in portable equipment. Table 7 gives typical examples of wheel recommendations made by the Norton Co. for portable equipment.

The following recommendations * are made for wheels for use in portable equipment in the tool room and die shop:

**Internal Grinding.**—For hardened or high-speed steels, 3846–J, aluminum oxide, vitrified; soft steels, 3846–K aluminum oxide, vitrified; cast iron, 3736–H silicon carbide, vitrified.

**Surfacing.**—Tool and die steels, 3846–K5B, aluminum oxide, vitrified.

**General Purpose Grinding.**—Tool and die steels, 3846–L5B aluminum oxide, vitrified.

**Off-hand Grinding.**—Dies, metal patterns and molds, from 36 to 60 grain size, grade P to R, aluminum oxide, vitrified, or 30–Q4T–2 aluminum oxide, resinoid; steel castings and forgings, 36–P5B aluminum oxide, vitrified; grinding keyways, slotting or cutting off small stock with wheels $^1/_{16}$ to $^1/_8$ in. thick, 60–P8T–2 aluminum oxide, resinoid.

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* For significance of the wheel symbols, see p. 21–67.

**WHEELS FOR SURFACE GRINDING.**—Vertical-spindle surface grinders and face grinders use wheels of the cup form, the grinding surface being the edge of the rim of the wheel. Built-up sectional wheels frequently are used in place of solid cup wheels both for surface grinders and for disc grinders.

The Bridgeport Safety Emery Wheel Company, Bridgeport, Conn., makes a wheel consisting of segments mounted in a steel chuck. The segments can be moved outwards from the face of the chuck to compensate for wear. Number of segments ranges from 8, in a 16-in. wheel, to 18 in a 42-in. wheel. The depth of the segments is 8 in. in all cases, and practically this whole depth can be worn away in the use of the wheel.

Another form of disc or cup wheel is the Besly wheel, which consists of a ring or segments of abrasive, cemented to a thin ribbed steel back, through which studs project for attaching it to the adaptor or face plate of the grinding machine. This type of wheel also permits 100% of the abrasive material to be used.

The Norton Co. recommends for surfacing the various grades and types of disc wheels shown in Table 8.

**ALLOWANCE FOR GRINDING.**—The amount of material allowed for removal by grinding varies with the character of the work and the type of machine. In cylindrical grinding it may range from 0.020 to 0.025 in. In internal grinding, allowances have ranged, depending on the class of work, from 0.006 to 0.008 in. and from 0.012 to 0.017 in.

**GRINDING TOLERANCES** have decreased with improvement in grinding machines and in grinding wheels. On cylindrical grinding tolerances in diameter of − 0.0000 to + 0.0005 in. are possible of attainment, and in out of roundness and in taper, tolerances of 0.000 05 have been specified. Tolerances smaller than these fall within the fields of lapping and honing.

**INTERNAL GRINDING OF SMALL AND MEDIUM SIZE HOLES.**—C. R. Alden describes (*Trans.* A.S.M.E., MSP 51–15, 1929) the grinding of holes as small as $^1/_{16}$ in. diam. An air turbine was used to give a spindle speed of 65,000 r.p.m. The wheel was a mounted point, that is, an abrasive tip cemented on a steel shank, substantially less in diameter than the tip, which in turn was smaller than the hole to be ground. An important consideration in the internal grinding of small holes is the type of grinding spindle and the bearings used. A solid projection type of spindle, in which the wheel is mounted on an integral projection of the spindle, was found to be best. The projection should be of the largest possible diameter and shortest possible length consistent with the diameter of the hole to be ground. This will limit the amount of wheel wear that can be permitted, since allowance of excessive wheel wear will necessitate a spindle and bearings

**Table 8.—Disc Wheels for Surfacing**

(Recommendations of Norton Co., Worcester, Mass.)

| Material Ground | Wheel | Type | Bond |
|---|---|---|---|
| Aluminum Castings. | 3720 I4T–2 | Crystolon | Bakelite |
| Axe Heads.......... | 20/1 O5T–2 | Alundum | Bakelite |
| Bolt Heads......... | 46/2 N4T–2 | Alundum | Bakelite |
| Brass and Bronze | | | |
| Rough.......... | 3720 M4T–2 | Crystolon | Bakelite |
| Finish.......... | 3760 J8 | Crystolon | Vitrified<br>#12 Tr. |
| Cast Iron | | | |
| Large Discs<br>Large Work .. | 3714 O5T–2<br>3714 M5 | Crystolon<br>Crystolon | Bakelite<br>Vitrified |
| Small Work.... | 3724 K4T–2<br>3730 M5 | Crystolon<br>Crystolon | Bakelite<br>Vitrified |
| Dies (Hardened) | | | |
| Rough.......... | 1924 I4T–2 | Alundum | Bakelite |
| Finish.......... | 46 J6T–2 | Alundum | Bakelite |
| Fiber............ | 3714 J4T–2 | Crystolon | Bakelite |
| Glass Plates....... | 37120 K9 | Crystolon | Vitrified |
| Edges .......... | 37120 O6T–2 | Crystolon | Bakelite |
| Malleable Castings . | 24/2 L5T–2 | Alundum | Bakelite |
| Piston-pin Ends.... | 46 M3T–2 | Alundum | Bakelite |
| Piston Rings | | | |
| Roughing........ | 3724 L5T–2 | Crystolon | Bakelite |
| Finishing........ | 3760 M6T–2 | Crystolon | Bakelite |
| Porcelain.......... | 37100 J8T–2 | Crystolon | Bakelite |
| Refractory Brick... | 3716 N5T–2 | Crystolon | Bakelite |
| Rubber (Hard)..... | 3760 M5 | Crystolon | Bakelite |
| Roller Bearing Roller Ends.............. | 1980 O5 | Crystolon | Vitrified |
| Roller Bearing Cages.. | 36 O3S | Alundum | Silicate |
| Springs (coil), Automotive | | | |
| Heavy Wire........ | 1916 P5B | Alundum | Vitrified |
| Medium Wire...... | 30 O5B | Alundum | Vitrified |
| Small Wire....... | 1946 O5B<br>1960 M5 | Alundum<br>Alundum | Vitrified<br>Vitrified |
| Extra Heavy (Railroad Type).. | 12 M4T–2<br>1920 O8B | Alundum<br>Alundum | Bakelite<br>Vitrified |
| Leaf (eye grind).... | 16/2 R7B | Alundum | Vitrified |
| Steel Forgings (Hard) | | | |
| Rough............. | 24 U5 | Alundum | Vitrified |
| Finished........... | 36 MS | Alundum | Silicate |
| Steel (Hardened)... | 3824 G6S<br>30 K5T–2 | Alundum<br>Alundum | Silicate<br>Bakelite |
| Steel (Soft) | | | |
| Roughing.......... | 16 M4T–2 | Alundum | Bakelite |
| Commercial Finish.. | 30 O6T–2 | Alundum | Bakelite |
| Fine Finish........ | 60 K5T–2 | Alundum | Bakelite |
| Stove Castings....... | 3714 M4T–2 | Crystolon | Bakelite |
| Tile................. | 3746 M5 | Crystolon | Vitrified |
| Wrench Heads...... | 36 P5T–2<br>36 L5B | Alundum<br>Alundum | Bakelite<br>Vitrified |

that will lack rigidity. It is recommended that 2-in. wheels for grinding 2-in. holes be allowed $^1/_2$ in. reduction of diameter by wheel wear, and that they then be transferred to the grinding of 1 $^1/_2$-in. holes, with a further allowance of $^3/_8$ in. for wheel wear, after which they could be used for the grinding of 1 $^1/_8$ in. or 1-in. holes. The wheel thus can be used up, and at the same time the conditions under which it works will be those of maximum efficiency. Fig. 1 shows the spindle speeds recommended for the grinding of small holes.

**GRAIN DEPTH OF CUT.**—An analysis of the action of the wheel when in operation shows how theoretical considerations bear out the truth of the empirical rules for the use of grinding wheels in machine grinding. A paper by Geo. I. Alden (*Trans.* A.S.M.E., xxxvi, p. 451, 1914) gives the essential distinction between the radial or real depth at which the wheel cuts and the depth which the abrasive grain in the wheel cuts into the material being ground. The latter depth is termed the "grain depth of cut." This grain depth of cut is the controlling factor in securing the correct working of the wheel. A formula is deduced for computing the grain depth of cut, the application of the analysis is explained and these conclusions reached by Prof. Alden: 1. Other factors remaining constant, increase of work speed increases grain depth of cut, and makes a wheel appear softer. 2. A decrease of wheel speed increases grain depth of cut. 3. Diminishing the diameter of the wheel increases the grain depth of cut; increasing the diameter of the wheel decreases the grain depth of cut. 4. Decreasing the diameter of work increases the grain depth of cut; conversely, increasing the diameter of work decreases the grain depth of cut. A table of arcs of contact of wheel and work for a limited range of diameters is given, also a table of values of one of the factors in the formula for grain depth of cut.

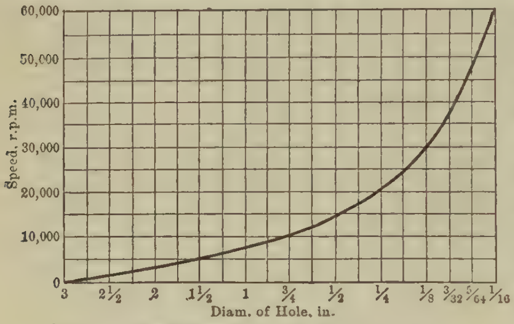


Fig. 1. Wheel Speeds for Grinding Small Holes

W. H. Chapman (*Trans.* A.S.M.E., xlii, p. 595, 1920), basing his study on Prof. Alden's paper set forth above, gives a mathematical analysis of the laws involved in cylindrical grinding and an analysis for grinding action for both draw-in and traverse cuts, deriving formulas and from them drawing practical conclusions as to the best methods to follow for maximum grinding efficiency.

**Draw-in Cuts.**—For draw-in cuts, the maximum depth of chip, in inches, corresponding to Prof. Alden's grain depth of cut, is given by

$$G = (\mu u/V)\,(\sqrt{2f/r}),\ \text{approximately} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where $G$ = maximum depth of chip, in., $\mu$ = intervals between grains of wheel in fractions of an inch, $u$ = work speed, ft. per min., $V$ = surface speed of wheel, ft. per min., $f$ = depth of cut, in., on work radius, $r$ = radius of work, in.

The index of the wheel-wearing action, $W_1$, due to grinding, is

$$W_1 = (\mu u/V^2)\,(\sqrt{2f/r}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

The actual wheel-wear can be computed by multiplying the value of $W_1$ by a constant derived by experiment, which may be taken as $(1800 \times 10^4)$. The amount of metal removed per minute may be determined by

$$P = 2\pi r f w \times \text{r.p.m.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where $P$ = cu. in. of metal removed per min., $w$ = width of wheel, in., and r.p.m. = speed of rotation of the work, other notation as before.

Table 9 compares wheel-wear and quantity of metal removed at different work speeds in a given machine, as calculated from the above formulas. Note that wheel-wear values increase rapidly as work speed increases. The value of 0.0166 for work 1 in. diameter at 53 r.p.m. becomes 0.1040 for work 4 in. diameter at 167 r.p.m. A wheel sufficiently soft to cut freely in the first case would wear so fast as to be nearly useless in the second case. Where the wheel-wear values are nearly the same, however, the same wheel will act properly in both cases, as, for instance, on 1-in. work at 138 r.p.m. and 4-in. work at 72.5 r.p.m. From Table 9, and by experimental tests with various wheels, certain ranges of wheel-wear values may be found to correspond to different grades of suitable grain size and abrasive for the material to be ground, and wheel-wear indices may be computed for any set of conditions to enable the proper wheel to be selected.

Production can be increased with a minimum increase in wheel-wear by an increase in the feed, up to the limit of the grain penetration. An example in the paper shows that doubling the feed approximately doubled the production but increased the wheel-wear only 35%. Doubling the work speed, on the contrary, doubled both production and wheel-wear.

**Traverse Cuts.**—The index of wheel-wear for traverse cuts is

$$W_t = (\mu u/V^2)\{u/(\sqrt{T^2 + u^2})\}\,(\sqrt{2f/r}), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where $T$ = traverse speed, ft. per min., other notation being as before. The expression $\{u/(\sqrt{T^2 + u^2})\}$ is known as the traverse factor. Where $T$ is large with respect to $u$ the factor will be appreciably less than unity, but as $u$ increases the reduction in wheel-wearing action is less pronounced. A table of wheel-wear values for traverse cuts can be made by multiplying the traverse factors determined as above by the wheel-wear value for the corresponding work speed and work diameter for draw-in cuts, such as are given in Table 9. Table 10 is such a table compiled for the same machine as was Table 9. By assigning certain wheels to specified ranges of wheel-wearing action, the proper wheel for traverse cuts may be selected from Table 10. The wheel used, however, would be some-

**Table 9.—Comparison between Wheel Wear and Quantity of Metal Removed at Different Work Speeds**

Wheel diam., 18 in.; surface speed, 6000 ft. per min.; diametral feed per work revolution (automatic feed), 0.001 in.; grain of wheel, 46 Alundum; material ground, mild machinery steel.

| Work Speed, r.p.m. | | Work Diameters | | | |
|---|---|---|---|---|---|
| | | 1 in. | 2 in. | 3 in. | 4 in. |
| 53 | Production * | 0.0835 | 0.1670 | 0.2500 | 0.3340 |
| | Wheel Wear | .0166 | .0236 | .0290 | .0334 |
| 72.5 | Production * | .1140 | .2280 | .3420 | .4560 |
| | Wheel Wear | .0226 | .0319 | .0390 | .0450 |
| 87 | Production * | .1370 | .2740 | .4110 | .5480 |
| | Wheel Wear | .0273 | .0387 | .0475 | .0546 |
| 101 | Production * | .1586 | .3172 | .4758 | .6344 |
| | Wheel Wear | .0316 | .0448 | .0540 | .0635 |
| 138 | Production * | .2160 | .4320 | .6480 | .8640 |
| | Wheel Wear | .0433 | .0612 | .0645 | .0865 |
| 167 | Production * | .2620 | .5240 | .7860 | 1.0480 |
| | Wheel Wear | .0522 | .0740 | .0910 | 0.1040 |

* In cu. in. of metal removed per minute.

**Table 10.—Wheel-Wear Values for Traversed Cuts**

(For same conditions as given for Table 9)

| Traverse, ft. per min. | Work r.p.m. | Lead, in. | Work Diameters | | | |
|---|---|---|---|---|---|---|
| | | | 1 in. | 2 in. | 3 in. | 4 in. |
| 10.02 | 53 | 2.3 | 0.01340 | 0.02240 | 0.02820 | 0.03280 |
| 14.00 | 53 | 3.1 | .01170 | .02080 | .02740 | .03230 |
| | 72.5 | 2.3 | .01820 | .02450 | .03775 | .04390 |
| 17.00 | 53 | 3.8 | .01050 | .02005 | .02685 | .03185 |
| | 72.5 | 2.8 | .01650 | .02900 | .03720 | .04370 |
| | 87 | 2.3 | .02190 | .03620 | .04620 | .05390 |
| 19.7 | 53 | 4.4 | .00955 | .01930 | .02620 | .03142 |
| | 72.5 | 3.2 | .01560 | .02820 | .03675 | .04350 |
| | 87 | 2.7 | .02060 | .03540 | .04570 | .05355 |
| | 101 | 2.3 | .02540 | .04185 | .05245 | .06260 |
| 26.9 | 53 | 5.9 | .00763 | .01695 | .02430 | .03000 |
| | 72.5 | 4.4 | .01290 | .02600 | .03520 | .04240 |
| | 87 | 3.6 | .01760 | .03320 | .04420 | .05235 |
| | 101 | 3.1 | .02220 | .03980 | .05125 | .06185 |
| | 138 | 2.3 | .03470 | .05130 | .06195 | .08500 |
| 32.6 | 53 | 7.35 | .00660 | .01545 | .02290 | .02885 |
| | 72.5 | 5.3 | .01140 | .02420 | .03380 | .04130 |
| | 87 | 4.4 | .01580 | .03150 | .04315 | .05160 |
| | 101 | 3.8 | .02000 | .03820 | .05000 | .06100 |
| | 138 | 2.8 | .03230 | .05580 | .06095 | .08450 |
| | 167 | 2.3 | .04210 | .06940 | .08800 | .10230 |

what softer in the traverse cut for the same index figure, because the actual depth of cut does not follow the indicated depth, as shown by the feeding mechanism, as closely for traverse work as for draw-in work. The following wheels are found to be satisfactory for traverse cuts on mild steel under the assumed conditions:

| Wheel Designation.... | 36 or 46I | 36 or 46J | 36 or 46K | 36 or 46L | 36 or 46M |
|---|---|---|---|---|---|
| Wheel-Wear Values, in Table 9............ | 0.0250 | .0250–.0400 | .0400–.0500 | .0500–.0600 | .0600 |

**Practical Conclusions.**—The practical conclusions drawn from the foregoing mathematical analysis are: The power drive must maintain the speed constant except as changed by the operator. The wheel must be in good running balance and in absolute truth, and held in position relative to the work within the closest possible limits. The work must be accurately held with respect to the wheel and must be uniformly rotated. The relative traverse between work and wheel must be uniform. The work must be rigidly supported over its entire length and no vibration allowed to occur between centers. Feeding must be at a rate such that the feed increment never exceeds the maximum grain penetration. Work must be kept at a uniform temperature and local heating prevented. A copious supply of grinding compound should be directed to the arc of contact at all times. In truing a wheel with a diamond, the cooling medium is vital to accuracy. Never feed a diamond over 0.001 in. per traverse. Do not try to use a single wheel for a variety of work sizes or materials.

## 3. CARE AND USE OF GRINDING WHEELS

**CARE OF GRINDING WHEELS.**—Wheels should be examined upon receipt to make sure that they have not been damaged in transit or otherwise. Each wheel should be held free and clear and tapped lightly with the handle of a screw driver. A wheel in good condition will give a clear ringing sound, while the absence of such ringing indicates a cracked wheel. Damp wheels or those in which the spaces between the grains are filled with sawdust, and also rubber and shellac-bonded wheels, will not ring when tapped.

Wheels should be stored in dry places of even temperature. Shellac- and rubber-bonded wheels, $^1/_4$ in. or less thick, should be laid flat on a straight surface to prevent warping. All other wheels should be supported on edge in racks. Before mounting, all wheels should be inspected again, as a wheel sometimes may crack merely by falling over from an upright position. The same " ring " test should be applied before mounting.

**OPERATING CONDITIONS. Work Rest.**—The work rest should be adjusted close to the work at all times, the maximum space between wheel and rest being $^1/_8$ in. The work rest should be of rigid construction and should be securely clamped after each adjustment. Adjustments should not be made while the wheel is in motion. Many wheel breakages are caused by work being caught between wheel and rest.

**Truth and Balance.**—The keeping of wheels true should be the function of especially experienced men. An inexperienced man frequently puts a wheel in worse condition than before. Wheels should be tested occasionally for balance. If out of balance, and the condition cannot be corrected by truing or dressing, the wheel should be removed from the machine. Wheels used for wet grinding should not be allowed to stand partly immersed in water, as the water absorbed may throw the wheel dangerously out of balance.

**Application of Work.**—When starting a wheel, the operator should stand aside until the wheel has run at full speed for at least one minute. Where work is applied continuously to the wheel, the heat generated should not be allowed to increase to a point where it will break the wheel. The generation of large amounts of heat may be overcome by a change in the grade of the wheel or by the application of a steady stream of water to the point of contact.

Grinding on the side of a straight wheel is prohibited in some places. Some classes of work, however, require the use of the side of the wheel for grinding. It is difficult to set a fixed rule as to when this practice is permissible, as each case must be considered individually. It should be remembered that thin wheels are not as strong as thicker ones and also that coarse wheels and those of softer grades can be broken more easily than the finer and harder wheels. If it is necessary continually to use the straight side of a wheel or to exert considerable pressure, cup wheels should be used.

**Eye Protection.**—Where danger of eye injury exists, operators should be provided with goggles. These should be so constructed that the lenses will be held firmly in the frame in case of breakage. To protect against fine flying dust, the frame should fit snugly around the eyes, following the contour of the face. The type of goggle held in place by an elastic head band has been found more satisfactory than those held in place with metal temple bows.

Shields of plate glass held in a metal frame fastened to the machine just above the point on the wheel where the work is done have proved quite satisfactory for certain classes of work.

**Wheel Mounting.**—Wheel flanges should be made in accordance with the dimensions in Tables 11, 12, 16 and 17. The inner flange should be keyed, shrunk or screwed to the spindle. The outer flange always should be of exactly the same diameter as the inner flange. The hole in the outer flange should be an easy sliding fit on the spindle or arbor. All flanges should be relieved or recessed at the center for at least 1/16 in. in order that the flanges may bear at the outer edge only. Flanges of the adapter and sleeve types, Table 12, should be recessed so that there is no bearing on the wheel within 1/8 in. of the hole. All flanges must be true and in balance.

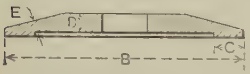


**Table 11.—Dimensions of Standard Cast-iron Flanges for Grinding Wheels**

A.S.A. Safety Code for Abrasive Wheels, 1935

| Wheel Diam. in. | B Min. Outside Diam. of Flanges, in. | C Radial Width of Bearing Surface, in. Min. | C Max. | D Min. Thickness of Flange at Bore, in. | E Min. Thickness of Flange at Edge of Recess, in. | Wheel Diam. in. | B Min. Outside Diam. of Flanges, in. | C Radial Width of Bearing Surface, in. Min. | C Max. | D Min. Thickness of Flange at Bore, in. | E Min. Thickness of Flange at Edge of Recess, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 3/8 | 1/16 | 1/8 | 1/16 | 1/16 | 16 | 5-1/2 | 1/2 | 1 | 1/2 | 5/16 |
| 2 | 3/4 | 1/8 | 3/16 | 1/8 | 3/32 | 18 | 6 | 1/2 | 1 | 5/8 | 3/8 |
| 3 | 1 | 1/8 | 1/4 | 3/16 | 3/32 | 20 | 7 | 5/8 | 1 1/4 | 5/8 | 3/8 |
| 4 | 1 1/4 | 3/16 | 3/8 | 3/16 | 1/8 | 22 | 7 1/2 | 5/8 | 1 1/4 | 5/8 | 7/16 |
| 5 | 1 1/2 | 3/16 | 3/8 | 1/4 | 1/8 | 24 | 8 | 3/4 | 1 1/4 | 5/8 | 7/16 |
| 6 | 2 | 1/4 | 1/2 | 3/8 | 3/16 | 26 | 8 1/2 | 3/4 | 1 1/4 | 5/8 | 1/2 |
| 8 | 3 | 1/4 | 1/2 | 3/8 | 3/16 | 28 | 10 | 7/8 | 1 1/2 | 3/4 | 1/2 |
| 10 | 3 1/2 | 5/16 | 5/8 | 3/8 | 1/4 | 30 | 10 | 7/8 | 1 1/2 | 3/4 | 5/8 |
| 12 | 4 | 5/16 | 5/8 | 1/2 | 5/16 | 36 | 12 | 1 | 2 | 7/8 | 3/4 |
| 14 | 4 1/2 | 3/8 | 3/4 | 1/2 | 5/16 | ...... | ....... | ...... | ...... | ....... | ....... |

**Table 12.—Dimensions of Flanges for Straight-side Grinding Wheels with Holes Larger than Machine Arbor**

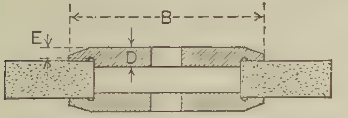


FIG. 2. Adapter Sleeves

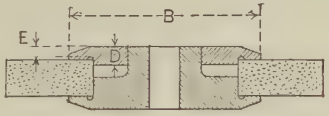


FIG. 3. Adapter Sleeves

A.S.A. Safety Code for Abrasive Wheels, 1935
All dimensions in inches.

Adapter Flanges. See Fig. 2

| Wheel Diam. | Wheel Hole Diam. | B Min. Diam. of Flange | D Min. Thickness at Bore | E Min. Thickness at Edge of Recess |
|---|---|---|---|---|
| 12 to 14 | 4 | 6 | 5/8 | 3/8 |
| | 5 | 7 | 5/8 | 3/8 |
| | 6 | 8 | 5/8 | 3/8 |
| Over 14 to 18 | 4 | 6 | 5/8 | 3/8 |
| | 5 | 7 | 5/8 | 3/8 |
| | 6 | 8 | 5/8 | 3/8 |
| | 7 | 9 | 5/8 | 3/8 |
| | 8 | 10 | 5/8 | 3/8 |
| Over 18 to 24 | 6 | 8 | 3/4 | 1/2 |
| | 7 | 9 | 3/4 | 1/2 |
| | 8 | 10 | 3/4 | 1/2 |
| | 10 | 12 | 3/4 | 1/2 |
| | 12 | 14 | 3/4 | 1/2 |
| Over 24 to 36 | 12 | 15 | 3/4 | 1/2 |

Sleeve Flanges. See Fig. 3

| Wheel Diam. | Wheel Hole Diam. | B Min. Diam. of Flange | D Min. Thickness at Bore | E Min. Thickness at Edge of Recess |
|---|---|---|---|---|
| 12 to 14 | 5 | 7 | 1/2 | 7/16 |
| Over 14 to 20 | 5 | 7 | 5/8 | 7/16 |
| | 6 | 8 | 5/8 | 7/16 |
| | 8 | 10 | 5/8 | 7/16 |
| | 10 | 11 1/2 | 5/8 | 7/16 |
| | 12 | 13 1/2 | 5/8 | 7/16 |
| Over 20 to 30 | 8 | 10 | 3/4 | 1/2 |
| | 10 | 11 1/2 | 3/4 | 1/2 |
| | 12 | 13 1/2 | 3/4 | 1/2 |
| | 16 | 17 1/2 | 3/4 | 1/2 |
| Over 30 to 36 | 12 | 14 1/2 | 3/4 | 1/2 |
| | 16 | 17 1/2 | 3/4 | 1/2 |
| | 18 | 19 1/2 | 3/4 | 1/2 |
| | 20 | 21 1/2 | 3/4 | 1/2 |

Compressible washers of blotting paper, rubber or leather should be used between the wheel and the flanges to compensate any slight unevenness in the surface of the wheel and to insure a uniform bearing around the entire periphery of the flange. Before mounting, both wheel and flange should be cleaned of foreign particles. The nuts should be tightened only enough to hold the wheel firmly. Excessive tightening may damage the wheel.

Other forms of mounting, such as cementing or sulphuring the wheel to iron backs and centers, or the clamping of segments to a metal ring are recommended only for special purposes and no definite specifications can be given. Mounting wheels with tapered holes and tapered arbors demands great care. Any forcing of the wheel necessary to get it in position tends to crack it.

**OPERATING SPEED.**—All types of wheels do not possess equal strength, and consequently it is not safe to operate them all at equal speeds. As a general rule, vitrified and silicate bonded wheels are classified in one group and the organic bonded wheels (shellac, rubber, and Bakelite) in another. Vitrified and silicate wheels are relatively brittle and a maximum operating speed of 6500 surface feet per minute has been established for this group. Wheels in the softer grades and of coarse grain size should not be operated as fast as this. The organic bonded wheels are more elastic and can be operated at higher speeds with safety. A maximum operating speed of 9500 surface feet per minute has been established for this group, with the exception of thin cutting-off wheels which may be operated up to 16,000 surface feet per minute under certain conditions. In the organic group as in the vitrified and silicate group, the operating speeds for the softer and coarser wheels should be lower.

The manufacturers of grinding wheels know from experience the maximum speeds at which the various wheels they produce should be operated. It is the usual custom to show this maximum operating speed on tabs accompanying the wheels. It is also customary to test all wheels 6 in. and larger in diameter at speeds sufficiently higher to insure a reasonable factor of safety.

Table 13 shows the maximum peripheral speed per minute permitted by the A.S.A. Safety Code for various types of wheels and bonds.

**FACTOR OF SAFETY.**—Since the centrifugal force tending to break a wheel increases as the square of the speed, it is, at 9000 surface ft. per min. 2.3 times greater than at 6000 surface ft. per min., and 3.3 times greater than at 5000 surface ft. per min. These values, 2.3 and 3.3, are called the established factor of safety for the operating speeds mentioned.

$$\text{Established factor of safety} = \text{testing speed}^2 \div \text{operating speed.}^2$$

$$\text{True factor of safety} = \text{breaking speed}^2 \div \text{operating speed.}^2$$

The true factor of safety cannot be calculated unless the breaking speed actually is measured. Experience shows the true factor of safety to be, usually, many times greater than the established factor.

The forces tending to break grinding wheels are due to other causes in addition to speed of rotation. Among these are: Poor mountings, wheel flanges of unequal diameter, unrelieved flanges, expansion from excessive and rapid heating of one periphery, work catching between wheel and work rest, improper grinding on side of wheel and excessive out-of-truth, or out-of-balance conditions. These conditions are cumulative and increase the breaking stresses due to rotation. Since these conditions exist, the established factor of safety is kept large. If the grinding conditions are correct, the wheel which has been subjected to the speed test should not give trouble, as it has been established that it is

**Table 13.—Maximum Permissible Grinding Wheel Speeds**

| Type of Wheel | Vitrified and Silicate Bonds | | | Organic Bonds | | |
|---|---|---|---|---|---|---|
| | Soft | Medium | Hard | Soft | Medium | Hard |
| | Peripheral Speed, ft. per min. | | | | | |
| Straight<br>Taper | 5500 | 6000 | 6500 | 6500 | 8000 | 9,500 |
| Recessed | 5500 | 6000 | 6500 | 6500 | 8000 | 9,500 |
| Cylinder | 4500 | 5500 | 6000 | 6000 | 8000 | 9,500 |
| Dovetail<br>Dish and Flaring Cup<br>Saucer | 4500 | 5500 | 6000 | 6000 | 8000 | 9,500 |
| Deep Recessed Cup | 4500 | 5000 | 5500 | 6000 | 7500 | 9,000 |
| Cutting Wheels: Over 16 in. diam. | ........ | ........ | ........ | ........ | ........ | 7,500 to 14,000 |
| Cutting Wheels: Up to 16 in. diam. | ........ | ........ | ........ | ........ | ........ | 10,000 to 16,000 |

**Table 14.—Minimum Sizes of Machine Spindles for Various Diameters and Thicknesses of Grinding Wheels**

| Diam. of Wheel, in. | Thickness of Wheel, in. | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/4 | 3/8 | 1/2 | 5/8 | 3/4 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/4 | 2 1/2 | 2 3/4 | 3 | 3 1/4 | 3 1/2 | 4 | 4 1/2 | 5 |
| | Diameter of Spindle, in. | | | | | | | | | | | | | | | | | | |
| 6 | 1/2 | 1/2 | 1/2 | 1/2 | 1/2 | 1/2 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 1 | 1 | 1 |
| 7 | 1/2 | 1/2 | 1/2 | 1/2 | 5/8 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 1 | 1 | 1 | 1 | 1 | 1 |
| 8 | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/4 |
| 9 | 5/8 | 5/8 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 3/4 | 1 | 1 | 1 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 |
| 10 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 1 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 |
| 12 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 1 | 1 | 1 | 1 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 | 1 1/2 |
| 14 | 7/8 | 7/8 | 7/8 | 7/8 | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 |
| 16 | .... | .... | .... | .... | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 3/4 | 1 3/4 | 1 3/4 | 1 3/4 |
| 18 | .... | .... | .... | .... | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 3/4 | 1 3/4 | 1 3/4 | 1 7/8 | 1 7/8 |
| 20 | .... | .... | .... | .... | .... | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 1/2 | 1 3/4 | 1 3/4 | 1 3/4 | 1 7/8 | 1 7/8 | 1 7/8 | 1 7/8 |
| 24 | .... | .... | .... | .... | .... | 1 1/2 | 1 1/2 | 1 1/2 | 1 3/4 | 1 3/4 | 1 3/4 | 1 3/4 | 1 3/4 | 1 3/4 | 2 | 2 | 2 | 2 | 2 |
| 26 | .... | .... | .... | .... | .... | .... | 1 1/2 | 1 1/2 | 1 3/4 | 1 3/4 | 1 3/4 | 1 3/4 | 2 | 2 | 2 | 2 | 2 1/4 | 2 1/4 | 2 1/4 |
| 30 | .... | .... | .... | .... | .... | .... | .... | 1 3/4 | 1 3/4 | 2 | 2 | 2 | 2 | 2 | 2 1/4 | 2 1/4 | 2 1/2 | 2 1/2 | 2 1/2 |
| 36 | .... | .... | .... | .... | .... | .... | .... | .... | 2 | 2 1/4 | 2 1/4 | 2 1/4 | 2 1/2 | 2 1/2 | 2 1/2 | 2 3/4 | 2 3/4 | 3 | 3 |

sufficiently strong to withstand the stress due to ordinary grinding conditions. A thorough investigation of a grinding wheel broken in operation usually will show the breakage to have been due to faulty operating conditions and not to the wheel itself.

**SIZE OF WHEEL SPINDLE.**—The minimum diameter of machine spindles which should be used with wheels of various sizes is given in Table 14. Bearing boxes must be of sufficient length to provide ample bearing surface.

**GRINDING WHEEL TROUBLES AND REMEDIES.**—The following is a summary of the common troubles of grinding wheel operation and the remedies therefor:

**Wheel Does Not Cut.**—1. If running too fast reduce speed of wheel. 2. Use a softer wheel. 3. Use a coarser wheel if finish is unimportant. 4. Be sure that abrasive is correct for material being ground.

**Wheel Wears Too Rapidly.**—1. If running too slow increase wheel speed. 2. Use harder wheel. 3. Use finer wheel. 4. Eliminate vibration in machine. 5. Reduce pressure of work on wheel.

**Wheel Does Not Give Good Finish** (precision grinding).—1. Dress wheel to a perfect cylinder. 2. Put smooth face on wheel with diamond. 3. Eliminate vibration of machine spindle. 4. Balance wheel if it is out of balance. 5. Use finer wheel. 6. Use shellac or rubber wheel.

**Wheel Loads.**—1. Increase speed if below that recommended. 2. Use coarser wheel. 3. Use softer wheel.

**Wheel Glazes.**—1. Reduce speed if above that recommended. 2. Use softer wheel. 3. Be sure correct abrasive is being used.

**Wheel Burns.**—See Wheel Does Not Cut.

**Wheel Does Not Hold Shape.**—See Wheel Wears too Rapidly.

## 4. PROTECTION DEVICES

Abrasive wheels should be guarded against breakage by protection hoods, protection flanges, protection bands or protection chucks.

**PROTECTION HOODS** provide the best protection and should be used wherever the work will permit. Few cases will be found where it is not possible to use some sort of hood. A properly designed hood, securely fastened to the base of the machine or floor, will retain the parts of a broken wheel and provides as nearly absolute protection to the operator as is possible. It is the only device affording protection against small particles of a broken wheel. Unless a hood is properly designed, however, it will give a false sense of security and thus may be worse than no protection at all. A hood should be strong enough to retain the parts of a broken wheel both at normal speed and high overspeed.

The A.S.A. Safety Code for the protection of abrasive wheels specifies the exposures of grinding wheel periphery shown in Figs. 4 to 9 for the various types of grinders. On bench and floor stands, exposure shall not exceed 90 deg., except in those cases where the nature of the work requires contact with the wheel below the horizontal, in which case the exposure shall not exceed 125 deg. See Fig. 5. Exposure in both cases shall begin not more than 65 deg. above the horizontal. For top grinding, *i.e.*, where the work is ground on the top of the wheel, the exposure shall be as small as practicable, with a maximum of 60 deg. Hoods shall be so constructed that the protecting member can be adjusted to the constantly decreasing diameter of the wheel by means of an adjustable

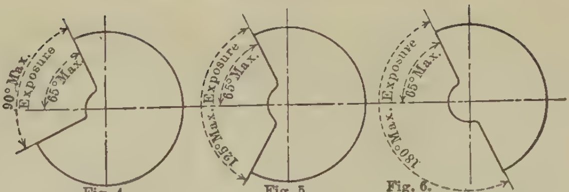


Fig. 4. Bench and Floor Stand

Fig. 5. Bench and Floor Stand

Fig. 6. Cylindrical Grinder

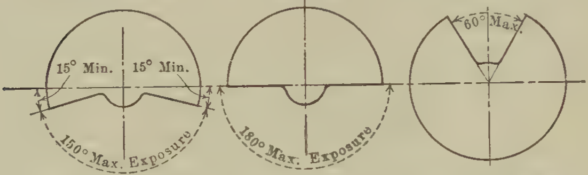


Fig. 7. Surface Grinders and Cutting Machines

Fig. 8. Swing Frame and Portable Grinders

Fig. 9. Top Grinding

FIGS. 4 to 9. Protection Hood Exposures

tongue or its equivalent so that the angular protection shown in Figs. 4 to 9 will be maintained throughout the life of the wheel, and the maximum distance between the wheel periphery and the adjustable member will not exceed 1/4 in. See Figs. 10 to 12. Table 15 gives the dimensions and material of the sides and peripheral members of hoods as fixed by the code. The code also presents drawings and dimensions of protection hoods constructed of structural steel.

**PROTECTION FLANGES.**—Tapered flanges when properly designed afford good protection and should be used where a hood is impracticable. They are used with tapered wheels, *i.e.*, wheels thicker at the hole than at the periphery. It is important that flanges be as large as possible and be changed as the wheel wears down close to the edge of the flange. Table 16, and Figs. 14 and 15 give dimensions of two satisfactory types of tapered flanges for wheels operating at a maximum peripheral speed of 6500 ft. per min. Fig. 16

(*Continued on page* 21–84)

**Table 15.—Minimum Dimensions of Peripheral and Side Members of Protection Hoods**

All dimensions in inches. See Fig. 13

| Material | Max. Thickness of Grinding Wheel | Grinding Wheel Diameters | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 3 to 6 in. | | 7 to 12 in. | | 13 to 16 in. | | 17 to 20 in. | | 21 to 24 in. | | 25 to 30 in. | | 31 to 48 in. | |
| | | *A* | *B* | *A* | *B* | *A* | *B* | *A* | *B* | *A* | *B* | *A* | *B* | *A* | *B* |
| **Speeds up to 7000 Peripheral ft. per min.** | | | | | | | | | | | | | | | |
| Cast Iron | 2 | 1/4 | 1/4 | 3/8 | 5/16 | 1/2 | 3/8 | 5/8 | 1/2 | 7/8 | 5/8 | 1 | 3/4 | 1 1/4 | 1 |
| | 4 | 5/16 | 5/16 | 3/8 | 5/16 | 1/2 | 3/8 | 3/4 | 5/8 | 1 | 5/8 | 1 1/8 | 3/4 | 1 3/8 | 1 |
| | 6 | 3/8 | 5/16 | 1/2 | 7/16 | 5/8 | 1/2 | 1 | 5/8 | 1 1/8 | 3/4 | 1 1/4 | 7/8 | 1 1/2 | 1 1/8 |
| Malleable Iron | 2 | 1/4 | 1/4 | 3/8 | 5/16 | 1/2 | 3/8 | 5/8 | 1/2 | 3/4 | 5/8 | 7/8 | 3/4 | 1 | 7/8 |
| | 4 | 5/16 | 5/16 | 3/8 | 5/16 | 1/2 | 3/8 | 5/8 | 1/2 | 3/4 | 5/8 | 7/8 | 3/4 | 1 1/8 | 7/8 |
| | 6 | 3/8 | 5/16 | 3/8 | 5/16 | 5/8 | 1/2 | 3/4 | 5/8 | 7/8 | 5/8 | 1 | 3/4 | 1 1/4 | 7/8 |
| **Speeds up to 10,000 Peripheral ft. per min.** | | | | | | | | | | | | | | | |
| Steel Castings | 2 | 1/4 | 1/4 | 1/4 | 1/4 | 3/8 | 3/8 | 1/2 | 7/16 | 5/8 | 1/2 | 3/4 | 5/8 | 7/8 | 3/4 |
| | 4 | 1/4 | 1/4 | 1/4 | 1/4 | 3/8 | 3/8 | 1/2 | 7/16 | 5/8 | 1/2 | 3/4 | 5/8 | 1 | 3/4 |
| | 6 | 3/8 | 1/4 | 3/8 | 1/4 | 1/2 | 3/8 | 5/8 | 1/2 | 3/4 | 5/8 | 7/8 | 3/4 | 1 1/8 | 3/4 |
| Structural Steel | 2 | 1/8 | 1/16 | 3/16 | 1/8 | 3/16 | 3/16 | 1/4 | 1/4 | 5/16 | 1/4 | 3/8 | 5/16 | 1/2 | 3/8 |
| | 4 | 1/8 | 1/16 | 3/16 | 1/8 | 3/16 | 3/16 | 1/4 | 1/4 | 5/16 | 1/4 | 3/8 | 5/16 | 1/2 | 3/8 |
| | 6 | 3/16 | 1/16 | 1/4 | 1/8 | 1/4 | 3/16 | 3/8 | 1/4 | 3/8 | 1/4 | 7/16 | 5/16 | 5/8 | 3/8 |
| Wrought Iron | 2 | 1/8 | 1/16 | 3/16 | 1/8 | 1/4 | 3/16 | 5/16 | 1/4 | 3/8 | 1/4 | 7/16 | 5/16 | 1/2 | 3/8 |
| | 4 | 1/8 | 1/16 | 1/4 | 1/8 | 1/4 | 3/16 | 5/16 | 1/4 | 3/8 | 1/4 | 7/16 | 5/16 | 5/8 | 3/8 |
| | 6 | 3/16 | 1/16 | 1/4 | 1/8 | 5/16 | 3/16 | 3/8 | 1/4 | 7/16 | 1/4 | 1/2 | 5/16 | 11/16 | 7/16 |
| **Cutting-off Wheels. 14,000 Peripheral ft. per min.** | | | | | | | | | | | | | | | |
| | | | | | | 12 to 16 in. | | 17 to 20 in. | | 21 to 24 in. | | 25 to 30 in. | | 32 to 36 in. | |
| | | | | | | *A* | *B* | *A* | *B* | *A* | *B* | *A* | *B* | *A* | *B* |
| Structural Steel | Up to 1/2 in. | | | | | 1/8 | 1/8 | 1/8 | 1/8 | 3/16 | 1/8 | 3/16 | 1/8 | 1/4 | 3/16 |

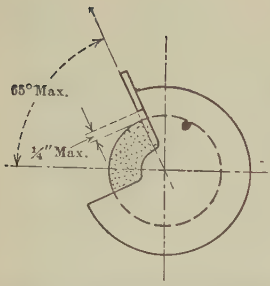


FIG. 10. Correct Adjustable Tongue, Giving Protection for All Sizes of Wheels

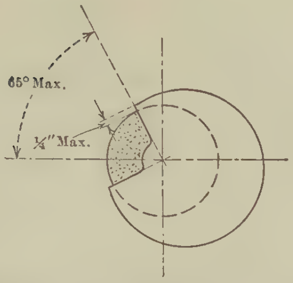


FIG. 11. Correct Movable Hood with Opening Small Enough for Protection with Smallest Wheel

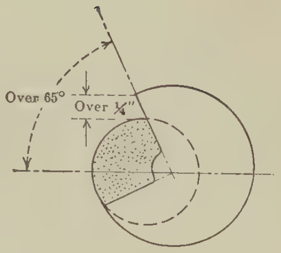


FIG. 12. Incorrect Movable Hood with Opening Correct for Full-sized Wheel but Too Wide for Smaller Wheel

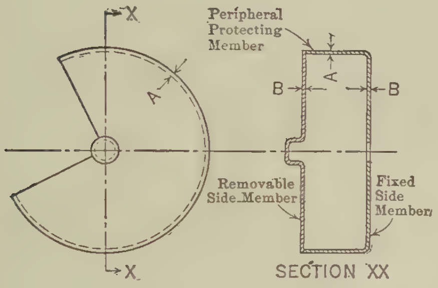


FIG. 13. Cast Protection Hood

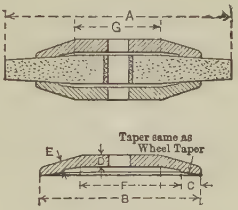


FIG. 14. Tapered Flange

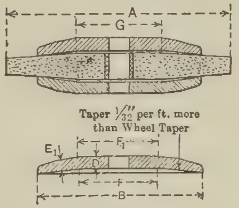


FIG. 15. Tapered Flange

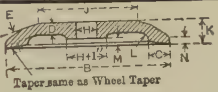
**Table 16.—Dimensions of Tapered Protection Flanges for Grinding Wheels, for Peripheral Speeds up to 6500 ft. per min.**

All dimensions in inches

| *A* | *B* | *C* | | *D* | *E* | *E*₁ | *F* | *F*₁ | *G* |
|---|---|---|---|---|---|---|---|---|---|
| Wheel Diam. | Min. Outside Diam. of Flanges | Radial Width of Bearing Surface, Fig. 14 Only | | Min. Thickness of Flange at Bore | Min. Thickness at Edge of Recess, Fig. 14 Only | Min. Thickness at Bevel, Fig. 15 Only | Diam. of Max. Flat Inside of Flange | Diam. of Flat Outside of Flange, Fig. 15 Only | Max. Diam. of Flat or Hub of Wheel |
| | | Min. | Max. | | | | | | |
| 6 | 3 | 1/4 | 1/2 | 3/8 | 3/16 | ...... | 0 | ...... | 1 |
| 8 | 4 | 5/16 | 5/8 | 3/8 | 1/4 | ...... | 0 | ...... | 1 |
| 10 | 5 | 1/2 | 1 | 1/2 | 1/4 | ...... | 0 | ...... | 2 |
| 12 | 6 | 1/2 | 1 | 5/8 | 5/16 | 1/2 | 4 | 5 | 4 1/2 |
| 14 | 8 | 5/8 | 1 1/4 | 5/8 | 3/8 | 1/2 | 4 | 5 | 4 1/2 |
| 16 | 10 | 3/4 | 1 1/2 | 5/8 | 3/8 | 1/2 | 4 | 6 | 6 |
| 18 | 12 | 1 | 2 | 3/4 | 1/2 | 5/8 | 4 | 6 | 6 |
| 20 | 14 | 1 1/4 | 2 1/2 | 3/4 | 1/2 | 5/8 | 4 | 7 | 6 |
| 22 | 16 | 1 3/8 | 2 3/4 | 3/4 | 9/16 | 5/8 | 4 | 7 | 6 |
| 24 | 18 | 1 1/2 | 3 | 3/4 | 9/16 | 5/8 | 4 | 8 | 6 |
| 26 | 20 | 1 1/2 | 3 1/4 | 3/4 | 5/8 | 5/8 | 4 | 8 | 6 |
| 28 | 22 | 1 3/4 | 3 3/4 | 7/8 | 5/8 | 5/8 | 4 | 8 | 6 |
| 30 | 24 | 2 | 4 | 7/8 | 3/4 | 3/4 | 4 | 8 | 6 |
| 36 | 28 | 2 | 4 | 1 | 7/8 | ...... | 4 | ...... | 6 |

**Table 17.—Dimensions of Tapered Protection Flanges for Grinding Wheels, for Peripheral Speeds of 6500 to 9500 ft. per min.**

All dimensions in inches

| Diam. of Wheel | Thickness of Wheel | *B* | *C* | |  <br>Taper same as Wheel Taper<br>FIG. 16. | | | | *L* | *M* | *N* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Max. | Min. | *D* | *E* | *J* | *K* | | | |
| 16 | 4 | 10 | 1 1/2 | 1 | 3/4 | 3/4 | 6 | 1 1/2 | 3/4 | 5/16 | 3/8 |
| 18 | 4 | 12 | 2 | 1 1/2 | 7/8 | 7/8 | 6 | 2 | 7/8 | 3/8 | 3/8 |
| 20 | 4 | 14 | 3 | 2 | 1 | 1 | 7 | 2 1/4 | 1 | 7/16 | 3/8 |
| 24 | 4 | 18 | 4 | 3 | 1 1/4 | 1 1/4 | 8 | 3 | 1 1/2 | 9/16 | 3/8 |
| 30 | 4 | 24 | 5 | 3 1/2 | 1 1/2 | 1 1/2 | 10 | 4 | 2 | 3/4 | 3/8 |

and Table 17 give the dimensions of flanges for peripheral speeds from 6500 to 9500 ft. per min. Tapered flanges over 10 in. diam. should be made of steel. All other flanges may be made of cast iron, or material of equal strength.

**PROTECTION CHUCKS** are used with cylinders or ring wheels. Any chuck so designed that the jaws can support the wheel up to a point not further distant from the grinding edge than 1/4 of the height of the wheel is considered satisfactory. Although protection chucks do not furnish absolute protection, the wheels used in them usually are run at comparatively low speeds and the danger is not great.

**PROTECTION BANDS** for cup, cylinder and sectional ring wheels should be made of steel plate or wrought iron, and conform as closely as practical to the periphery of the wheel. Ends shall be so riveted, bolted or welded as to leave no projections on the inside of the band. Thickness and rivet spacing are as follows (A.S.A. Code, 1935):

| Wheel diam., in. | Under 8 | 8 to 24 | 25 to 30 |
|---|---|---|---|
| Thickness of band, in. | 1/16 | 1/8 | 1/4 |
| Min. diam. of rivets | 3/16 | 1/4 | 3/8 |
| Max. center distance of rivets | 3/4 | 1 | 1 1/4 |

If band is bolted or welded, strength of the connection should at least equal that of the above riveting.

## 5. GRINDING MACHINES

**TYPES OF GRINDING MACHINES.**—Grinding machines may be classified as cylindrical, internal, and surface grinders, based on the general character of the work. Each of these classes may be further subdivided according to special characteristics either of work or machine. The following paragraphs show the types of grinding machines most commonly used, together with their fields and limitations.

**External Cylindrical Grinding Machines** for general purpose grinding of cylindrical surfaces, as shafts, axles, etc., range from 6 to 16 in. swing. Standard sizes are 6 × 18 in.; 6 × 30 in.; 10 × 18 in.; 10 × 36 in.; 10 × 48 in.; 10 × 72 in.; 10 × 96 in.; 10 × 120 in.; 14 × 36 in.; 14 × 48 in.; 14 × 72 in.; 14 × 96 in.; 14 × 120 in.; 14 × 144 in.; 16 × 36 in.; 16 × 48 in.; 16 × 72 in.; 16 × 96 in.; 16 × 120 in.; 16 × 144 in.

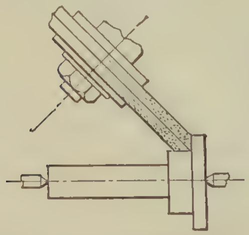


FIG. 17. Arrangement for Grinding Shoulders

Machines are built with hydraulic, mechanical or hand traverse or with all three, of both table and grinding wheel head. Grinding wheel and work drives usually are from separate motors through V-belts. For plunge cuts, *i.e.*, wheel fed perpendicular to work, steady rests and reciprocation of the wheel spindle are advisable. Cylinder grinders can be built to function automatically or semi-automatically, through a complete cycle, producing work ground within predetermined limits. They also may be fitted with attachments for cam, roll or crank-shaft grinding, although for regular production of this character, special machines are advisable. For grinding faces of shoulders and the adjacent cylindrical surfaces, cylindrical grinders are built with the wheel slide set at an angle of 45 deg. to the axis of the work, a V-shaped wheel being used. See Fig. 17.

**Roll Grinding Machines** are cylindrical grinders of 20, 24, 28, 32 and 36 in. swing by 96, 120, 144, 168, or 192 in. between centers. These are standard sizes. Special machines have been built up to 60 in. swing by 26 ft. long between centers. Roll grinding machines are built with the work traversing past a grinding wheel in a fixed position, or with the wheel traversing past the work. Rolls are crowned or concaved by moving the wheel, by appropriate mechanism, away from or toward the roll as the wheel traverses it.

**Cam and Shape Grinding Machines** are a special form of cylindrical grinder. In these machines, the work is rotated in a swinging member and is moved toward and away from the grinding wheel by means of a master cam in contact with a roller. The master cam is on the swinging member, the roller being fixed on the machine. Where the cams are integral with the cam shaft, they are carried on centers in the machine, one pair being in the master cam spindle and the other in a footstock mounted on the swinging member. Single cams are mounted in fixtures or on arbors carried by the master cam spindle. For high production, as automotive cam shafts, completely automatic machines are used. One type is a hydraulically-operated machine that successively grinds each cam to size, indexing after each grinding operation. When the last cam is ground, the table reverses, returns to the starting position and the work stops rotating. The wheel is trued automatically by a diamond, and its position changed to compensate for the truing. The table then stops in the loading position. This machine, made by the Norton Co., has a swing over the rocking bar of 6 in. Length of work between centers is 26, 30, 36 or 42 in.

**Centerless Grinders** are a special form of cylindrical grinder. Instead of the work being supported on centers, it is supported on a work rest, and held against a regulating wheel. See Fig. 18. By setting the regulating wheel at an angle to the grinding wheel, the former will act as a feeding device and will traverse the work past the grinding wheel. The regulating wheel is positively driven and its speed determines the rotational speed of the work. The centerless grinder operates on a diametral dimension instead of on a radial dimension of the work, as in grinders where the work is held on centers, and therefore under certain conditions it will remove double the amount of metal in a given time. On slender work, much higher grinding pressure can be used in the centerless machine, as the work is rigidly supported by the work rest and regulating wheel, whereas when supported on centers it will tend to spring away from the wheel under heavy pressure. Centerless grinders usually are built to take work up to 9 in. diam., and of any length.

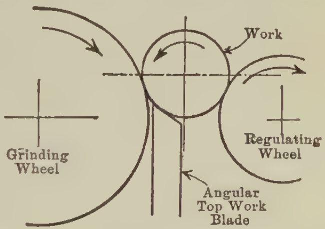


Fig. 18. Principle of Centerless Grinder

**Internal Cylindrical Grinders** comprise a reciprocating wheel head carrying a grinding wheel on a spindle long enough to reach to the bottom of the hole, and a work head in which the work can be rotated while in contact with the wheel, both carried on a suitable bed or frame, with provision for adjustment of both wheel and work. The field of the internal grinder is the finishing of holes in gears, bushings, bearings, etc., and the finishing of the inside surface of cylinders, as automotive cylinders. The cylinder grinder is a special form of internal grinder, and in one type the wheel head gives a planetary motion to the wheel spindle. H. L. Blood (*Trans.* A.S.M.E., MSP 54–1, 1932) describes an automatic internal grinder which rough grinds, trues the wheel, finish grinds the hole to size, runs the wheel out of the hole and brings all parts to the rest position. The range of capacity of internal grinders is holes from $^1/_4$ to 24 in. diam. and from 3 to 15 in. deep. Cylinder grinders are built to grind cylinders from 2 $^3/_8$ to 14 $^1/_2$ in. diam. by 11 to 18 in. long. Internal grinders are built by the Heald Machine Co., Worcester, Mass.

**Surface Grinders** are built in a variety of forms, *viz.*, with reciprocating or rotary tables, and with horizontal or vertical spindles. Grinding is done either with the face or side of the wheel, cup wheels being used in the latter case. Reciprocating tables usually are traversed hydraulically. In certain machines the rotating table can be set at an angle to permit grinding of concave or convex surfaces. Both direct motor drive and belt drive of grinding wheels are available. The capacity of the usual reciprocating table grinders ranges from work 6 × 18 in. to 14 × 60 in. Rotating-table surface grinders are built with tables up to 36 in. diam. Work usually is held on the table by magnetic chucks.

**Disc Grinders** are a form of surface grinder using abrasive discs mounted on a steel backing. The work is ground against the side of the disc. Single-spindle machines, with a disc on each end of the spindle, are built to use discs up to 40 in. diam. The work is supported on tables, equipped with either hand or power feed. The spindles are driven either direct from the motor, or through V-belts.

**The Double-spindle Disc Grinder** is a machine for grinding two parallel surfaces. It comprises, virtually, two single-end disc grinders, mounted on a common base, the wheel spindles being in line, with the wheels facing each other. The work, supported by a table, is fed in between the wheels, which are spaced the correct distance apart. Wheel spindles are driven direct from the motor or through V-belts. These machines are built in various sizes, with discs ranging from 15 to 72 in. diam., and with maximum opening between wheels ranging from 20 to 30 in.

**Face Grinders** are a form of surface grinder in which the work travels past the side of a wheel, carried on a horizontal spindle mounted at right angles to the line of travel of the work. The wheel grinds the vertical surface of the work at right angles to the surface of the table. The table is traversed hydraulically, and the wheel is driven by silent chain or V-belt. Work can be held by bolting to the table or to angle plates or by magnetic chucks. These machines are used for surface grinding and for the grinding of machine knives, shear blades, etc. Tables range from 22 to 27 in. wide and from 60 to 168 in. long.

**Universal Grinding Machines** are cylindrical grinders provided with a swiveling table, and swiveling headstocks and swiveling wheel slide. Their field is tool room work and manufacturing where a variety of parts in small lots are to be ground. They range in size from 12 to 16 in. swing and from 24 to 72 in. between centers.

**Tool and Cutter Grinding Machines** are designed primarily for grinding of reamers, milling cutters, etc. In their simplest form they comprise a motor-driven grinding wheel and a pair of centers upon which the work is mounted, carried on a table that can be traversed past the wheel by hand. More elaborate machines comprise a power-operated swiveling table, a power-driven headstock to permit the machine to be used for cylindrical grinding, and attachments that permit of internal grinding and various forms of surface grinding. The capacity of tool grinders ranges usually up to 12 in. diam. by 30 to 36 in. between centers.

### Table 18.—Horsepower of Grinding Machine Motors

| Type of Grinder | Size,* in. | Wheel | | | Work Speed | | Motor | |
|---|---|---|---|---|---|---|---|---|
| | | Diam., in. | Face, in. | R.p.m. | R.p.m. | Ft. per min. | Hp. | R.p.m. |
| Cam | 10 | 24 | ...... | ......... | ...... | ...... | 3/4–1 | 1200 |
| Car Wheel | 44 | 24 | 2 3/4 | 1050 | 9, 12 | ...... | 25† | 1200 |
| Centerless | 4 | 20 | 4 | ......... | ...... | ...... | 15 | 1200 |
| " | 6–9 | 24 | 6 | ......... | ...... | ...... | 25–30 | 1200 |
| Crank-pin | 17 | 42 | ...... | ......... | ...... | ...... | 20 | 1200 |
| Cutter | 10 | 5 1/2 | 5/16 | 3770 | ...... | ...... | 1/2 | 3600 |
| " | 12 | 8 | 1/2 | 2830 | ...... | ...... | 3/4 | 3600 |
| Cylinder | 2 3/8–3 | 2 3/4–3 1/2 | ...... | ......... | ...... | ...... | 7 1/2 | 900 |
| Cylindrical | 6 | 20 | 2 | ......... | 70 | ...... | 7 1/2 | 1800 |
| " | 10 | 24 | 3 | ......... | 65 | ...... | 7 1/2–15 | 1800 |
| " | 14 | 30 | 3 | ......... | 27–81 | ...... | 20–40 | 1800 |
| " | 16 | 30 | 3 | ......... | 50–150 | ...... | 20–40 | 1800 |
| Disc—1 spindle | ...... | 18 | ...... | 1375 | ...... | ...... | 5–7 1/2 | 1750 |
| " 1 " | ...... | 30 | ...... | 750 | ...... | ...... | 15–20 | 1750 |
| " 2 " | ...... | 15 | ...... | ......... | ...... | ...... | 5† | 1750 |
| " 2 " | ...... | 72 | ...... | ......... | ...... | ...... | 40† | 1160 |
| Face | 23 | 32 | 8‡ | 500 | ...... | 5–85 | 40–50 | 1200 |
| Floor | ...... | 24 | 4 | 1520–2120 | ...... | ...... | 10–15 | 1800 |
| " | ...... | 30 | 4 | 1245–1910 | ...... | ...... | 15 | 1800 |
| Internal | 1/4–1 | ...... | ...... | ......... | 500–1200 | 5 | ......... | 1800 |
| " | 9 1/4 | ...... | ...... | ......... | 160–480 | 5 | ......... | 3600 |
| " centerless | 1/4 | ...... | ...... | ......... | ...... | 7 1/2 | ......... | 1800 |
| " " | 12 | ...... | ...... | ......... | 80–160 | ...... | 2 | 1200 |
| " " | 20 | ...... | ...... | ......... | 50–200 | ...... | 7 1/2 | 3600 |
| " " | 24 | ...... | ...... | ......... | 35–105 | ...... | 25 | 1800 |
| Knife | 19 | 26 | 8‡ | 680 | ...... | 0–90 | 15–25 | 1200 |
| Roll | 20–36 | 30 | 3 | ......... | 6–24 | ...... | 20–30 | 1800 |
| Surface | 6 | 8 | 3/4 | 6000 | ...... | 36 | 1 1/2 | 1800 |
| " | 10 | 10 | 1 1/2 | 4700 | ...... | 100 | 7 1/2 | 1200 |
| " | 12–14 | 14 | 3 | 1200 | ...... | 30–100 | 15 | 1200 |
| " Vertical-spindle | 6 | 5 | 5/8§ | 4700 | ...... | 36 | 3 | 3600 |
| " " " Rotating Table | 30 | 18¶ | ...... | 720 | 6–33 | ...... | 25 | 720 |
| " " " Rotating Table | 12 | 8 | 5/8§ | 2000 | 6–310 | ...... | 7 1/2–10 | 1750 |
| Tool | 10 | 5 1/2 | 5/16 | 3770 | ...... | ...... | 1/2 | 3600 |
| Universal | 12–16 | 14 | 1 1/2 | ......... | 60–240 | ...... | 5 | 1750 |

* Diam. of cylindrical work; diam. of holes, internal grinding; width of surface, flat work. † Each motor, 2-wheel machine. ‡ Sectional wheel—dimension given is width of rim or working face. § Width of working face or rim of cup wheel. ¶ Segmental wheel. ‖ Minimum diam. of hole; max. outside diam. of work, 3 in.

**HORSEPOWER REQUIRED FOR GRINDING** will vary directly with the quantity of metal removed. R. E. W. Harrison gives (*Trans.* A.S.M.E., MSP 51–18, 1929) the relation between horsepower and metal removal in centerless or controlled center cylindrical grinders as follows:

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Horsepower at grinding wheel spindle ....... | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 |
| Metal removal capacity, cu. in. per min ..... | $\frac{1}{2}$ | 1 | $1\frac{1}{2}$ | 2 | $2\frac{1}{2}$ | 3 | $3\frac{1}{2}$ | 4 |

Table 18 shows the size of motors fitted to various types of grinders as built by several manufacturers.

**GRINDSTONES** are natural sandstones, in which silicon oxide, $SiO_2$ comprises the abrasive medium. Formerly, widely used for sharpening tools, finishing of cutlery, snagging of castings, etc., they have largely been displaced by vitrified abrasive wheels and by polishing wheels.

Surface speed of grindstones ranges from 500 to 600 ft. per min. for grinding woodworking tools, from 800 to 1000 ft. per min. for metal working tools, and from 1500 to 3500 ft. per min. for polishing operations. The limiting speed is the strength of the stone, which is variable. Wet stones may have only 50% of the strength they possess when dry. Also, stones from the same quarry may vary in strength. In general, 3400 ft. per min. surface speed should not be exceeded, and 2500 ft. per min. is a safer limit. Revolutions per minute corresponding to 3400 and 2500 ft. per min. for various sizes of grindstones are as follows:

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Diam., in ................ | 36 | 42 | 48 | 54 | 60 | 66 | 72 | 78 | 84 | 90 | 96 |
| R.p.m. (2500 ft. per min.) .. | 265 | 227 | 199 | 177 | 159 | 145 | 133 | 123 | 114 | 106 | 99 |
| R.p.m. (3400 " " " ) .. | 360 | 309 | 270 | 240 | 216 | 197 | 180 | 166 | 154 | 144 | 135 |

Grindstones should not be mounted on square shafts as has been common practice. The wooden wedges driven between the shaft and the stone, will, if driven too tightly, or when swollen when wet, tend to start a crack in the corners of the square hole in the stone. This crack gradually extends until the stone ruptures. After the stone has been centered properly on a round shaft, the space between it and the shaft should be filled with lead or cement. Large flanges should be used to support the stone, separated from it by pads of leather, rubber or paper to compensate for irregularity in the mating surfaces.

When grindstones are used to produce smooth surfaces in manufacturing operations, they frequently are hacked around the circumference in a herringbone pattern with cuts about $1\frac{1}{4}$ in. apart to make the stone cut faster. The operation is repeated whenever the hacked surfaces are worn smooth. Grindstones never should be allowed to stand in water when not in use. They will absorb water which will decrease the strength, and also throw the stone out of balance.

## 6. LAPPING AND HONING

**LAPPING** is used to produce exceedingly smooth and highly accurate surfaces, either flat or cylindrical. A lap, usually of cast iron, charged with fine abrasive, is rubbed against the surface of the work either by hand or by machine. Since about 1930 a lap consisting of fine abrasives bonded together has been used in many instances to replace the metal lap charged with loose abrasive. Laps are used with kerosene and lard oil lubricants.

Formerly limited in use to the production of precision surfaces in gages and measuring instruments, lapping has since 1930 been generally adopted as a production process in many industries, particularly the automotive. In the latter, it is used for the finishing of crank-pins, bearings, piston rings, ball-bearing races, valve tappets and other wearing parts. Lapping tolerances of 0.000 05 in. in diameter and 0.000 025 in. in straightness and roundness are possible of attainment.

Sydney Player describes (*Trans.* A.S.M.E. MSP 52–14, 1930) the lapping of plug gages 0.875 in. diameter ±0.000 02 in. Two lapping operations were required. The grinding limit was +0.000 15 to +0.0003 in. Rough lapping with M–304 American Optical emery brought the gages to within 0.000 05 in. of finished size. For finish lapping, the laps were hand scraped and worked together with levigated alumina until the surface of the laps were mirror finished. The author of this paper also describes lapping, in lapping machines, of piston-pins to limits of 0.0001 in. for straightness and roundness, the production being 500 pins per hour.

**Lapping Machines** consist essentially of a pair of laps which may be rotated independently of each other and in opposite directions if desired. The work to be lapped, which may be either cylindrical or flat, is carried in work holders, which positions the work between the two laps, and at the same time moves it between the lap faces with a planetary motion in a constantly varying line of travel relative to the lap surfaces. For cylindrical work, the pieces are carried in the work holder at an angle to radial lines. An eccentric motion imparted to the work holder which, in conjunction with the non-radial position of the work, causes all portions of the surface to come in contact with the laps and a true lapped surface is produced. The capacity of lapping machines ranges up to 3 in. in thickness or diameter by 6 in. long.

The centerless lapping machine is similar in its principle of operation and in general construction to the centerless grinding machine. The lapping is done by wheels of finely divided abrasive suitably bonded. The lapping wheel is driven at a speed of 150, 195 or 225 r.p.m. Through-feed work from $^1/_2$ to 3 in. diameter and of any length can be lapped in the centerless machine. By means of special equipment, work which cannot be passed transversely through the machine can be lapped up to 6 in. diameter and 15 in. long.

**Manufacturers.**—Lapping machines are built by Cincinnati Grinders, Incorporated, Cincinnati, Ohio; Norton Co., Worcester, Mass.

**HONING** is a term applied to the lapping of the interior of cylindrical bores. It is used for the accurate finishing of automotive cylinders, compressor cylinders, cylinders in hydraulic and refrigeration units, bushings, bearings, connecting rods, etc. It consists essentially in the passage of a honing stone or stones in an axial direction through the bore, the hone being rotated at the same time. Tolerances of 0.0005 to 0.0002 in. are commercially possible. J. W. Hindes and J. G. Young (*Trans.* A.S.M.E. MSP 53–13, 1931) state that a ratio of about 1 to 3 of reciprocation to rotation speed gives the most satisfactory results (65–70 surface ft. per min. reciprocation, 200 ft. per min. rotation). If the hone is hard and tends to glaze, this condition can be corrected by increasing the ratio of reciprocating to rotating speed.

The hone usually is a form of expanding segmental wheel, whose surface is formed by a series of abrasive stones carried in a holder. Means are provided to expand the stones to the wall of the cylinder being ground, and to hold them against it under pressure, while the hone is reciprocated and rotated. Wear of the stones is compensated by floating cones in the holders which feed the stones outward and maintain them in parallelism.

J. A. Carlin (*Auto. Ind.*, Dec. 9, 1933) describes a method of honing blind end bores, by shifting the center of pressure on the abrasive stones toward the blind end to compensate for the loss of abrasive effect resulting from inability of the hone to overrun the end of the cylinder. J. E. Kline (*Iron Age*, April 5, 1934) illustrates a fixture used for precision finishing by honing connecting rod crank-pin bores.

**Lubricant.**—Kerosene is generally used as a coolant and lubricant in honing cast iron and other metals. Turpentine gives a freer cutting action on bronze and brass. Water is unsatisfactory.

**Manufacturers.**—Honing machines are made by Barnes Drill Co., Rockford, Ill., Hutto Engineering Co., Detroit, Micromatic Hone Co., Detroit.

## 7. POLISHING AND BUFFING

### Polishing

Polishing is the removal of metal by means of a flexible wheel coated with abrasive. The wheels are made of various materials as described below. The abrasive may be any of the many forms of manufactured abrasives or natural emery. It is held to the face of the wheel by means of an adhesive, usually glue.

**POLISHING WHEELS.**—The earliest form of polishing wheel was a wood disc with a strap of leather fastened to its circumference. This form of wheel has practically disappeared. Polishing wheels in common use are made of buff sections (see below) sewed or glued together to make a wheel of the desired face, of discs of canvas sewed or glued together, or of discs of leather, sheepskin, walrus hide, or similar material. The compress wheel is made of blocks of canvas, leather, sheepskin, paper and other materials, compressed to the required density. The field of application and the characteristic of the several types of wheel are described below.

**Clothflex Wheels** is a general term applied to polishing wheels built of buff sections. The density, *i.e.*, resiliency, of the wheel is determined by the sewing of the buff sections and by the extent of the area of the buff that is glued. Buff sections in which the rows of concentric or spiral sewing are spaced closely, say $^1/_4$ in. apart, and which are glued to the circumference of the buff form the densest clothflex wheel. Wheels of lesser density can be made of sections with wider spaced sewing, say from $^3/_8$ up to $^3/_4$ in., the glue being stopped between 1 to 4 in. from the periphery of the wheel. A wheel that is not glued to its periphery, however, has a tendency to separate at the joints between the buff sections and to leave streaks on the surface of the work. Variation in density is best obtained by wider spacing of the sewing of the buff sections. The least dense type of clothflex wheel is obtained by omitting the glue entirely and holding the buff sections together by several rows of hand sewing.

Clothflex wheels are made both of full disc buffs and sewed piece buffs. Both types are made with cloth of different weaves and each type and weave has its special field of service. Manufacturers of polishing wheels should be consulted on the selection of the grade and construction of wheel that is best adapted to a particular class of work.

Polishing wheels made of sewed piece buffs are used on plumbing supplies, such as faucets, tubular plumbing goods, and on brass, iron, and copper castings with contoured surfaces, and on steel stampings. They may be used on work where the corners are rounded and where it is not necessary to maintain sharp edges. They are not adapted to work where the edges of openings must be held square or where corners must not be rounded over.

The full-disc polishing wheel, *i.e.* made of full disc buff sections, is adapted to use on automatic machines and to work where perfect wheel balance must be maintained. Any desired density of full disc clothflex wheel may be obtained by building it of packed buff sections, that is, sections with discs of smaller diameter in between the buff discs.

**Disc Canvas Wheels** are made of discs of canvas sewed or glued together either singly or in sections of two or three discs. Density can be varied in the same manner as with clothflex wheels by variations in sewing and gluing. The disc canvas wheel is adapted to heavier and rougher work than the clothflex wheel, and is widely used for polishing automobile bumpers, mould boards of plows, harrow discs, etc. It is especially adapted to breaking through scale and to removing deep pits or irregularities in the surface of the work. The disc canvas wheel often is used for roughing with the coarser abrasive grains, finishing being done by a clothflex or a compress wheel coated with finer grain.

**Compress Wheels.**—The process of making a compress wheel consists in assembling blocks of the wheel material into a ring about $1\ ^1/_2$ times the final diameter of the wheel. The assembled ring then is forced through a hollow cone about 3 ft. long, which compresses it radially and circumferentially to the required diameter and discharges it into a cast-iron ring which retains it until it is assembled into the completed wheel. Grooves are cut in the sides of the ring of compressed material into which are dovetailed the edges of flanged steel side plates mounted on a hub. The side plates are riveted together and retain the compressed material in form and place. The assembled wheel then is removed from the cast-iron retaining ring and turned to final diameter in a lathe.

Canvas is the material most widely used in compress wheels. Other materials are leather, sheepskin, felt, walrus hide and paper. Each material has its specific use and produces a different character of surface. Wheel manufacturers should be consulted concerning special applications, and also in regard to the density and cushion to be used for various classes of work.

The density of a compress wheel can be varied over a wide range by varying the number of blocks of material that are assembled into the wheel. The cushion, *i.e.* the distance that the material projects above the side plates, also can be varied within certain limits. Two-inch cushion is the most commonly used, but 3-in. cushion is used where a particularly yielding wheel is desired, and 1-in. cushion where the wheel is required to be rigid.

In the compress canvas wheel, seven densities are regularly manufactured ranging from super-soft to super-hard. The softer wheels are used for contour work or work where the surface may vary more or less from established dimensions and a yielding wheel is necessary to adapt itself to the surface. In general, the softer densities are used in the cutlery trade, on double-header machines, for finishing large knives and on other work where the wheel must flex to cover the surface. Soft wheels should not be used under heavy pressure, as the resulting flexure of the fabric will generate sufficient heat to char the fabric.

The medium density compress canvas wheel is a general all-around wheel largely used as a finishing wheel on such work as automobile bumper bars, sewing machine parts, hardware, automobile accessories and similar work. It frequently is made with formed faces to fit contoured work. The harder densities of wheels are widely used in the cutlery trade for glazing cutlery and table knives. The super-hard density is used for the honing of razor blades. The selection of the proper density is highly important, a difference of even one-half grade in density often being the cause of success or failure of a wheel on a given job.

The compress leather wheel is used where edges and corners are to be held square and where holes are present in the work. These wheels are largely used to polish flat-iron sole plates, both for roughing and finishing, excepting the final operation which is performed with a compress canvas wheel. Compress leather wheels also are used in the cutlery trade for the finishing of knife blades.

Compress walrus and compress sheepskin wheels are used in the cutlery trade, the latter wheel giving a fine surface and good color.

**PREPARATION OF POLISHING WHEELS.**—Various adhesives have been used to hold the abrasive grain on polishing wheels. High-grade hide glue, however, has proven the most satisfactory, as it is the only adhesive that seems to possess all the characteristics required in a polishing wheel adhesive, *i.e.* strength, toughness, flexibility and heat resistance.

The wheel, if new, should first be sized with glue, as should old wheels if the abrasive head has been completely removed. The sizing solution is one part glue to 12 parts water by weight. It should be thoroughly brushed into the surface of the wheel and allowed to dry for a period of 24 hours. Narrow-faced disc wheels should be dried between flat weighted plates to avoid warping.

The abrasive head is applied by thoroughly coating the face of the wheel with a solution of glue of the proper density and immediately rolling the wheel with a firm downward pressure in a trough filled with abrasive. The length of the trough should be somewhat longer than the circumference of the largest wheel to be coated. The practice of using short troughs and bumping the wheel into the abrasive while turning it on an arbor cannot be recommended. The head thus produced is uneven and tends to throw the wheel out of balance. A second coat of abrasive should be applied after an interval of from 20 to 30 minutes. The wheels should be at a temperature of from 75° to 90° F. when the glue is applied. After coating with abrasive they should be dried for not less than 48 hours at a temperature not to exceed 75° F. Shorter drying times or higher temperatures will shorten the life of the wheel head and increase the cost by the useless expenditure of abrasive and the increased labor for setting up. Where a large number of polishing wheels are used daily, air conditioning of the drying room will be profitable. The best conditions for drying are a temperature range of 70° to 75° F. and relative humidity of 50 to 55%.

**Preparation of the Glue.**—Correct handling of the glue is the most important factor in economical polishing. Density of the glue, *i.e.* proportion of water to dry glue should be varied in accordance with the size of the grain to be applied. Table 19 shows proportions of water and glue found to be best for the various sizes of abrasive grain. The glue should be soaked in cold water from three to four hours, and after all the water is absorbed the glue should be melted at a temperature of 150 to 160° F. It then will be at about its temperature of maximum strength (134° F.) when the wheel is rolled in the abrasive. Glue temperature should not exceed 160° F. since it rapidly loses strength at elevated temperatures. Glue deteriorates in strength also by standing under heat. Therefore, no more glue should be mixed and melted at one time than will suffice for about 4 hours work. It is advisable to melt glue in thermostatically-controlled electrically-heated glue pots, which will maintain the temperature within ± 2° F. As glue is highly susceptible to deterioration from organic matter, glue pots should be thoroughly cleaned after use, and the use of remelted glue absolutely prohibited. Glue brushes should be soaked in a 5% solution of carbolic acid when not in use. Only distilled water should be used in the preparation of glue.

**WHEEL BALANCE.**—All polishing wheels should be carefully balanced before use. Unbalanced wheels produce poor work, have short life of wheel head and increase the cost of production both by reason of the extra effort required to produce good surfaces and the lower production resulting from increased fatigue of the operator on hand work. On automatic machines good work is almost impossible with unbalanced wheels. Balancing is performed by mounting the wheels on an arbor and placing them on balancing ways. Clothflex and disc canvas wheels are balanced by tacking pieces of lead on the side of the wheel. Compress wheels are balanced by inserting lead weights in balancing tubes which are built into the wheel. This type of wheel is susceptible to much more accurate balance than disc wheels. For high precision work, dynamic balance of wheels is advisable, and in some cases imperative.

**APPLICATIONS OF POLISHING WHEELS.**—It is impossible to lay down rules covering all cases of the application of polishing wheels. The wide range of variables

**Table 19.—Proportions of Glue and Water for Setting Up Polishing Wheels**

| Size of abrasive grain | 36 | 46 | 60 | 90 | 120 | 150 |
|---|---|---|---|---|---|---|
| Percentage of dry glue | 45 | 40 | 35 | 33 | 30 | 25 |
| Lb. of water per lb. of glue | 1.22 | 1.50 | 1.86 | 2.30 | 2.50 | 3.00 |
| Qt. of water per lb. of glue | 0.59 | 0.72 | 0.90 | 0.98 | 1.00 | 1.40 |

encountered both in hand lathes and automatic machines, the wide variety of materials polished, and the constantly varying character of the work surface, set up conditions that are peculiar to the individual job. Wheel speeds that are correct for one set of conditions would be highly inefficient under another set of conditions. Wheel pressure that is necessary for one class of work would ruin another class. Work speed in relation to wheel speed must be varied in accordance with the character of the material, the shape of the work and the degree of finish desired, particularly with automatic machines. Table 20 gives data of typical polishing jobs that will serve as a guide in the application of polishing wheels.

**Table 20.—Applications of Polishing Wheels**

| Metal | Part | Operation | Wheel | | | | | See Note No. |
|---|---|---|---|---|---|---|---|---|
| | | | Roughing or Finishing | Type * | Density | Cushion, in. | Surface speed, ft. per min.† | |
| Aluminum.... | Radio horn....... | Outer convex surface...... | R,F | CC | Soft | 2 | 7500 | 1 |
| Brass (cast)... | Bracket.......... | Flat Surface............. | R,F | CC | Soft | 2 | 7500 | |
| “ “ ... | “ .......... | Inside rounded corner..... | R,F | CF | 3/8″ sewed | ...... | 9000 | 2 |
| “ “ ... | Faucet........... | External body surface..... | R,F | CF | 3/8″ “ | ...... | 8500 | 3 |
| Cast iron..... | Flat iron......... | Sole plate face........... | R | CL | Med. hard | 2 | 7500 | 4 |
| “ “ ..... | “ “ ......... | “ “ “ .......... | F | CC | “ “ | 2 | 7500 | 5 |
| “ “ ..... | Cement corner tool | Inside corner............. | R,F | CL | Medium | 2 | 7500 | 4 |
| “ “ ..... | Machine handle... | External surface.......... | R,F | CC | Soft | 2 | 7500 | 6 |
| “ “ ..... | Base standard..... | Flange and rounded corner | R,F | CF | 3/8″ sewed | ...... | 8500 | 3 |
| Silver........ | Cup.............. | External surface.......... | F | CL | Soft | 1 1/2 | 1200 | 7 |
| Stainless steel. | Table knife....... | Blade flat................ | F | CC | Superhard | 2 | 5700 | 8 |
| Steel......... | Auto bow-socket.. | External surface.......... | R,F | CC | Supersoft | 2 | 7500 | 9 |
| “ ......... | Pocket knife...... | Bolster................... | R,F | CL | Medium | 2 | 7500 | 4 |
| “ ......... | “ “ ...... | Blades.................... | F | CC | Soft | 3 | 5700 | 10 |
| “ ......... | Auto bumper..... | Convex face............... | R | DC | 2 ply | ...... | 7500 | 11 |
| “ ......... | “ “ ..... | “ “ ............. | F | CC | Medium | 2 | 7500 | 6 |
| “ ......... | Safe door handle... | Ball...................... | R,F | CC | Soft | 2 | 7500 | 6 |
| White metal.. | Teaspoon......... | Handle edge............... | F | CP | Hard | 2 | 7000 | 12 |

* CC = Compress canvas. CF = Clothflex. CL = Compress Leather. CP = Compress paper. DC = Disc canvas.

† See paragraph Polishing Wheel Speeds.

Note 1.—Compress canvas is used to avoid streaking the work. The soft density permits the work to embed itself in the wheel, which thus covers the irregular contour.

Note 2.—A clothflex wheel is used because the shape of the work requires sidewise flexibility in the wheel. The face of the wheel is formed to the radius of the corner.

Note 3.—The face of the wheel is formed to fit the outline of the faucet. The clothflex wheel is used on account of its flexibility and relative cheapness for a job whose volume is not sufficient to warrant the increased expense of a compress wheel.

Note 4.—Compress leather or hard compress canvas is used where a square edge must be preserved, or where sharp corners are to be polished, or a truly plane surface is to be maintained. Such wheels are rigid and yet resilient.

Note 5.—The compress canvas wheel will produce a color on the work that cannot be obtained by the compress leather wheel.

Note 6.—Compress canvas can be formed to the contour of the work and will hold its shape indefinitely. The soft density enables the wheel to cover the entire surface to be polished.

Note 7.—This wheel operates without abrasive glued to the face. Loose abrasive, slightly moistened with oil, is thrown between the wheel and the work. The wheel is formed to the contour of the cup. The operation is known as sand buffing, and compress leather wheels are always used.

Note 8.—A superhard wheel is required in order to hold the surface uniformly flat and to avoid the formation of waves and ridges.

Note 9.—A supersoft wheel will crush sufficiently to conform to the slight irregularities in contour of the work. This avoids using a second formed wheel to finish the irregular surface.

Note 10.—This operation is performed in a double-header machine. The wheels must be soft enough and flexible enough to cover the entire blade surface from heel to edge. The soft density and deep cushion fulfill these requirements.

Note 11.—A disc canvas wheel is best suited to rough work on an irregular surface.

Note 12.—An extremely hard wheel is necessary to maintain contour on a narrow edge and to prevent rounding of the edges.

**POLISHING WHEEL SPEEDS.**—A surface speed of 7500 ft. per min. has commonly been used for polishing wheels working on steel, iron or brass, and due to long usage has been regarded as the most efficient speed. The figures for surface speed given in Table 20 are the speeds that actually were used on commercial jobs. Some recent research, however, has demonstrated that the commonly accepted surface speed of 7500 ft. per min, is not necessarily the most efficient speed. A speed of 6700 ft. per min., polishing medium steel showed an improvement of about 15% from the standpoint both of metal removal and time consumed. Similar results were obtained with a speed of about 7000 ft. when polishing cast iron. Investigation has not (1937) been sufficiently detailed to determine whether or not these two speeds are the most efficient. It, therefore, is advisable for manufacturers polishing large quantities of metal to conduct experiments to determine the most efficient speeds for their particular conditions.

**ABRASIVES.**—The type and size of abrasive used on polishing wheels depends on the material polished and the finish desired. Turkish emery formerly was used almost exclusively. It has been displaced by the manufactured abrasives, except as a finishing abrasive where high color is desired. For steel and brass the aluminum oxide abrasives give the highest cutting efficiency and longest wheel life. For cast iron, a silicon carbide abrasive should be used.

The size of abrasive grain to be used depends on the initial character of the surface and the finish to be produced. The first wheel should be coated with abrasive that will cut down to the bottom of the deepest pits and break through scale. The second wheel should be coated with a finer abrasive that will cut out the grain scratches made by the coarser wheel. The next wheel should have still finer grain, and so on until the desired surface has been produced. The final wheels should be greased with oil or tallow to modify the cut, and to produce a more lustrous surface.

A common sequence of grain sizes for the hand lathe polishing steel is: First wheel, No. 60 or 90; second wheel, No. 90 or 120; third wheel, No. 120 or 150. If deep pits are present in the work, it may be necessary to use a coarser abrasive on the first wheel, say No. 46. It should be remembered, however, that each wheel must remove the grain scratches of the preceding wheel and it may be more economical to use a finer wheel for a longer time to remove pits, than to spend time removing coarse grain scratches. Highly polished steel is finished with a sequence of grain sizes of Nos. 60, 90, 120, 180, and 220. All the wheels finer than No. 90 should be oiled or greased. The sequence of grains may be extended even farther to produce mirror-like, scratchless surfaces. Rolls for copper and brass rolling mills have been finished with a final wheel coated with levigated alumina, which is about equivalent to a grain size of No. 640.

**DESCALING.**—The scale on castings or forgings imposes a heavy burden on the initial polishing wheels, which must break down this scale. Some scales are so hard as to absolutely resist the action of the wheel. Such scales should be removed by pickling or by one of the electrochemical processes of descaling. For continuous production, descaling will soon pay for itself in increased wheel life and decreased number of wheels necessary to produce the desired surface.

**Pickling Solutions.**—Castings may be pickled in a solution of sulphuric acid, diluted with water to a specific gravity of 25 deg. Baumé. Forgings should be pickled in a solution of sulphuric acid and water of a specific gravity of 30 deg. Baumé, for from 3 to 12 hours. After pickling, forgings and castings should be washed with hot water to remove the sand and acid. A disadvantage of acid pickling is the tendency to etch pits still deeper, thus adding to the work of the polishing wheels.

**Bullard-Dunn Process.**—This process, patented by the Bullard Co., Bridgeport, Conn., is a simple, rapid and inexpensive electrochemical method of removing scale and oxides from the surface of metals. It is particularly adapted to descaling iron and steel, including the stainless steels. The characteristic feature is a continuous, non-porous metal film electro-deposited on the work simultaneously with the scale and oxide removal. The film prevents damage to the work by protecting it from surface pitting and etching. While it can be removed easily and completely, it is usually advantageous to permit it to remain on the work. The equipment used is an ordinary electroplating tank, using direct current of 6 volts or less. The current density is about 60 amperes per square foot of exposed surface. Anodes of whatever metal is desired for the film are used, lead and tin being the most common. The film deposited is on the order of 0.000 001 in. thick. For large output, automatic conveyors are used in the tanks; manual operation of tanks is satisfactory for smaller outputs. In operation, the film is deposited instantaneously on the removal of the scale, and inhibits further action. No allowance, therefore, need be made in basic dimensions to compensate for dimensional losses. Scale is completely removed from bottoms of threads, from cracks, crevices and internal surfaces. The removal of scale is accomplished rapidly.

## Buffing

Buffing consists in leveling off minute projections on the surface of metal, and in producing a sheen or luster on the surface so prepared. The first operation is termed *cutting down* and the second *coloring*. The work is performed with a buffing wheel which is composed of a number of circular discs of cloth mounted on an arbor and rotated at high speed. Buffs or buffing wheels are made in several forms each of which has its particular field of application.

**BUFFING WHEELS.**—Buffing wheels are made by assembling on an arbor a sufficient number of *buff sections* to form a wheel of the requisite width of face. The various forms of buff that are used are: 1. Full disc buffs, which may be either *loose* or *sewed*. 2. Sewed piece buffs, which always are sewed. 3. Some form of patented folded buff.

Buff material usually, but not always, is cotton sheeting, bleached or unbleached. The most widely used grade of sheeting has a warp of 64 threads per inch and a filling of 68 threads. It commonly is known as 64 × 68 cloth. Another widely used fabric is woven 84 × 92. Buffs made of this fabric are harder and will stand rougher usage than those made of 64 × 68 fabric. They generally are used for the cutting down operation, the 64 × 68 buff generally being used for coloring. Buffs made of bleached material are harder than those of unbleached fabric.

**Full Disc Buffs** are made of complete discs of the fabric and are assembled in *sections*, usually of 20 plies of fabric. The loose buff is sewed with three or four closely spaced rows of sewing around the hole to hold the discs together. The sewed buff in addition to the sewing around the center hole is sewed with concentric or spiral rows of sewing, spaced at any desired interval, from the center hole to the periphery. The wider the spacing of this sewing the more flexible and yielding is the buff. Other forms of sewing sometimes are employed, as radial sewing in which the sewing extends on radial lines from the center to the periphery; tangent sewing in which the sewing is tangent to a circle of about 3-in. diameter, and extends to the periphery of the buff; various forms of geometric patterns. While these special buffs have some fields of application, the concentric or spirally sewed buff generally is satisfactory for most work.

The 64 × 68 loose buff is used for coloring soft nickel, brass, and cast iron. When sewed it may be used for cutting down copper, soft brass, zinc and similar metals on hand lathes. The 84 × 92 buff both in sewed and loose form is used to cut down copper, brass, German silver and heavy nickel plate on bumper bars, builders' hardware, etc. For chromium plate, it should be sewed.

**Sewed Piece Buffs.**—The sewed piece buff is built up of more or less irregularly shaped pieces held between two full discs of fabric and sewed from the center hole to the periphery with spiral sewing. The same grades of fabric are used in sewed piece buffs of the better grades as in the full disc buffs. Many of the cheaper buffs of this character contain much coarser and flimsier fabric. The better grades of sewed piece buffs are built of relatively large pieces whose area ranges from $^1/_2$ to $^1/_4$ the area of the full disc. The cheaper grades contain smaller pieces which may range down to 5% of the area of the disc. The buffs with the large pieces are more durable and are better balanced than those built of small pieces, and from an operating cost standpoint, are much more economical than the others. It is impossible to determine the size of the pieces used in building a sewed piece buff without destroying the buff, and buff users should not be misled by low prices. Where sewed piece buffs are purchased in large quantities, it is wise to procure them only from manufacturers who have established reputations for quality. The sewed piece buff costs less than the full disc, and for many buffing operations is equally satisfactory.

Sewed piece buffs are largely used for cutting down cast plumbing goods, brass stampings, brass tubular goods, and aluminum castings.

**Folded Buffs.**—Folded buffs consist of full discs folded and assembled in various ways to form pockets which then are assembled to form a complete circle and are sewed either as a loose buff or as a spirally sewed buff. These buffs are used both for cutting down and for coloring. Their action is somewhat different from that of the loose or the sewed piece buff, in that the edge of the fabric passes across the work at an angle to the plane of the buff wheel. Buff marks on the work which are produced by each separate layer of fabric in the other types of buff thus are avoided. Another advantage of the folded buff is that the pockets tend to trap air, which has a cooling effect on the buff surface. Folded buffs are relatively more expensive than any other type, because of the large amount of hand labor required in their manufacture.

**BUFF VENTILATION.**—Buffs continuously operating under heavy pressure must be ventilated in order to dissipate the heat generated. Without adequate ventilation the buff may char or even be set afire. Folded buffs have provision for ventilation inherent in their construction. Full disc buffs may be ventilated by the process known as *packing*. This consists in assembling one or two discs of smaller diameter than the buff disc between each pair of buff discs, or at such intervals as may be necessary, depending upon the severity of the work. Sewed piece buffs can be ventilated by inserting a few discs of fabric or washers of felt or leather between each section. Packed buffs should be oscillated axially in relation to the work in order to avoid distinct streaks at the points where the discs or sections are separated by the packing.

**SPEED OF BUFFS.**—The surface speed at which a buff should operate depends on the type of metal being buffed. Table 21 shows the speeds commonly used with various metals and various grades of buffs. In general, higher speeds are used for cutting down than for coloring with the copper bearing metals.

Since a buff wears down in use, its peripheral speed constantly decreases. Provision should be made to maintain the peripheral speed fairly constant either by the provision of a variable speed drive or by changing the buff to a machine of higher spindle speed for each decrease of 2 in. diameter.

**PRESSURE AND HORSEPOWER FOR BUFFING.**—The effectiveness of buffing depends largely on the pressure of the wheel on the work. Increasing the pressure increases the rapidity of cutting, but at the same time increases the liability of cutting through the plating on plated articles. Definite rules cannot be given since the pressure that safely can be applied depends on the hardness of the plating and its thickness. A heavy coat of hard nickel can withstand much greater pressure than a thin coat of soft nickel. The horsepower required for buffing is a function of the pressure of the wheel on the work. In general, for buffing nickel plate on steel on hand lathes 3 to 5 Hp. at the buff is the usual practice. For heavy work, such as automobile bumper bars, hand lathes usually are equipped with 10 to 15 Hp. motors.

Brass requires considerably more horsepower for buffing than does steel or nickel. Experiments made by the author indicated that a pressure of 15 lb. per inch of width is required for cutting down sheet brass, and 10 lb. per inch of width is required for coloring. Table 22 shows the power required for cutting down and coloring sheet brass. The figures are based on finishing the brass at a work speed of 18 lineal ft. per min. in an automatic machine.

**BUFFING COMPOSITIONS.**—The following compositions are adapted to the different varieties of work: For cutting down brass, bronze and Britannia metal preparatory to plating, tripoli; for smooth surfaces on nickel and brass, crocus; for coloring brass, copper, nickel, bronze, German silver, etc., either in solid or plated metal, White Diamond XXX composition; for chased or embossed parts, or for cutting down silver-plated pieces which are afterward to be colored with rouge and alcohol, White Diamond XXXX composition is used; for nickel-plated pieces with a high luster, Vienna lime. Where rapid, sharp, even cutting is desired, emery cake is used. Chandelier rouge is used to produce a deep color on brass and bronze parts. Chromium oxide is used for chromium plated articles and stainless steel.

## Polishing and Buffing Machinery

**Hand Lathes.**—For irregularly shaped pieces, or where work is performed in relatively small lots, the hand lathe is almost universally used both for polishing and buffing. These

**Table 21.—Buffing Practice for Various Metals**

| Material | Cutting Down | | | | | Coloring | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Buff | | | Composition | | Buff | | | Composition | |
| | Weave | Sewing | Surface Speed, ft. per min. | Kind * | Grade | Weave | Sewing | Surface Speed, ft. per min. | Kind * | Grade |
| Aluminum | 84×92 | 0 | 8500 | T | X67 | 64×68 | 0 | 7500 | WD | XXX |
| Brass (cast) | 84×92 | 0 | 8000 | T | X67 | 64×68 | 0 | 7500 | WD | XXX |
| | Pieced | 3/8 | 8000 | T | X67 | | | | | |
| Bronze (cast) | 84×92 | 0 | 8000 | T | X67 | 64×68 | 0 | 7500 | WD | XXX |
| Copper | 84×92 | 0 | 9000 | T | X67 | 64×68 | 0 | 7500 | T | XX |
| Chromium | 84×92 | 0 | 8000 | CO | ...... | ...... | ...... | ...... | ...... | ...... |
| Monel Metal | 84×92 | 0 | 7500 | C | ...... | 64×68 | 0 | 7500 | L | ...... |
| Nickel | 64×68 | 0 | 7500 | L | ...... | 64×68 | 0 | 7500 | L | ...... |
| Stainless Steel | 84×92 | 0 | 8500 | CO | ..... | 84×92 | 0 | 8500 | CO | ...... |
| | Pieced | 3/8 | 8500 | CO | ..... | | | | | |
| White Metal (Die cast) | Pieced | 3/8 | 6000 | T | X67 | 64×68 | 0 | 6000 | WD | XXX |

* C = Crocus. CO = Chromium oxide. L = Lime. T = Tripoli. WD = White diamond.

**Table 22.—Horsepower of Motors Required for Buffing Sheet Brass**

| Wheel Pressure per Inch of Face Width | Width of Sheet, Inches | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
| | Horsepower | | | | | | | | | | | |
| 10 lb., for Coloring | 3 | 5 | 7 1/2 | 10 | 10 | 15 | 15 | 20 | 20 | 25 | 25 | 30 |
| 15 lb., for Cutting Down | 5 | 7 1/2 | 10 | 15 | 20 | 20 | 25 | 30 | 30 | 40 | 40 | 50 |

are built in a wide variety of sizes and styles ranging from a simple cupola jack to large double-spindle machines driven by independent motors and built with the precision of high-grade machine tools. A common type of motor-driven machine is the single spindle machine with the motor armature on the spindle. The spindle projects at either end to form arbors for the polishing or buffing wheels. Many motor-driven lathes are available with the motor mounted in the base, driving the spindle through flat- or V-belt drives. These machines have the advantage, in common with the plain belt-driven machine, of the possibility of changing spindle speed by changing the driving or driven pulley. The better grades of polishing and buffing lathes are built with ball-bearing spindles.

Points to be considered in the selection of a lathe for polishing and buffing are: Adequate diameter of spindle between bearings and adequate diameter of wheel arbors; adequate overhang of wheels beyond the base to permit the handling of long and irregularly shaped pieces without interference with the operator working at the wheel at the other end of the spindle; correct height of spindle above the floor, since a few inches difference in height will engender fatigue and decrease production; motor of ample size to handle the heaviest work which may come to the lathe.

Lathe arbors commonly have been made 1 in. to 1 $^1/_4$ in. diameter. This is adequate for the usual range of buffing jobs, but is too light for many polishing jobs particularly if wide, heavy compress wheels are used. Arbors of 1 $^1/_2$ in. diameter will prove more satisfactory for the general run of polishing, and 1 $^3/_4$ in. is better for severe duty like the polishing of automobile bumper bars where the wheel is subjected to heavy pressure.

Motors for the general run of light work should not be less than 5 Hp. For medium duty work, motors of 7 $^1/_2$ to 10 Hp. will prove satisfactory, while for the heaviest class of work, 15 to 20 Hp. motors should be used. It is economy to install more, rather than less, horsepower than is actually needed both on polishing and buffing lathes.

**AUTOMATIC AND SEMI-AUTOMATIC MACHINES.**—For long runs of similar work, automatic or semi-automatic polishing and buffing machines are advisable. They are economical of labor, have high capacity and produce uniform work. In one instance, automatic equipment reduced the number employed on a given line of work from 31 to 5 operators, increasing production at the same time. Automatic machines for automobile bumpers have been built with an output of upwards of 300 sets of bumpers per hour. Automatic machines may be divided generally into three classes, straight-line machines, multiple-head machines, and contour machines.

**Straight-line Machines** comprise essentially a conveyor on which the work is carried in appropriate chucks or holders underneath a series of polishing wheels, successively coated with the various sizes of grain necessary to obtain the desired finish. The wheels are arranged to float, either on an arm oscillating vertically on a horizontal shaft, or between vertical guides. Each wheel is driven by an independent motor, direct-connected to the wheel spindle or driving the wheel spindle through V-belts. The conveyor is driven by an independent motor. The more common form of conveyor is a belt, although various other forms embodying roller chain, or other construction, have been built. The conveyor usually is driven through a speed changing device at speeds ranging from 15 to 35 ft. per min.

The use of floating wheels permits the straight-line machine to polish and buff work that is irregular in contour, for example, automobile bumper bars, which are offset, *i.e.* all longitudinal elements not in the same straight line. The wheels strike the lowest part of the bar as it moves under them and float upwards as the bar moves, following the curve of the bar. By the use of wheels of the proper contour the rise of the wheel will bring a different portion of its face into contact with the bar at the offset, thus polishing the entire surface. By a study of wheel forms, wheel pressures and work speeds, many irregularly shaped forms can be finished in straight-line machines. They are widely used not only for automobile bumpers, but for flat-iron sole plates, automobile door hinges, pliers, open-end wrenches, typewriter and adding machine parts, washing and ironing machine parts, electrical appliances, etc.

Motor horsepower ranges from 5 Hp. per spindle for light work such as typewriter and adding-machine parts, to 20 Hp. on automobile bumpers. Conveyor motor horsepower ranges from 2 $^1/_2$ on short machines for light work up to 10 or 15 Hp. for 16- or 20-wheel bumper machines.

**Multiple-head Machines.**—Multiple-head machines are used principally for the finishing of circular work. In their simplest form they consist of a head carrying two spindles on which chucks are mounted to hold the work. The head is indexed by hand to bring the work on each spindle in contact with the polishing or buffing wheel. The spindle in the working position is rotated by a small motor mounted on the machine. The idle spindle is unloaded and reloaded during the operating time of the working spindle. Other forms of machines have heads carrying from 4 to 6 spindles which automatically are indexed into contact with the polishing or buffing wheels at predetermined intervals. Other forms of this type of machine are provided with mechanisms to oscillate the head in a direction parallel to the face of the wheel, the work spindle being rotated at the same time. This enables long work to be finished with a narrow wheel. Machines of the foregoing types usually are built separate from the polishing wheel. In general, they are relatively light machines which easily can be moved into position alongside of a hand lathe and removed when the work is completed or the lathe is needed for other purposes. Larger forms, however, have the polishing or buffing wheels incorporated in the machine.

Other forms of multiple-head machines comprise horizontal tables carrying from 4 to 6, or even more, work chucks. These are indexed successively under a series of polishing or buffing wheels. The number of wheels is one or two less than the number of chucks, thus providing a station for loading and unloading. The polishing or buffing wheels either are integral with the machines or carried on separate stands and driven by individual motors. The wheels usually float, as in the straight-line machine. Machines of this type are essentially high production machines. The use of several polishing wheels, coated with different sizes of abrasive from the coarsest to the finest, enables a piece to be finished complete in one passage through the machine. These machines are used largely for finishing flat-iron sole plates, automobile hub caps, and similar work.

**Contour Machines** for polishing or buffing articles of rectangular outline or forms not truly circular, are essentially a rotating spindle which carries the work, combined with a mechanical movement which causes the work to approach or recede from the wheel in conformity with the contour, and which maintains uniform contact of the work and wheel at all times. The mechanical movement may be a cam of appropriate shape or a geared chuck.

The geared chuck (Fig. 19) consists of a chuck, on the back of which is mounted an internal gear, or continuous rack *A*. The pitch line of the rack is parallel to the outline *B* of the surface to be finished. The outside edge *C* of the rack also is parallel to the outline *B*, and bears against a roller *D* which is mounted on the machine frame. A pinion *E* drives the rack. The centers of the polishing or buffing wheel, roller, and pinion, should be in the same straight line. The rack being held always in the same relative position to the wheel by the roller and pinion, rotation of the pinion will cause every point of the outline *B* to come in contact with the wheel.

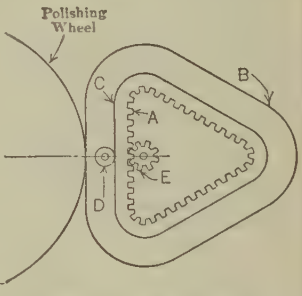


FIG. 19. Geared Chuck

**Special Machines** have been built for many different purposes including the finishing of ice-skate blades, windshield wipers, blank book backs, sewing thimbles, tubing, sheet and strip metal, heads of screws and bolts and various forms of cutlery. Some of these machines employ polishing wheels in pairs, the work passing between the wheels in order to finish two sides at one operation.

**PRINCIPLES FOR POLISHING AND BUFFING MACHINE DESIGN.**—The following principles should be embodied in polishing machines: 1. *Variable Wheel Speed.* Where a machine is to be used with wheels of a variety of diameters, or where the diameter decreases in service, as with a buffing wheel, provision should be made for a variable-speed wheel spindle. Wheel speed then may be maintained constant at the most efficient point. 2. *Variable Work Speed.* Provision should be made to vary the speed at which the work moves past the wheel. The time required for a wheel to do its work varies with different materials, and provision should be made to change the work speed as necessary. Work speeds range from 200 ft. per min. with strip zinc, to 15 ft. per min. with stainless steel. Hardness of material and character of finish desired also influence work speed. 3. *Variable Wheel Pressure.* Excessive pressure of the wheel on the work may damage the wheel and work, change the character of the finish, and increase the cost of polishing or buffing. Hence, all machines should have provision to regulate wheel pressure. 4. *Wheel Oscillation* in an axial direction is desirable to avoid deep scratches and wheel marks. 5. *Vibration* should be eliminated by adequate foundations, sufficient weight in machine bases, shafts of adequate diameter, ample bearings and dynamic balance of rotating parts. Machine vibration will spoil finished surfaces and increase costs. 6. *Work Holding Fixtures* of such design as to properly present work to the wheel, and to facilitate rapid loading and unloading greatly reduce the cost of machine operation.

For many data on the design and operation of polishing and buffing machines, see Robert T. Kent, Fundamentals of Machine Polishing, *Trans.* A.S.M.E., MSP–53–9, 1931.

# BROACHING *

**THE BROACHING PROCESS.**—Broaching is suited for machining holes of any shape, and is an economical method for the duplication of parts in quantity. A large variety of work may be handled without holding fixtures, and intricate shapes are easily held in simple jigs. The accuracy of broaching is comparable to that of milling under ideal conditions.

**BROACH TERMINOLOGY.**—Certain terms have come to apply, through common usage, to various types of broaching tools.

**Keyway Cutter, or Cutterbar** (Fig. 1).—A broach of rectangular section operating in a slotted bushing which adapts it to the bore of the piece in which the keyway is to be cut. The depth of cut may be regulated by a shim in the work bushing slot below the cutterbar.

**Round Broach** (Fig. 2).—A broach for bringing to size a hole which has been drilled, bored or cored round before the broaching operation.

**Multiple Spline Broach** (Fig. 3).—A broach designed to produce simultaneously two or more keyways in one bore. Its common forms are the 6-, 10- and 12-spline broaches used on automobile transmission gears.

**Square Broach** (Fig. 4).—A broach to produce a square hole, starting from a drilled round hole, or a cored square hole.

**Roughing Broach.**—A broach designed to remove a large amount of metal and prepare the hole for the sizing broach, which brings the hole to size and imparts a smooth finish. The roughing broach, not being of such accurate dimensions is cheaper to make and may be re-ground when dull.

**Pull Broach.**—A broach operated by applying power to the shank of the tool and drawing it through the work.

**Push Broach** (Fig. 5).—A broach forced through the work by applying pressure at the finishing end. Presses of various types are used for this work.

**BROACH MATERIAL.**—Broaches are made of carbon steel of 1.00 to 1.10% carbon, high-speed steel, and occasionally of case-hardened machine steel. The latter has a carbon content of about 0.25%. Broaches made from it are case-hardened in a tube packed with charred granulated bone, by heating to 1700° F., cooling slowly, reheating to 1450° F., and then quenching. They are drawn to a temperature of 375° F. Carbon-vanadium steel also is used. It is hardened at from 1350 to 1425° F., and drawn to a temperature of about 460° F.

**MATERIALS THAT MAY BE BROACHED.**—Cast iron, brass, aluminum and steel are readily broached, and many parts made of fiber have been handled satisfactorily.

Soft " stringy " steels often tear out at the edge of the hole. This can be eliminated

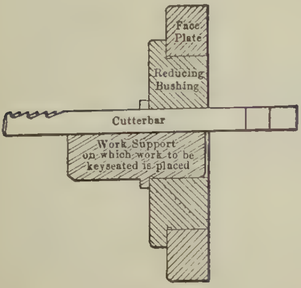


FIG. 1

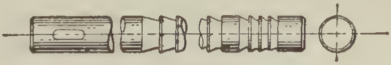


FIG. 2

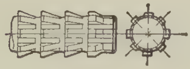


FIG. 3

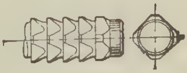


FIG. 4

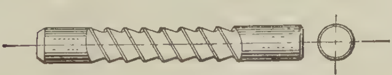


FIG. 5

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* Originally contributed by Louis E. Peck. Revised by Robert T. Kent.

and the finish improved by heat treating to give a scleroscope hardness of from 25 to 30. Harder material causes excessive wear on the broach.

**PREPARING WORK FOR BROACHING.**—The hole in which the broach is to start should be drilled or bored. It is possible to broach from a cored hole, but the sand and scale soon impair the accuracy of the broach. The face of the work that rests against the face-plate of the broaching machine should be finished true with the bore.

**BROACH DESIGN.**—**Pitch of Teeth.**—For best results three teeth usually should be in the work at one time, although this varies with the conditions. A formula which has been published and which may be used for determining approximately the proper tooth pitch for broaches is as follows:

$$P = \sqrt{L} \times 0.35,$$

where $P$ = pitch of tooth, and $L$ = length of work.

**Depth of cut** for each tooth depends largely upon the pull that the smallest section of the broach will stand. In general, the minimum cut should be 0.00075 in. and rarely should it exceed 0.004 in. per tooth. The rise of the successive teeth depends on the material being broached, its hardness and toughness, and the proportions of the broached surfaces. For cast iron or brass, it may be double the rise for steel. Tooth rise usually is greater for the first few teeth on the broach, the rise diminishing toward the finishing end. The last four teeth on the broach or the last 2 in. are made to full cutting size so that the cutting teeth may be sharpened several times without reducing the size of the broached hole.

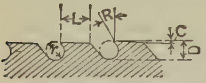

FIG. 6

Other tooth dimensions are as follows (see Fig. 6): $D$, the total depth of tooth depends on the diameter of the broach. It must be calculated for every tool, to provide as much chip room as possible without weakening the tool beyond a reasonable factor of safety; $r$, the radius at the bottom of the tooth should be slightly more than half the total depth of the tooth; $P$ should be slightly more than $^1/_2$ $D$; $L$ = $^1/_{16}$ in. to $^1/_8$ in.; $C$ = 1 $^1/_2$ deg. to 3 deg.; $R$ = 8 deg. to 10 deg.

Push broaches generally are made from 6 to 15 in. long, depending on the character of the work. Several may set in sequence in order to produce the desired finish and accuracy. Pull broaches may be made of any length necessary, and usually are designed to finish the hole at one pass of the broach.

**Chip Breaking.**—In designing round and square broaches, and other types having wide cutting surfaces, chip breakers must be provided. On round broaches a staggered nick should be used. In broaches of small diameter a large number of nicks will be required in each tooth.

**Information Necessary for Design of Broach.**—1. Kind of material broach is to cut, which limits the cut per tooth and the clearance angle at top of tooth. Rake and clearance angles should be identical with those of a lathe tool cutting the same metal. 2. Length of work, *i.e.*, length of surface actually to be broached, which determines the pitch of teeth. This dimension represents $L$ in the formula previously given. 3. Method of preparing the work for broaching, which determines the shank diameter of the broach. If the hole is cored, the first cutting teeth should be made very strong and given sufficient cut to get beneath the scale. Unless two or more broaches are to be used it is not advisable to start the broach in a cored hole as the hard wear on the tool quickly impairs the accuracy. 4. Tolerances. The required limits of the hole to be broached, when finished, should be carefully considered. It is always best to make the broach to the high limit so as to allow maximum wear before the broach is worn undersize. Where work will permit, liberal tolerances should be allowed in the broach dimensions, as this will make the tool less expensive.

**BROACH SPEEDS AND FEEDS.**—Cutting speeds on hydraulic broaching machines range from 10 to 42 ft. per min. Return speeds are from two to four times greater.

**TESTING OF BROACH.**—The broach should be tested in the material in which it is to be used before starting work on a production basis. If there is a tendency to "drift" or cut oversize on any side, the side from which the broach drifts should be examined for dull teeth, and these sharpened with an oil stone.

**CAPACITY OF BROACH.**—A broach should be used only in work for which it was designed. If the piece to be broached is longer than that for which the broach was designed, there will not be sufficient chip clearance. The teeth will fill with chips and roughen the surface of the work. If the strain becomes too great, the broach will break. If the work is much shorter than the intended capacity of the broach, there will not be enough teeth in the work to prevent crawling and cutting oversize. The broach then should be changed or redesigned in accordance with the formula given above.

**LUBRICATION.**—The cutting lubricant is important in broaching operations. There is a wide difference of opinion as to the best lubricant for various metals. Those mentioned below have been used successfully.

**For Cast Iron.**—Soda and water will improve the finish of broached cast iron, will wash chips from the broach teeth, and prevent clogging.

**For Steel.**—A soluble oil made as follows is popular and inexpensive: 2 1/4 lb. soda ash, 3 gal. mineral lard oil, 50 gal. water. Mix the soda and lard oil together in 10 gal. of water, then add 40 gal. of water. Kerosene is frequently added to this mixture if the steel is very tough.

**For Aluminum.**—In broaching aluminum, turpentine or gasoline are satisfactory.

The chief use of the lubricant is to cool the broach teeth and wash out chips. The pump on the broaching machine or press should deliver a sufficient quantity of lubricant to flood the work thoroughly.

Square broaches should be operated with one corner uppermost. The cutting lubricant then flows down and around both sides and does not spatter off, as it would with the flat surface on top. Spline broaches should be so slotted that the channel between two of the splines will be at the top and conduct the lubricant into the hole with the broach.

**Removing Unfinished Work from Broach.**—Should work become stuck on the broach during the cut, by the breaking of broach teeth or crowding of chips in the teeth, further damage to the broach may be avoided by placing it in a lathe and turning the piece away. If the broach has no centers, or it is more convenient, the piece may be sawed through on a milling machine or with a hack saw.

**Care of Broaches.**—Clean the broaches after using and store them on racks or shelves where teeth are in no danger of striking metal objects. Many broaches are ruined by dropping on the floor or striking hard objects which chip the teeth.

**SURFACE BROACHING** is a process of machining the surface of work by means of a broach of appropriate contour, and producing a finished surface from the rough piece in a single operation. The finished surface may be plane, curved or a combination of the two. Surface broaching is essentially a high-production operation, and since its inception about 1932 has replaced many milling operations, especially in the automotive industry. It also is used to replace surface grinding, and to rough and finish gears.

High-speed steel generally is used for surface broaches, although for certain classes of work, broaches with teeth tipped with cemented carbide have been used. Surface broaches generally are made in sections, from 6 to 14 in. long. The sections are assembled in a holder which is mounted on the ram of the broaching machine. Details of tooth spacing, chip thickness, clearances, etc., must be determined for each job. Usually some experimenting is necessary to ascertain the best form of the broach for production work.

Surface broaching has been used to machine the bearing surface and parting faces of automobile engine connecting rods, automobile steering knuckle faces, slots in speed reducing rings, shock absorber arms, certain parts of automobile engine cylinder blocks, notches in Stillson wrench and plier faces, slots in castle nuts, wrench openings, and many other similar operations. The speed of surface broaching is from 15 to 25 times as fast as for milling similar work. Joseph Geschelin, Current Practice in Surface Broaching (*Trans.* A.S.M.E., MSP—56–1, 1934) gives data of a number of surface broaching jobs. See Table 1.

**Table 1.—Surface Broaching Operations**

| Work | Operation | Material | Broach | | | | Cutting Compound |
|---|---|---|---|---|---|---|---|
| | | | Length, in. | Speed, ft. per min. | 1000 Pieces per Grind | Life, 1000 Pieces | |
| Countershaft | Form tongue, one end | 1020 C.R.S. | 48 | 30 | 7.0 | 42 | Soluble oil* |
| Shifter shaft | Cut notches and locate slot (1 operation, 2 broaches) | 1020 C.R.S. | 32 | 30 | 7.5 | 37.5 | Soluble oil |
| Brake sectors | Cutting teeth† | ........ | 24 | 25 | 6.5 | 39 | 4% Sulphurized mineral oil |
| Clutch sleeve | Cutting slots‡ | 6150 A Vanadium | 62 | 36 | 8.4 | 11.2§ | Soluble oil* |
| Transmission housing | Shift-rail slot | Cast iron | 6 1/2 | 12 | 70.0 | 150 | None |
| Shifter Fork<br>1st operation | Broach slots, both sides | 1035 Forging | 46 | 27 | 2.5 | 15 | Soluble oil* |
| 2nd " | Finish pads, cut clearance | | 47 | 27 | 2.6 | 15 | Soluble oil* |

* 10 to 1 mixture. † 61,010 1/4-in. pieces (stampings) per cut.
‡ 3 in. on each end in separate machines. § 3 slots in each piece.

Millard Romaine (*Trans.* A.S.M.E., vol. 57, 1935, p. 258) discusses the limitations of surface broaching. He shows it to be best adapted to jobs where the stock removal is comparatively small (say $^1/_4$ in. maximum) and where this stock is held within fairly close limits. A large quantity of stock to be removed necessitates an excessively long broach, with a resultant increase in the time per piece. Otherwise, excessive pressure is put on the first tooth, causing increased wear and increasing the hazard of breakage. Surface broaching has the advantage of high production and low labor cost, but has the disadvantage of high tool cost as compared with milling. At the present stage of development (1937) it is limited to the high-production industries.

**SURFACE BROACHING MACHINES** are made both horizontal and vertical, single and duplex, and for hydraulic and mechanical operation. One type of vertical machine has two broach holder slides that counterbalance each other, one rising while the other is descending on the working stroke. Each slide carries a rack meshing with a helical pinion driven by a motor. Other types use hydraulic cylinders to actuate the slides. In some machines, the hydraulic plunger pushes the broach across the work and in others it pulls it. In the duplex machines, work is clamped in fixtures mounted on a table that swings the work into the path of the descending broach. The other fixture is simultaneously swung away from the ascending broach for unloading and loading.

The following are data of vertical duplex surface broaching machines as built by the Cincinnati Milling Machine Co.:

| | | | | | | |
|---|---|---|---|---|---|---|
| Max. stroke, in. . . . . . . . . | 36 | 48 | 42 | 54 | 54 | 66 |
| Max. broach length, in. . . | 32 | 44 | 38 | 50 | 50 | 62 |
| Broaching force, lb.*. . . . | 5000 | 5000 | 10,000 | 10,000 | 20,000 | 20,000 |
| Max. speed, ft. per min. . . . | 38 | 38 | 41.4 | 41.4 | 37.2 | 37.2 |
| Motor horsepower. . . . . . | 15 | 15 | 25 | 25 | 40 | 40 |
| Overall height. . . . . . . . . . | 9′ 1″ | 11′ 1″ | 9′ 4″ | 11′ 4″ | 11′ 8″ | 13′ 8″ |
| Floor space, in. . . . . . . . . . | 97 × 50 | 97 × 50 | 97 × 55 | 97 × 55 | 115 × 64 | 115 × 64 |

* Force for which tools should be designed.

**Manufacturers.**—Surface broaching machines are built by American Broach and Machine Co., Ann Arbor, Mich.; Cincinnati Milling Machine Co., Cincinnati; Foote-Burt Co., Cleveland; LaPointe Machine Tool Co., Hudson, Mass.

# METAL SAWING

The generally-used methods of sawing metal are: 1. Hand hack saws, for occasional jobs at the bench, or on assembly work on the floor or in the field; 2. Power hack saws, for production work on all types of material and on light and heavy sections; 3. Band saws for production work on light and medium sections, and for sawing irregular contours; 4. Circular cut-off saws for very heavy sections; 5. Friction discs, for rails, structural sections, billets, bars, slabs, etc., either hot or cold.

## 1. HACK SAWS

**HACK SAW BLADES** generally are of high-speed steel, although carbon steel and semi-high-speed steel also are used for some purposes. Teeth are of various forms, arranged to cut only in one direction. Teeth without rake are the most common. Teeth with negative rake are used for thin sections, but require greater cutting pressure than straight teeth. Table 1 gives data on standard high-speed steel blades. The following are the fields of the several classes of teeth: *Regular*, soft steel, cast iron, bronze; *medium*, annealed high-carbon steel, high-speed steel; *fine*, solid brass, iron pipe, heavy tubing; *tubing*, thin tubing, sheet metals. Pitch of teeth also should be chosen with reference to

**Table 1.—Dimensions and Teeth of High-speed Hack Saw Blades**

| | Hand Saws | | Power Saws | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Length, in. | 10 | 12 | 12 | | 14 | | | 17 | | | 18 | | 21 | 24 |
| Width, in. . . . | $^9/_{16}$ | $^9/_{16}$ | 1 | 1 | 1 | 1 | 1 $^1/_4$ | 1 | 1 | 1 $^1/_4$ | 1 | 1 $^1/_4$ | 1 $^1/_2$ | 1 $^1/_2$ |
| Thickness, in. | 0.025 | .025 | .049 | .065 | .049 | .065 | .065 | .049 | .065 | .065 | .065 | .065 | .065 | .065 |
| Teeth per in. | | | | | | | | | | | | | | |
| Regular. . . | 18 | 18 | . . . . . | . . . . . | . . . . . | . . . . . | 4 | . . . . . | . . . . . | 4 | . . . . . | 4 | 4 | . . . . . |
| Medium. . . | 24 | 24 | . . . . . | 6 | . . . . . | 6 | 6 | . . . . . | 6 | 6 | . . . . . | 6 | 6 | 6 |
| Fine. . . . . . | 32 | 32 | . . . . . | 10 | . . . . . | 10 | 10 | . . . . . | 10 | 10 | 10 | . . . . . | 10 | 10 |
| Tubing. . . . | . . . . . | . . . . . | 14 | . . . . . | 14 | . . . . . | . . . . . | 14 | . . . . . | . . . . . | . . . . . | . . . . . | . . . . . | . . . . . |

the size of the section being cut. For wide heavy sections, coarse pitch teeth should be used to give ample chip clearance. For thin sections the pitch should be chosen so that two or more teeth are in contact with the work. Otherwise, the teeth will straddle the work and strip the saw.

O. W. Boston and C. E. Kraus, *Trans.* A.S.M.E., RP–56–5, 1934, describe tests to determine the performance of hack saws when operated with different cutting fluids on various metals. Of the eleven fluids tested, a sulphurized mineral oil with asphaltic base containing 4.38% sulphur, and a viscosity of 105 seconds at 100° F., was found to be the best, giving the least sawing time and the least wear on the blade.

**HACK SAW MACHINES,** or power hack saws, range in capacity from stock 4 in. square up to stock 13 × 16 in. One type, built with a gap-frame, can cut stock 16 × 26 in. by turning the work over when half sawed through. Stroke of saw is 4 in. in the smaller machines to 6 in. in the larger ones. Some machines feed by gravity, the saw frame being provided with weights that can be shifted to give greater or less pressure on the saw as required. Other machines have a positive feed which ranges from 0.001 to 0.025 in. per stroke, depending on the class of material cut. When hard spots are encountered, the feed automatically is stopped or shortened to decrease the pressure on the saw until the hard spot has been cut through. All machines lift the saw clear of the work on the return stroke. Hack saw machines are both belt-and motor-driven. Motors range from $^1/_4$ Hp. for a 4 × 4 in. machine to 2 Hp. in the largest size. High duty machines have three speeds. The Racine Tool and Machine Co. recommends the following speeds, in strokes per min.: For cold-rolled or machine steel, brass, soft metals, 135; alloy steel, annealed tool-steel, cast iron, 90; high-speed steel, unannealed tool-steel, non-corrosive steel, 60. Cast iron should be sawed dry. Other materials should be sawed with a cutting compound of water to which sufficient soluble oil has been added to whiten it. The principal function of the fluid is to cool the saw.

## 2. BAND SAWS

Metal sawing band saws are similar in appearance to band saws for woodworking. They range in capacity from the lightest sections up to work 12 in. thick, and are made in sizes from light bench-type machines to heavy production equipment. They are widely used in non-ferrous foundries for cutting of gates, sprues and risers. Band saws are built with one or more speeds to adapt them to a variety of metals. Speeds used are as follows, in feet per minute: Aluminum, 1000–2500; brass (sheets and tubes), 750–1500; cast iron, 100–150; malleable iron, 150–200; steel, 100–200; steel (high-speed) 90–120. Hand or power feed may be used. In some machines the work is fed along the table, and in others, the work is stationary, and the column and saw fed into it. Band saws are especially adapted to light thin sections, which would easily be bent out of shape by the heavier pressure of the hack saw, and to saw pieces of shapes that are inaccessible to the hack saw or circular saw.

## 3. CIRCULAR CUT-OFF SAWS

Circular cut-off saws fall into two general classes: 1. Saws with teeth whose action is similar to that of a milling cutter; 2. Friction or abrasive discs.

**METAL CUTTING SAWS** are made with teeth cut in the periphery of the disc or with inserted teeth. Saws up to 8 in. diameter generally are termed metal-slitting saws. From 9 in. diameter to 60 in., or larger, they are called cold saws.

**Metal Slitting Saws** are made with a variety of forms of teeth for sawing different materials. For steel, the teeth either are plain or have side clearance. The latter have a rake of about 10 deg. For heavy work in steel, alternate teeth have side clearance on opposite sides. For cast iron the teeth are more widely spaced than for steel, and have side clearance, and a rake of 3 to 4 deg. For non-ferrous metals, widely spaced teeth with side clearance and a rake of about 13 deg. are used. The body of the saw should be thinner than at the periphery, or the teeth should be offset to provide ample kerf to clear the saw.

Speeds and feeds with metal slitting saws vary with the class of material cut. For cast iron, speed ranges from 70 to 90 ft. per min. and feed from 3 to 6 in. per min. For ordinary steel, speeds range from 110 to 125 ft. per min., and feeds from 5 to 8 in. per min. Corresponding figures for alloy steel are 50 to 60 ft. speed, and $^1/_2$ to 2 in. feed. Brass may be cut at speeds up to 250 ft. per min., with feeds of 10 or 12 in. per min. Speeds for copper range from 150 to 300 ft. per min., and for aluminum 500 to 600 ft. per min. speed, with feeds of 10 to 20 in. per min. for both metals.

**Inserted Tooth Saws** comprise a disc, in the periphery of which notches are cut, into which cutting teeth of high-speed steel, or of steel tipped with cemented carbide are inserted. The method of holding the inserted teeth forms the principal point of difference

Table 2.—Cutting Speed and Rake Angle of Inserted Tooth Saws for Various Metals

| Material | Cutting Speed, ft. per min. | Tooth Rake, deg. | Material | Cutting Speed, ft. per min. | Tooth Rake, deg. |
|---|---|---|---|---|---|
| Aluminum | 1200 | 25–30 | Steel, 42–64,000 T. S.* | 80–90 | 15 |
| Brass, cast | 260–400 | 15–20 | " 71–85,000 " | 70–80 | 15 |
| " rolled | 400–800 | 15–20 | " 85–100,000 " | 55–65 | 15 |
| Bronze | 260–400 | 15–20 | " castings, high tensile | 45–55 | 8 |
| Cast iron | 35–45 | 8 | " " soft | 55–65 | 15 |
| Copper | 800–1200 | 20–25 | Structural steel, 64,000 T. S.* | 80–90 | 8 |
| Chrome-nickel steel, annealed | 35–45 | 15 | " " 71-85,000 T. S.* | 70–80 | 8 |
| German silver | 100–130 | 15 | Tool steel, annealed | 35–45 | 15 |
| Nickel | 70–80 | 15 | Tubes, thick wall | 80–90 | 15 |
| Rails | 55–65 | 8 | " thin " | 130–230 | 15 |
| " hard | 45–55 | 8 | Zinc | 800–1200 | 20–25 |

* Tensile strength, lb. per sq. in.

in saws as built by various manufacturers. Rake angle varies with the material being cut as shown in Table 2, adapted from O. W. Boston, Engineering Shop Practice (John Wiley & Sons). Clearance angles range from 5 to 12 deg.

Feeds with inserted tooth saws range about as follows, the figures being inches per min.: Aluminum, 90; brass, 80; cast iron, 5; copper, 60; steel 2–10.

**COLD SAW CUT-OFF MACHINES** are built with a range of speeds and feeds. They usually are individually motor-driven. Work-holding fixtures may be adapted to present work to the saw at various angles, to grip work of irregular shape, as structural sections, and to take either single pieces or a large number of pieces which are sawed at a single setting.

**FRICTION DISCS** are relatively thin discs of alloy steel which are rotated at peripheral speeds of from 20,000 to 25,000 ft. per min. The intense heat, generated by the friction of the saw against the work, fuses the metal of the latter. The disc, which is water-cooled, thus makes a thin slit in the work similar to a saw cut. Frequently, teeth of V or square shape are cut in the periphery of the disc. These teeth do not have any cutting effect, but serve to increase the friction and thus accelerate the generation of heat. When worn down they are re-cut. Friction discs are widely used for cutting off structural sections and bars. Their action is extremely rapid, cutting off 8-in. I-beams in from 5 to 15 seconds, and 40-in. beams in about $1\frac{1}{2}$ minutes. Friction discs are mounted in motor-driven machines, which often rotate the work as well as the disc.

For data on design and operation of friction saws, for sawing hot structural sections, see A. B. Pearson, Design and Construction of Hot Saws for Cutting Heavy Sections, *Trans.* A.S.M.E., IS–52–10, 1930.

**ABRASIVE CUT-OFF DISCS** are very thin abrasive wheels which are used in cut-off machines mainly for cutting off smaller sections and bars, of very hard materials and the non-ferrous metals.

## MACHINE TOOL MOTOR DRIVES

**SIZES OF MOTORS FOR MACHINE TOOLS.**—The size of motor applied to a machine tool driven by individual motor usually is determined by the experience of the manufacturer, rather than by any formula. The same lathe, for instance, will be fitted with a larger motor if it is required to take heavy roughing cuts continuously in tough steel, than if it is to have a more general run of lighter work in cast iron. Even if it does, under the latter conditions of service, occasionally receive a job up to the limit of its capacity, the motor will be able to stand the temporary overload, whereas a continuous overload soon would ruin it. The conditions under which the machine will operate should, therefore, be stated when it or the motor to drive it is purchased. Tables 1–19 give sizes of motors for machine tools as recommended by the Westinghouse Elect. & Mfg. Co. The sizes given embody average practice. The type of motor to be used varies with the conditions of service, and the type suitable for the different classes of machinery are indicated by the following symbols (see also p. 25–50):

*A.* Adjustable-speed, shunt-wound, direct-current motor; used where a number of speeds are essential.

*B.* Constant-speed, shunt-wound, direct-current motor; used when the desired speeds are obtainable mechanically by a cone pulley or gear box, or where only one speed is required.

*C*. Squirrel-cage induction motor; used where direct-current is not available. A cone pulley or gear box is necessary if more than one speed is desired.

*D*. Constant-speed, compound-wound, direct-current motor, used when different speeds are obtainable by means of a cone pulley or gear box, or where but one speed is necessary.

*E*. Wound secondary or squirrel-cage induction motor with about 10% slip; used where direct-current is not available, or where one speed is required.

*F*. Adjustable-speed, compound-wound, direct-current motor; used where a number of speeds are necessary.

*G*. Standard bending roll motor.

*H*. Standard machine tool traverse motor.

**Table 1.—Motor Horsepower Required for Lathes**

Motor: Type *A*, *B* or *C*

| Swing, in. | Service | | Swing, in. | Service | | Swing, in. | Service | |
|---|---|---|---|---|---|---|---|---|
| | Average | Heavy | | Average | Heavy | | Average | Heavy |
| | Horsepower | | | Horsepower | | | Horsepower | |
| Engine Lathes | | | | | | | | |
| 12 | 1 | 2 | 24 | 7.5 | 10–15 | 42 | 20 | 25–30 |
| 14 | 1.5 | 2–3 | 27 | 10 | 10–15 | 48 | 20 | 25–30 |
| 16 | 2.5 | 3–5 | 30 | 10 | 10–15 | 54 | 25 | 30–35 |
| 18 | 3.5 | 5–7.5 | 33 | 10 | 10–15 | 60 | 25 | 30–35 |
| 20 | 3.5 | 5–7.5 | 36 | 15 | 15–20 | 72 | 25 | 30–35 |
| 22 | 5 | 7.5–10 | .. | ..... | ..... | .. | ... | ..... |

| Driving Wheel Lathes * | | Car Wheel Lathes * | | Axle Lathes | |
|---|---|---|---|---|---|
| Swing | Horsepower | Swing | Horsepower | Swing | Horsepower |
| 51 | 15 | 42 | 40 | 10 | 15 |
| 60 | 20 | 46 | 50 | 12.5 | 25 |
| 69 | 20 | 48 | 50 | 14 | 25 |
| 79 | 25 | 50 | 50 | .. | .. |
| 84 | 25–50 | .. | .. | .. | .. |
| 90 | 30–50 | .. | .. | .. | .. |
| 100 | 50 | .. | .. | .. | .. |

* Tail stock motor 8.5 Hp. for all sizes.

**Table 2.—Motor Horsepower Required for Planers**

| Size of Planer, in. | Type of Motor | | Cross Rail Motor | Size of Planer, in. | Type of Motor | | Cross Rail Motor |
|---|---|---|---|---|---|---|---|
| | *A*, Reversing | *C*, *D* or *F*, Non-reversing | | | *A*, Reversing | *C*, *D* or *F*, Non-reversing | |
| | Horsepower | | | | Horsepower | | |
| 24×24 | 5 | 5 | 2 1/4 | 72× 72 | 35 | 35 | 5 |
| 30×30 | 5–10 | 7.5 | 2 1/4 | 84× 84 | 35–50 | 35–50 | 5 |
| 36×36 | 10 | 10 | 2 1/4 | 96× 96 | 50 | 50 | 7 1/2 |
| 42×42 | 10–20 | 15 | 3 1/2 | 120×120 | 50 | 50 | 7 1/2 |
| 48×48 | 20–25 | 20 | 3 1/2 | 144×144 | 50 | .. | 8 1/2 |
| 54×54 | 25 | .. | 3 1/2 | 168×168 | 50–75 | .. | 10 |
| 60×60 | 25–35 | .. | 5 | 192×168 | 75–100 | .. | 10 |

**Table 3.—Motor Horsepower Required for Shapers**

Ordinary Shaper. Motor: Type *A*, *B* or *C*

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Max. stroke, in. ........ | 14 | 16 | 18 | 20 | 24 | 26 | 28 | 32 |
| Horsepower ........... | 1–2 | 3–5 | 3–5 | 3–6 | 3–8 | 4–8 | 5–8 | 5–8 |

Traversing Head Shaper. Motor: Type *A*, *B* or *C*

| | | | | | | |
|---|---|---|---|---|---|---|
| Max. stroke, in. ...................... | 17 | 18 | 20 | 22 | 26 | 36 |
| Horsepower ............................ | 5 | 5 1/2 | 5–7 1/2 | 5–7 1/2 | 7 1/2–10 | 10 |

**Table 4.—Motor Horsepower Required for Slotters**

Crank Slotters. Motor: Type *A*, *B* or *C*

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Max. stroke, in... | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 28 | 30 | 36 |
| Horsepower ..... | 1–2 | 2–3 | 3–5 | 3–5 | 5–7 | 5–7.5 | 5–10 | 7.5–15 | 10–15 | 10–15 | 10–15 | 10–20 | 10–20 |

Geared Slotter. Motor: Type *A*, *B* or *C*

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Max. stroke, in... | 24 | 30 | 36 | 40 | 50 | 60 | 68 | 72 | 92 |
| Horsepower...... | 10 | 10 | 10–12 | 10–15 | 12–15 | 12–15 | 15–20 | 20–25 | 20–25 |

### Table 5.—Motor Horsepower Required for Keyseaters

Motor: Type *A*, *B* or *C*

| Size of Keyway, in. | 5/8 | 3/4 | 1 | 1 1/4 | 1 1/2 | 2 | 2 1/2 | 3 1/2 | 4 |
|---|---|---|---|---|---|---|---|---|---|
| Horsepower (for steel) | 1–2 | 1–2 | 2–3 | 2–3 | 3–5 | 3–5 | 4–6 | 5–7.5 | 5–7.5 |

### Table 6.—Motor Horsepower Required for Rotary Planers

Motor: Type *A*, *B* or *C*.

| Cutter diam., in. | 25 | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 | 120 |
|---|---|---|---|---|---|---|---|---|---|---|
| Horsepower | 5 | 7.5 | 10 | 15 | 20 | 25 | 30 | 40 | 40 | 40 |

### Table 7.—Motor Horsepower Required for Milling Machines

Universal, Plain and Vertical Milling Machines. Motor: Type *A*, *B* or *C*

| Max. Movement of Feed, in. | | | Horsepower | | | Max. Movement of Feed, in. | | | Horsepower | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Lengthwise | Lateral | Vertical | Universal Milling Machines | Plain Milling Machines | Vertical Milling Machines | Lengthwise | Lateral | Vertical | Universal Milling Machines | Plain Milling Machines | Vertical Milling Machines |
| 22 | 8 | 18* | 3–5 | 3 | ....... | 34 | 14 | 22 | ....... | ....... | 7.5–10 |
| 22 | 12 | 18 | ....... | ....... | 3–5 | 42 | 14 | 20 | 10–15 | 10–15 | ....... |
| 22 | 13 | 20 | ....... | ....... | 5–7.5 | 42 | 15 | 22 | ....... | ....... | 10–15 |
| 28 | 10 | 18* | 5–7.5 | 5–7.5 | ....... | 50 | 14 | 20* | 15–20 | 15–20 | ....... |
| 34 | 12 | 19* | 7.5–10 | 7.5–10 | ....... | 52 | 12 | 24 | ....... | ....... | 15–20 |

Slabbing Machines. Motor: Type *A*, *B* or *C*

| Vertical Millers | | | Horizontal Millers | | | | | |
|---|---|---|---|---|---|---|---|---|
| Table Width, in. | Max. Height, in. | Horsepower | Table Width, in. | Max. Height, in. | Horsepower | Table Width, in. | Max. Height, in. | Horsepower |
| 18 | 12 | 7.5 | 19 | 22 | 7.5 | 42 | 50 | 35–50 |
| 24 | 22 | 10 | 23 | 25 | 10–15 | 50 | 57 | 40–60 |
| 30 | 26 | 10 | 27 | 25 | 15–20 | 60 | 72 | 75 |
| 36 | 27 | 15 | 30 | 37 | 20–25 | .. | .. | .. |

* Vertical movement 1 in. longer in plain milling machines.

### Table 8.—Motor Horsepower Required for Drilling Machines

Motor: Type *A*, *B* or *C*

| Single-spindle Sensitive Drills | | Multiple-spindle Drilling Machines | | | | | | Upright Drills | | Radial Drills | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Max. Drill Size, in. | Horsepower | Max. Drill Diam., in.* | Max. No. of Spindles | Horsepower | Max. Drill Diam., in.* | Max. No. of Spindles | Horsepower | Max. Work Diam., in. | Horsepower | Max. Drilling Radius, ft. | Hp., Average Service | Hp., Heavy Service |
| 3/16 | 1/4 | 3/16–3/8 | 20–8 | 5 | 3/8–1 | 20–10 | 15–10 | 12–20 | 1–2 | 2.5 | 3 | ...... |
| 3/8 | 1/4–1/2 | 1/4–5/8 | 14–4 | 7.5–3.5 | 5/8–1 | 24–12 | 20–15 | 24–28 | 2–3 | 3 | 3 | 5 |
| 1/2 | 1/2–3/4 | 1/4–5/8 | 24–12 | 10–7.5 | 5/8–1 1/2 | 22–8 | 20 | 30–32 | 3–5 | 4 | 5 | 7.5 |
| 5/8 | 1/2–1 | 1/4–1 | 24–6 | 10–7.5 | 3/4–1 | 12–8 | 20–10 | 36–40 | 5–7.5 | 5 | 6–7.5 | 10 |
| 7/8 | 1 | 5/16–1 | 52–16 | 25–20 | 3/4–1 1/2 | 22–10 | 40–25 | 50–60 | 7.5–10 | 6 | 10 | 15–20 |
| 1 1/8 | 1–1.5 | 3/8–1/2 | 16–12 | 10–7.5 | 1 | 12 | 15–20 | | | 7 | 10 | 15–20 |
| | | | | | | | | | | 8 | 10 | 15–20 |

* For cast iron.

### Table 9.—Motor Horsepower Required for Boring, Turning and Drilling Machines

Motor: Type *A*, *B* or *C*

| Vertical Machines | | | | Horizontal Machines | | Cylinder Boring Machines | |
|---|---|---|---|---|---|---|---|
| Swing, in. | Horsepower | Swing, in. | Horsepower | Spindle Diam., in. | Horsepower | Max. Diam., in. | Horsepower |
| 24 | 5 | 100 | 20–25 | 2 1/2 | 5 | 20 | 7.5 |
| 36 | 5–10 | 120 | 25–35 | 3 1/2 | 5–7.5 | 30 | 7.5–10 |
| 42 | 7.5–10 | 144 | 25–35 | 4 1/2 | 7.5–10 | 40 | 10–15 |
| 48 | 10–15 | 168 | 30–40 | 5 | 10–12.5 | 50 | 10–15 |
| 54 | 10–15 | 192 | 30–40 | 6 1/2 | 10–15 | 60 | 15–20 |
| 60 | 10–15 | 240 | 40–50 | 7 | 15–20 | 70 | 15–20 |
| 72 | 15–20 | ... | ..... | 8 | 15–20 | 80 | 15–20 |
| 84 | 15–20 | ... | ..... | 9 1/2 | 20–25 | 100 | 20–25 |

### Table 10.—Motor Horsepower Required for Shears

Motor: Type *D* or *E*

| Vertical Shears | | | Lever Shears | | | | Plate Shears | | |
|---|---|---|---|---|---|---|---|---|---|
| Width, in.* | Plate Thickness, in. | Horsepower | Size, Square Bar, in.* | Horsepower | Size, Square Bar, in.* | Horsepower | Size of Plate, in. | Length of Stroke, in. | Horsepower |
| 30–42 | 1/32 | 3/4–1 | 3/4 | 2–5 | 2 1/2 | 10–20 | 3/8×24 | 3 | 10 |
| 36–62 | 1/16 | 2–3 | 1 | 3–5 | 2 3/4 | 15–20 | 1 ×24 | 3 | 15 |
| 36–144 | 1/8 | 3–10 | 1 1/4 | 5–7.5 | 3 | 15–25 | 2 ×14 | 4 1/4 | 30 |
| 36–144 | 3/16 | 4–12 | 1 1/2 | 5–7.5 | 3 1/4 | 20–30 | 1 ×42 | 4 | 20 |
| 42–168 | 1/4 | 6–20 | 1 3/4 | 5–10 | 3 1/2 | 20–40 | 1 1/2×42 | 4 1/2 | 60 |
| 54–126 | 3/8 | 15–20 | 2 | 7.5–10 | 4 | 30–50 | 1 1/4×54 | 6 | 75 |
| | | | 2 1/4 | 10–15 | | | 1 1/2×72 | 5 1/2 | 100 |

* Soft steel.

### Table 11.—Motor Horsepower Required for Punch Presses

Motor: Type *A*, *C*, *D* or *E*

| Hole Diam., in.* | Plate Thickness, in. | Horsepower | Hole Diam., in.* | Plate Thickness, in. | Horsepower | Hole Diam., in.* | Plate Thickness, in. | Horsepower | Hole Diam., in.* | Plate Thickness, in. | Horsepower |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/4 | 1/4 | 1/2–1 | 7/8 | 7/8 | 1.5–5 | 1 3/4 | 1 | 10 | 2 1/2 | 1 1/2 | 15–20 |
| 3/8 | 3/8 | 1/2–1.5 | 1 | 1 | 2–6 | 2 | 1 | 10 | 3 | 2 | 20–25 |
| 1/2 | 1/2 | 3/4–3 | 1 1/4 | 1 | 3–8 | 2 1/4 | 1 1/8 | 10–15 | 4 | 1 1/2 | 25 |
| 5/8 | 5/8 | 1.5–2 | 1 1/2 | 1 | 7.5 | 2 1/4 | 1 3/8 | 15–20 | 6 | 1 1/2 | 40 |
| 3/4 | 3/4 | 1–5 | | | | | | | | | |

* In soft steel.

### Table 12.—Motor Horsepower Required for Power Hammers

Motor: Type *D* or *E*

| Weight, lb. | Horsepower | Weight, lb. | Horsepower | Weight, lb. | Horsepower | Weight, lb. | Horsepower |
|---|---|---|---|---|---|---|---|
| 15 | 1/2–3/4 | 100 | 2–5 | 250 | 7.5–15 | 700 | 15–25 |
| 25 | 3/4–2 | 125 | 2–5 | 300 | 7.5–15 | 1000 | 25–35 |
| 50 | 1–3 | 150 | 2.5–7.5 | 400 | 10–15 | ......... | ......... |
| 75 | 1.5–3 | 200 | 5–10 | 500 | 10–15 | ......... | ......... |

Note.—For drop-hammers, approximately 1 Hp. per 100 lb. of hammer-head weight is required.

### Table 13.—Motor Horsepower Required for Forming or Bending Machines

| Bulldozers, Forming or Bending Machines. Motor: Type *D* or *E* | | | Bending or Straightening Rolls. Motor: Type *E* or *G* | | | | | |
|---|---|---|---|---|---|---|---|---|
| Width, in. | Head Movement, in. | Horsepower | Width, ft.* | Stock Thickness, in. | Horsepower | Width, ft.* | Stock Thickness, in. | Horsepower |
| 29 | 14 | 5 | 5 | 3/8 | 5–7.5 | 10 | 1 1/8 | 35–40 |
| 34 | 16 | 7 1/2 | 6 | 1/2 | 7.5–10 | 10 | 1 1/4 | 35–40 |
| 39 | 16 | 10 | 8 | 5/8 | 10–15 | 10 | 1 3/8 | 40–45 |
| 45 | 18 | 15 | 10 | 3/4 | 20–25 | 10 | 1 1/2 | 45–50 |
| 63 | 20 | 20 | 10 | 7/8 | 25–30 | 10 | 1 5/8 | 50–55 |
| | | | 10 | 1 | 30–35 | | | |

* Soft steel capacity.

### Table 14.—Motor Horsepower Required for Cold Cutting-off Machines

Motor: Type *A*, *B* or *C*

| Saw Diam., in. | Max. Diam. of Work, in.* | Horsepower | Saw Diam., in. | Max. Diam. of Work, in.* | Horsepower | Saw Diam., in. | Max. Diam. of Work, in.* | Horsepower | Saw Diam., in. | Max. Diam. of Work, in.* | Horsepower |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 12 | 3 | 1–2 | 22 | 7 | 3–4 | 36 | 11 1/2 | 10–15 | 50 | 17 | 15–25 |
| 14 | 4 | 1–2 | 24 | 7 1/2 | 3–5 | 40 | 13 | 10–15 | 54 | 18 | 20–25 |
| 16 | 5 1/2 | 1–2 | 26 | 7 1/2 | 5–7.5 | 42 | 14 | 15–20 | 58 | 19 | 20–25 |
| 18 | 5 1/2 | 2–3 | 30 | 9 | 5–7.5 | 48 | 15 | 15–25 | 62 | 21 | 20–25 |
| 20 | 6 | 3–4 | 32 | 9 1/2 | 7.5–10 | | | | | | |

* Soft round steel.

**Table 15.—Motor Horsepower Required for Bolt Machinery**

| Bolt Heading, Upsetting and Forging Machines<br>Motor: Type *D*, *E* or *F* | | | | | | Single-spindle Bolt Cutters<br>Motor: Type *A*, *B* or *C* | |
|---|---|---|---|---|---|---|---|
| Size, in. | Horsepower | Size, in. | Horsepower | Size, in. | Horsepower | Size, in. | Horsepower |
| 3/4 | 5 | 2 | 15 | 4 | 25 | 1/4–1 | 1 1/2 |
| 1 | 7.5 | 2 1/2 | 15 | 5 | 35 | 1/4–1 1/2 | 2 |
| 1 1/4 | 7.5 | 3 | 20 | 6 | 40 | 1/2–2 | 3 |
| 1 1/2 | 10 | 3 1/2 | 25 | | | | |

**Table 16.—Motor Horsepower Required for Pipe Threading and Cutting-off Machines**
Motor: Type *A*, *B* or *C*

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Pipe size, in. . . . . . . . . . | 1/4–2 | 1/4–3 | 1–4 | 1 1/4–6 | 2 1/2–8 | 2 1/2–10 | 4–12 | 8–18 |
| Horsepower. . . . . . . . . . . | 1–2 | 2–3 | 3–4 | 4–5 | 4–5 | 5–6 | 5–6 | 7.5 |

**Table 17.—Motor Horsepower Required for Cylindrical Grinding Machines**
Motor: Type *A*, *B* or *C*

| Wheel Size, in. | Distance between Centers, in. | Horse-power | Wheel Size, in. | Distance between Centers, in. | Horse-power | Wheel Size, in. | Distance between Centers, in. | Horse-power |
|---|---|---|---|---|---|---|---|---|
| 10 × 3/4 | 20–30 | 2–4 | 12 × 2 1/2 | 32–96 | 10–12 | 20 × 2 | 36–96 | 10–15 |
| 10 × 1 1/2 | 20–30 | 2–4 | 14 × 1 1/2 | 20–86 | 5–8 | 20 × 2 1/2 | 39–123 | 12–15 |
| 12 × 1/4 | 32–66 | 5–8 | 16 × 3 | 30–90 | 7.5–10 | 24 × 2 | 96–168 | 15–20 |
| 12 × 1 1/2 | 32–66 | 5–8 | 18 × 2 | 27–120 | 7.5–10 | 24 × 3 | 98–172 | 25–35 |

**Table 18.—Motor Horsepower Required for Grinding Wheel Stands**
Motor: Type *B* or *C*

| Wheel Size, in. | No. of Wheels | Horse-power | Wheel Size, in. | No. of Wheels | Horse-power | Wheel Size, in. | No. of Wheels | Horse-power |
|---|---|---|---|---|---|---|---|---|
| 6 × 1 | 2 | 1 1/2 | 14 × 3 | 2 | 4–5 | 26 × 4 1/2 | 2 | 10–12 |
| 8 × 1 | 2 | 1 1/2–2 | 16 × 3 | 2 | 5–7.5 | 30 × 4 1/2 | 2 | 10–15 |
| 10 × 2 | 2 | 2–3 | 20 × 3 | 2 | 7.5–8 | 36 × 4 | 2 | 10–15 |
| 12 × 2 1/2 | 2 | 3–4 | 24 × 3 1/2 | 2 | 8–10 | | | |

**Table 19.—Motor Horsepower Required for Hydraulic Wheel Presses**
Motor: Type *B* or *C*

| | | | | | | |
|---|---|---|---|---|---|---|
| Capacity, tons. . . . . . . . . . . . | 100 | 200 | 300 | 400 | 500 | 600 |
| Horsepower. . . . . . . . . . . . . . | 3–3.5 | 5–7.5 | 6–7.5 | 7.5–10 | 10–15 | 12.5–15 |

**GROUP DRIVES.**—While the tendency is to drive machine tools, especially the larger ones, with individual motors, there are nevertheless many conditions under which group drive will be more economical. Group drive may take various forms. For example, a series of line-shafts, each driven by a motor, with countershafts for the individual machines, is one form. Another form is a series of motor-driven short jack-shafts, driving directly a compact group of similar machines, the control and speed variation mechanism being built into the individual machines. The total motor horsepower required with group drive may range from 1/3 to 3/4 of the sum of the total power required by the individual machines, since, under the conditions under which group drive should be used, only part of the machines will be operating at any one time. The higher the load factor, the smaller are the advantages of group drive.

No definite rules can be laid down either for the arrangement of group drives or the motor capacity necessary. Each installation must be designed to conform to the conditions under which it is to operate. In general, however, the motor requirements will be the sum of the total machine horsepower multiplied by the average load factor, plus the friction loss of the line- and countershaft bearings. Motor capacity somewhat in excess of the figure so found is advisable to take care of short peaks when more machines than the average may be in operation.

The following figures show the average power utilized, in percentage of actual machine power, in group drives on various classes of work. These figures, which are given for example only, should not be used in design, as they will vary in every case with load factor, character of material cut, and with variation in machine speed, depth of cut and rate of feed.

Engine lathes, 16-in., 10–20%; automatic lathes, 36%; vertical lathes, 74%; plain milling machines, 50%; horizontal milling machines, 4-head, 35%; turret-lathes, 55%; drill presses, 24-in., 19%; multiple-spindle drills, 30%; horizontal boring mills, 18.5%; cylindrical grinders, 23%.

# MEASUREMENTS

Measurements in the machine shop may be roughly classified as non-precision, semi-precision and precision measurements. Non-precision measurements may be defined as those in which the permissible limit of error is 0.01 in. Semi-precision measurements may be defined as those in which the permissible limit of error is 0.001 in. Precision measurements are those in which the permissible limit of error is less than 0.001 in. High-precision measurements can be made within less than 0.0001 in.

## 1. NON-PRECISION MEASUREMENTS

**THE FOOT RULE** or machinist's scale ordinarily cannot be used for direct measurement with an accuracy closer than $^1/_{64}$ in., although highly expert machinists can set calipers by means of the machinist's scale to within 0.01 in.

**CALIPERS,** considered as non-precision measuring devices, may be of two types: *a.* Those that transfer a dimension from a scale or gage block to the work, or *vice versa.* *b.* Those in which the measuring scale is incorporated in the caliper itself, as shown in Fig. 1. When a vernier is added to the caliper of Fig. 1, it becomes a semi-precision measuring instrument, as the instrument then can be read to 0.001 in. The micrometer, although actually a semi-precision instrument, may be used to advantage for non-precision measurements, because of its accuracy and ease of setting to a given dimension. If a vernier is added to the micrometer it becomes a precision instrument.

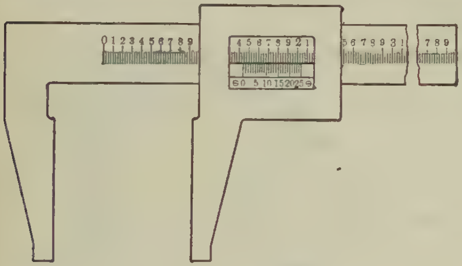

FIG. 1. Vernier Caliper

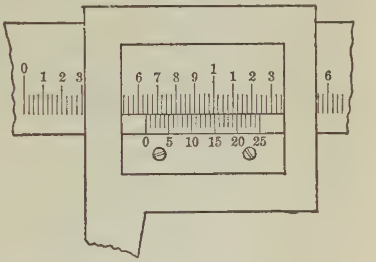

FIG. 2. Vernier

## 2. SEMI-PRECISION MEASUREMENTS

**THE VERNIER CALIPER** is shown in Fig. 1. Each main division of 0.1 in. on the scale is divided into four subdivisions. Hence, each space on the scale equals $^1/_{40}$ in. If a distance of 0.6 in. on the vernier, *i.e.*, 24 spaces on the scale, be divided into 25 spaces, the vernier will read to 0.001 in., since $^1/_{25} \times {}^1/_{40} = {}^1/_{1000}$. The vernier is read by noting the number of the division on the vernier scale that exactly coincides with a division on the main scale. This number on the vernier is the distance in thousandths of an inch that is to be added to the reading on the main scale corresponding to the number of divisions over which the zero of the vernier has moved.

EXAMPLE.—In Fig. 2, the reading on the main scale is 0.625+ in. (Each small division on the scale = 0.025 in.) The division .019 on the vernier exactly coincides with a division on the main scale. The reading of caliper then is $0.625 + 0.019 = 0.644$ in.

**THE MICROMETER** ordinarily is read to the nearest 0.001 in. The main divisions of the scale on the frame are 0.1 in., each division being subdivided into 4 equal divisions. Each space on the stem then equals $^1/_{40}$ in. (0.025 in.), which is the pitch of the micrometer screw. The circumference of the thimble is divided into 25 equal spaces. Each space on the thimble represents $^1/_{25} \times {}^1/_{40} = 0.001$ in. By the addition of a vernier scale to the stem, the micrometer can be read to 0.0001 in. The reading on the frame, plus the reading on the thimble, plus the reading on the vernier is the total distance that the end of the stem has been moved out from the anvil.

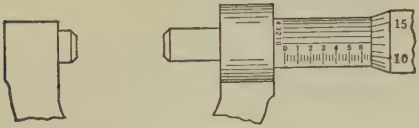

FIG. 3. Vernier Micrometer

EXAMPLE.—The thimble of the micrometer, Fig. 3, has uncovered 6 main divisions and 3 subdivisions on the frame. The scale on the thimble shows that it has been turned through 12+ divisions past the zero line on the frame. The third line on the vernier scale exactly coincides with a line on the scale on the thimble. The reading of the micrometer then is:

| | | |
|---|---|---|
| Scale on frame | 0.6 + (3 × 0.025) = | 0.675 |
| Scale on thimble | 12 × 0.001 = | .012 |
| Scale on vernier | 0.0003 = | .0003 |
| Micrometer reading | | 0.6873 |

**DIAL INDICATORS,** Fig. 4, may be used both for semi-precision and precision measurements as they are made to measure to 0.001 in. or 0.0001 in. By splitting the graduations on the dial, readings can be taken to half of the foregoing amounts. The indicator consists essentially of a movable spindle on which is cut a rack, meshing with a gear train which actuates the pointer. Movement of the spindle causes rotation of the pointer, the amount of the spindle movement being indicated by the graduations on the dial. A hair spring within the instrument returns the pointer and spindle to their initial positions when the pressure on the spindle end is removed. The dial can be rotated to bring zero opposite any position of the pointer.

Dial indicators are used in many different ways. One of the commonest uses is that of an inspection gage, as shown in Fig. 5. The instrument is set by means of a gage block or other means to the desired dimension, with sufficient pressure on the spindle to rotate the pointer through one turn. The dial then is set to zero opposite the end of the pointer. The work to be inspected is placed under the spindle and the deviation plus or minus from the standard dimension can be read on the dial. Other uses of the dial indicator are measuring the depth of holes, determining out-of-roundness of circular pieces, the setting of work parallel with a given plane, etc. When used as a depth gage, the indicator is mounted on a base through which the spindle may be projected downward by a push button until it strikes the bottom of the hole. For determining out-of-roundness, the work may be centered in a lathe. The indicator is placed in the tool post, with the spindle bearing on the work. Dial indicators are made in a variety of forms for special purposes. For details of the various forms, see catalog of L. S. Starrett Co., Athol, Mass.

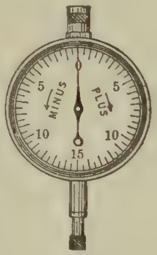

FIG. 4. Dial Indicator

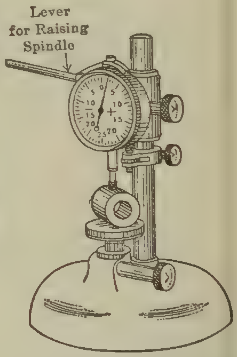

FIG. 5. Dial Indicator Used as Inspection Gage

## 3. PRECISION MEASUREMENTS

**MEASURING MACHINE.**—The standard measuring machine as built by the Pratt & Whitney Co. is a machine for originating or duplicating measurements within 0.000 01 in. It is used for checking of master gages which in turn are used to check the service gages used in manufacturing operations. The basis of the machine is a measuring bar at the rear of the machine in which are inserted at 1-in. intervals hardened steel plugs marked with a microscopic line. These graduations on the measuring bar are calibrated at the U. S. Bureau of Standards.

The measuring head consists of an anvil which is advanced by a measuring screw, the pitch of which is $^1/_{25}$ in. An index wheel on the screw is graduated to read directly to 0.0001 in., and by means of a vernier can be read to 0.000 01 in. The tailstock of the machine is set with the face of its anvil opposite the zero point on the measuring bar. The head stock, or measuring head, is set to the next greater inch on the measuring bar than the length of the piece to be measured; the fractions of the inch are determined by the measuring head. A microscope is used to determine the coincidence of the zero lines on the heads and the inch marks on the measuring bar. The work to be measured is supported on elevated ways in the plane of the anvil, with one end in contact with the tailstock anvil. The measuring head anvil then is carefully advanced by means of the screw until it bears against the opposite end of the piece to be measured. Provision is made in the machine that the pressure of the anvil on the work always will be duplicated at each measurement, this pressure ranging from 8 oz. to 3 lb., depending on the character of the work being measured.

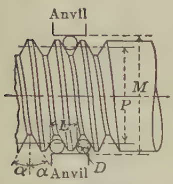

Fig. 6. Thread Measurement by Three-wire Method

The measuring machine is widely used for the measuring of pitch diameters of thread gages by the 3-wire method as shown in Fig. 6. The *wires* are steel rolls of uniform hardness. They must be uniformly round and hard, and must be straight and round within 0.000 02 in., and of the same diameter within 0.000 02 in. Furthermore, they must be the theoretically correct size within 0.000 02 in. The recommended sizes of wire for checking 60-deg. threads are given in Table 1. The pitch diameter of a thread whose sides are symmetrical about a line through the vertex perpendicular to the axis of the thread is

$$P = M + (\text{cotan}\ \alpha/2\ n) - D\ (1 + \text{cosec}\ \alpha) \quad . \ . \ . \ . \ . \ . \ . \quad [1]$$

For a 60-deg. thread with a helix angle smaller than 6 deg., equation [1] becomes

$$P = M + (0.86603/n) - 3\ D \quad . \ . \ . \ . \ . \ . \ . \quad [2]$$

For a helix of 6 deg. or more, the equation becomes

$$P = M + (\text{cotan}\ \alpha/2\ n) - D\ \{1 + \text{cosec}\ \alpha + (T^2/2)\cos\alpha\ \cot\alpha\} \quad . \ . \quad [3]$$

In these equations $P$ = pitch diameter, $M$ = measurement over the wires, $\alpha$ = 1/2 included angle of thread, $n$ = threads per inch, $D$ = diameter of wires, $T$ = tangent of helix angle = $L/\pi P$, where $L$ = lead of thread.

**Table 1.—Recommended Wire Sizes for Measuring 60-deg. Threads**

All dimensions in inches

| Threads per in. | Wire Size | Threads per in. | Wire Size | Threads per in. | Wire Size | Threads per in. | Wire Size | Threads per in. | Wire Size | Threads per in. | Wire Size |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 80 | 0.00722 | 44 | 0.01312 | 27 | 0.02138 | 14 | 0.04124 | 10 | 0.05773 | 5 | 0.11547 |
| 72 | .00802 | 40 | .01443 | 24 | .02406 | 13 | .04441 | 9 | .06415 | 4 1/2 | .12830 |
| 64 | .00902 | 36 | .01604 | 20 | .02887 | 12 | .04811 | 8 | .07217 | 4 | .14434 |
| 56 | .01031 | 32 | .01804 | 18 | .03208 | 11 1/2 | .05020 | 7 | .08248 | 3 1/2 | .16496 |
| 48 | .01203 | 28 | .02062 | 16 | .03608 | 11 | .05249 | 6 | .09623 | 3 | .19245 |

**THE ELECTROLIMIT GAGE** is a comparator that will measure under shop or production conditions to 0.000 005 in. The principle of the gage is shown in Fig. 7. The spindle $S$ with which the work to be measured is brought into contact, is continuously in contact with the armature $A$ that floats midway between the electromagnet coils $M_1$ $M_2$. The coils are so adjusted by screws $W$ that the current in them is balanced when the armature $A$ is central between the coils. A movement of spindle $S$, as produced by contact of work with it, changes the position of $A$ with respect to $M_1$ and $M_2$. The magnetic fields then are unbalanced, producing a corresponding unbalance in the current in the coils. The amount of unbalance and, therefore, the movement of spindle $S$ is shown on a microammeter, the scale of which is graduated to read in decimals of an inch suitable to the character of the work being measured. The armature $A$ is carried by a flat spring pivot $P$, at one end, and always is returned to a central position by suitable springs and stops.

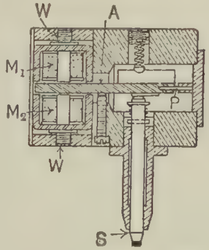

Fig. 7. Principle of Electrolimit Gage

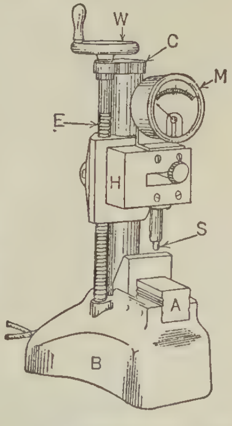

Fig. 8. Electrolimit Gage Used as a Comparator

The gage can be arranged in many different ways to adapt it to the particular type of work on which it is to be used. Several heads (Fig. 7) may be arranged in combination to measure two or more dimensions simultaneously, or to determine squareness or parallelism of a piece at the same time that a dimension is determined. Fig. 8 shows the device arranged as a simple comparator. The base $B$ supports the column $C$, which carries the gaging head $H$. The position on the column of the gaging head may be varied by the elevating screw $E$ and hand wheel $W$. The a handle. Double plug gages, Fig. 14, with a knurled handle between them are used as limit gages, the two plugs being lapped to the minimum and maximum diameters representing the prescribed limits. Plug gages should be made single-ended. If they are to be double-end limit gages, they should be made as two single-end gages with a hole in the gage into which the handle is fitted. The handle is shouldered and held by a nut. In screw gages, it also is keyed. The " no go " gage should be made shorter than the " go " gage in order to distinguish it.

Hilding Tornebohm (*Mech. Engg.*, July, 1936, p. 411) describes a combined "go" and "no go" gage that is used in Sweden. It comprises a spherical body *A*, Fig. 15, of the same diameter as the minimum permissible dimension of the hole. Near the outer edge of the body is a spherical projection *B*. The distance from the surface *B* to the diametrically opposite point on *A* is equal to the maximum permissible dimension of the hole. The gage is used by inserting it in the hole with the axis of the handle parallel to the axis of the hole. If it will enter, the hole is larger than the minimum limit. The handle then is tilted to the position *C*, moving projection *B* to position *D*, in contact with the side of the hole. If the tilting movement is impeded the diameter of the hole is less than the maximum limit. If it is not impeded the hole is too large. By turning the gage in the hole to several different positions, and tilting it as described above, the hole may be checked for roundness. Any ellipticity of the hole, within the tolerance built into the gage, *i.e.*, the distance that *B* projects above the surface of the spherical body *A*, is immediately detected. Such a check for ellipticity is not possible with the conventional plug gage.

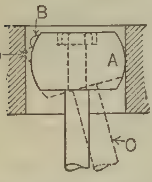

Fig. 15. Combined "Go" and "No Go" Spherical Plug Gage

Ring gages similarly may be bored and lapped to maximum and minimum diameters and used as " go " and " no go " gages. Ring gages should be knurled on the outside, and should be heavily chamfered on the " go " end to distinguish it from the " no go " end.

**Screw Plug and Ring Gages** are used to determine the correctness of screw threads. Both " no go " plug and ring gages are made to check pitch diameter only. The crest of the thread is removed down to the specified dimension of the major diameter of the " go " gage. That portion of the thread at the root of the standard form also is removed. The minimum length of thread parallel to the axis of the gage is 1.5 × basic diameter. The end threads on screw gages, both plug and ring should be flattened, not chamfered, to avoid forming a feather edge. Ring gages should be made adjustable. Plug gages for inspection service should be provided with dirt grooves extending into the gage a depth equal to one to four threads. For tolerances and limits on standard screw threads, see pp. 9–06 to 9–13. For the three-wire method of checking pitch diameter of thread gages, see p. 21–109.

**CONTOUR GAGES** are used to determine the correctness of contour of a piece. Fig. 16 shows a typical contour gage, which not only gages the contour, but determines whether or not a hole in the work is correctly located. The pin *a* is fitted into the hole in the work, which also should fit the contour in the gage recess. If limits are to be maintained, contour gages also may be made " go " and " no go."

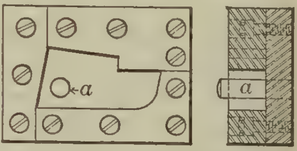

Fig. 16. Contour Gage

**STEEL FOR GAGES.**—A satisfactory steel for snap gages is a machine steel containing 0.20% C, 0.90 to 1.10% Mn, with S and P not over 0.05%. Silicon should not be present, as it tends to warp the gage in hardening. Carbon may be as low as 0.15% and as high as 0.25%. For plug and ring gages carbon may be as high as 0.50%. Gages made of these steels are pack hardened and ground. High-carbon steel, with 0.90% C, 0.30% Mn, S and P not over 0.025, and 0.15% Si, often is used for gages. When made of this steel, they can be hardened more rapidly than the machine steel gages. Alloy steels also are used for gages.

**PRECISION BLOCK GAGES** consist of a series of flats, either round or square, of hardened steel, with parallel opposite faces, lapped flat to within one or two millionths of an inch of the dimensioned size. These blocks can be wrung together to form a single gage of any desired length, with a cumulative error so small that it can be determined only by means of extremely accurate measuring apparatus. Prior to 1920, all precision blocks were imported into the U.S., but since then the Hoke blocks, made by the Pratt & Whitney Co., have had wide use. These blocks are approximately 0.95 in. square, with a hole in the center. They are made in various thicknesses, and arranged in sets. By combination of various sets and individual blocks any dimension up to 20 in. or more, varying by increments of 0.000 05 in. may be obtained. Precision blocks frequently are used as the master gage for checking reference gages.

## 5. SURFACE QUALITY MEASUREMENT

**MEASUREMENT OF SURFACE QUALITIES.**—Improvements in manufacturing methods have enabled the production of finer surfaces, *i.e.*, surfaces which approach more closely a true plane or a true cylinder. Since about 1930, methods of measuring the quality of a surface have been in process of development, and a committee of the A.S.A., sponsored by the A.S.M.E. is (1937) establishing standards for surface qualities and methods of determining them.

The character of surface varies with the method of producing it. Thus, there is a distinct difference between a turned and a milled surface, and other differences between machined and ground surfaces, and still other differences between ground, lapped and honed surfaces. In general, all surfaces consist of a series of more or less microscopic peaks and valleys. The variation in the character of surface depends on the spacing and the heights and depths of the peaks and valleys, and also on their degree of regularity or irregularity in cross-section. Fig. 17 shows characteristics of several different surfaces.

**METHODS OF MEASURING SURFACE QUALITY.**—Several methods have been devised for determination of surface quality: 1. Optical methods which include microscopic or photomicroscopic examination of specially prepared profile sections, usually copper plated before cutting; the projection microscope which throws on a screen an enlarged image of the surface, magnified from 50 to 100 diameters, also has been used. 2. Tracing the surface with a phonograph pick-up and determining the resultant voltage, or amplifying the voltage and applying it to a loud-speaker, thereby permitting a comparison of the sound produced by different surfaces. 3. Tracing the profile by means of a profilograph, which moves a sharp point over the surface. The movements of the point are enlarged by optical methods and recorded photographically. Of the three methods, the last is the only one that has been reduced to commercial use (1937). In the profilograph record the horizontal distances are compressed to about $^1/_{50}$ of their original length by gearing down the photographic drum. This causes all irregularities to appear about 50 times as sharp as they actually are. However, since initially they are relatively dull, it still is possible to distinguish individual irregularities. Fig. 17 shows typical profilograph records. For a detailed description of the profilograph, see *Auto. Ind.*, Aug. 19, 1933, p. 204.

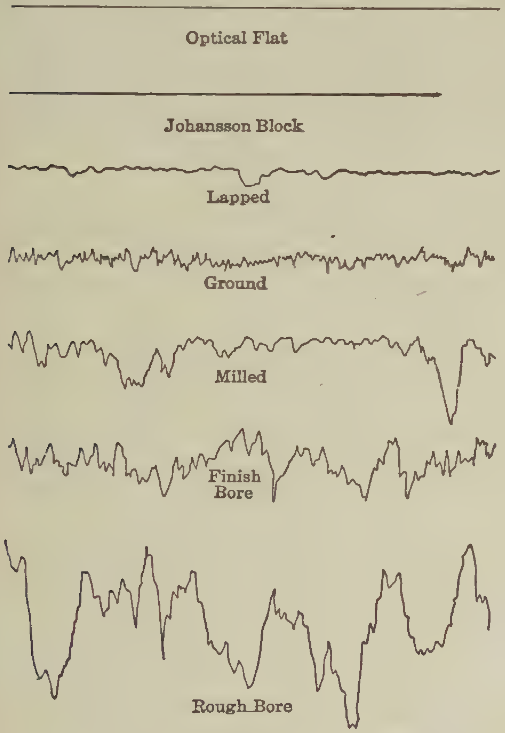

FIG. 17. Characteristics of Various Surfaces

The profilograph is essentially a laboratory instrument. Its development to an instrument for use in the shop in checking the quality of surface in production work is described in *Mech. Engg.*, March 1938, p. 205. The production instrument is known as a profilometer. The tracer point is a sharp diamond held in contact with the work by delicate spring tension. The springs are arranged so that the point is free to move only in a direction perpendicular to the surface being measured. A small coil located in the field of a permanent magnet is mounted integral with the tracer support. Movements of the tracer effect changes in the voltage in the coil which are proportional to the velocity of movement of the tracer point. The voltages are amplified and caused to produce displacements of the spot of a cathode-ray oscillograph, which displacement can be made proportional to the displacements of the tracer points, and magnified vertically to any desired degree. Magnification in a horizontal direction is obtained by photographing the displacements of the oscillograph spot on a film moving at right angles to them. The degree of horizontal magnification is determined by the speed of the film.

Among the characteristics of the profilometer are: Sensitivity sufficient to measure the smoothest metal surfaces, and sufficient ruggedness to measure surfaces 1000 times as rough without change of set up; available magnification far greater than is possible with microscopic equipment; speed of operation about 1000 times as fast as with previous instruments; a continuous running average of height of surface irregularities can be taken automatically and rapidly; presence of individual scratches and other unusual irregularities is indicated; cutting of small specimens from large pieces is unnecessary and measurements can be made directly on practically any surface, and on curved and eccentric pieces without these characteristics affecting either the reading or the record. No damage is done to the specimen and the tracer point lasts indefinitely.

Information on the state of the art to date (1938) may be obtained from the following articles:

R. E. W. Harrison, Survey of Surface Quality Standards, *Trans.* A.S.M.E., MSP 53–12, 1931; R. C. Deale, Standardization of Machine Finishes, *Mech. Engg.*, Oct. 1931, p. 723; John Gaillard, Quality of Metal Surfaces, *Am. Mach.*, Nov. 23, 1932; E. J. Abbott and F. A. Firestone, Specifying Surface Quality, *Mech. Engg.*, Sept., 1933, p. 569; E. J. Abbott, What is Surface Finish and How Can It Be Measured?, Paper before S.A.E., Detroit meeting, January 1935; Robert J. Walker, Designation of Surface Qualities, *Prod. Engg.*, August 1935; E. J. Abbott, and Edgar Goldschmidt, Surface Quality, *Mech. Engg.*, Nov. 1937, p. 813; Stewart Way, Surface Finish, *Mech. Engg.*, Nov. 1937, p. 826. E. J. Abbott and D. E. Williamson, The Profilometer, *Mech. Engg.*, March, 1938, p. 205.

# Section 22

# WOODWORKING

By Thomas D. Perry

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# WOODWORKING

By Thomas D. Perry

## 1. MATERIALS

### Logs and Logging

**RIVER LOGGING** formerly was practiced in all forest areas where cold weather facilitated operations by freezing the ground, making roads passable, and where snow aided transportation of logs on sleds to the river side. Logs were decked high at convenient waterways, awaiting Spring freshets that would float the logs to the saw mills. Saw mills had, connected by channel to the river, log ponds for storage and sorting. Logging in the woods was largely a Winter operation; the log drives and jams provided many serious hazards of bridge damage and the risk to the rivermen. Saw mills were most active in the Spring after the arrival of the logs and tapered off in the Fall. Winter logging still prevails in sparsely-wooded areas, often with second-growth timber, but is little used in the far West and deep South.

**TRACTOR LOGGING** was developed because the immense size of trees on the Pacific Coast necessitated mechanical handling of logs, especially as logs up to 40- and 60-ft. lengths were available and desirable for large timbers. The usual practice is to "snake" or "skid" logs by cables. A track cable is attached at one end to a high tree stump, and at the other to a steam or Diesel-driven "skidder." The skidder cable is reeved through a sheave that travels on the track cable, and is attached to one end of the log. The winding drum on the skidder then lifts and drags the logs to a convenient central point, where they are loaded by derrick or small skidders on railroad cars or tractor-trucks. Skidders can operate through a radius of 1000 ft.

Logging railroads usually are temporary, with little grading, and are pulled up and relocated as stumpage is exhausted. Because of excessive grades and curvature, geared locomotives are used. Corduroy roads often are required for truck and trailer operation. Truck and trailers are of several types, trailers with regular or caterpillar tractors, and trucks of the "straddle" type, where logs are hoisted and suspended between the wheels, being common.

**SWAMP LOGGING** often is imperative in the warmer districts where roadways do not freeze. Corduroy roads usually are necessary, with smaller mechanical equipment than on the Pacific Coast, as long logs are few in number, and most logs can be loaded by man power. In the larger swamps, like the Dismal Swamp on the Virginia-North Carolina line, canals have been dug to simplify log transportation, and many mills in such areas are supplied with logs by barge and tug. Logs also are handled by skidders to logging railroads built on crib-work or fills.

**SELECTIVE LOGGING** is obligatory in removing timber from government-owned and controlled areas. Such a policy can make the saw mill and its adjoining homes a continuous operation. It eliminates the inexcusable waste that occurs when a mill "cuts out," and all related real estate values are practically obliterated.

Much of the early logging, and the era is not wholly past, completely cleared the entire timber stand and left almost worthless cut-over land, frequently burned over, which usually reverted to state or national control. The re-establishment of either growing timber or any other useful yield has proved to be very expensive and to require many years. Selective logging is the economically sound practice, which permits the felling of only those trees which will produce merchantable logs, leaving well-distributed and healthy seed-propagating trees for the establishment of a continuous growth. If brush is cleared away and fire kept out, a 20-year stand should produce pulpwood, poles, etc.; a 40-year stand should supply fair-size saw logs, although some species may take longer.

**LOG SCALES** resulted from the attempts of the early timberman to state the amount of lumber that reasonably could be expected from the average log. He endeavored to eliminate the slabs, edgings and saw kerf (see p. 22–34), and to allow for the oversize of board thickness provided for shrinkage during drying. These scales were applied to the small end of the log, and the figures usually stamped thereon. The taper of the log was a factor in some scales, so that a 24-ft. log does not necessarily scale twice that of a 12-ft. log.

From this unscientific and rule-of-thumb beginning, numerous log scales developed, many of them only of local importance. A few which have received definite and wide-spread recognition are given in Table 1 for a log 16 ft. long. Other sizes, under these scales, and other scales can be found in several published woodsmen's handbooks. Fig. 1 is a graphical comparison of the several log scales applied both to sawn lumber and to veneer, using a log 18 in. diameter as an example.

**Table 1.—Comparison of Log Scales, 16 Ft. Lengths**

Cylindrical content $= \frac{1}{4}\pi D^2 \times (16/12) = \frac{1}{3}\pi D^2$ board feet

| Diam., Small End., in. | Doyle Scale | Scribner Scale | Doyle-Scribner Scale | Scribner Decimal Scale | Spaulding Scale | Cylindrical Content |
|---|---|---|---|---|---|---|
| | Board Feet | | | | | |
| 6 | 4 | 18 | 4 | 20 | ...... | 37.699 |
| 7 | 9 | 28 | 9 | 30 | ...... | 51.313 |
| 8 | 16 | 32 | 16 | 30 | ...... | 67.021 |
| 9 | 25 | 40 | 25 | 40 | ...... | 84.823 |
| 10 | 36 | 50 | 36 | 60 | 50 | 104.720 |
| 11 | 49 | 65 | 49 | 70 | 63 | 126.711 |
| 12 | 64 | 79 | 64 | 80 | 77 | 150.796 |
| 13 | 81 | 97 | 81 | 100 | 94 | 176.976 |
| 14 | 100 | 114 | 100 | 110 | 114 | 205.250 |
| 15 | 121 | 142 | 121 | 140 | 137 | 235.620 |
| 16 | 144 | 159 | 144 | 160 | 161 | 268.083 |
| 17 | 169 | 185 | 169 | 180 | 188 | 302.640 |
| 18 | 196 | 213 | 196 | 210 | 216 | 339.292 |
| 19 | 225 | 240 | 225 | 240 | 245 | 378.038 |
| 20 | 256 | 280 | 256 | 280 | 276 | 418.879 |
| 21 | 289 | 304 | 289 | 300 | 308 | 461.814 |
| 22 | 324 | 334 | 324 | 330 | 341 | 506.844 |
| 23 | 359 | 377 | 359 | 380 | 376 | 553.968 |
| 24 | 400 | 404 | 400 | 400 | 412 | 603.186 |
| 25 | 441 | 459 | 441 | 460 | 449 | 654.499 |
| 26 | 484 | 500 | 484 | 500 | 488 | 707.906 |
| 27 | 530 | 549 | 530 | 550 | 528 | 763.407 |
| 28 | 576 | 582 | 576 | 580 | 569 | 821.003 |
| 29 | 625 | 609 | 609 | 610 | 612 | 880.693 |
| 30 | 676 | 657 | 657 | 660 | 656 | 942.478 |
| 31 | 729 | 710 | 710 | 710 | 701 | 1006.36 |
| 32 | 784 | 736 | 736 | 740 | 748 | 1072.33 |
| 33 | 841 | 784 | 784 | 780 | 796 | 1140.40 |
| 34 | 900 | 800 | 800 | 800 | 845 | 1210.56 |
| 35 | 961 | 876 | 876 | 880 | 896 | 1282.82 |
| 36 | 1024 | 923 | 923 | 920 | 950 | 1357.17 |
| 37 | 1089 | 1029 | 1029 | 1030 | 1006 | 1433.60 |
| 38 | 1156 | 1068 | 1068 | 1070 | 1064 | 1512.15 |
| 39 | 1225 | 1120 | 1120 | 1120 | 1124 | 1592.80 |
| 40 | 1296 | 1204 | 1204 | 1200 | 1185 | 1675.52 |
| 41 | 1369 | 1272 | 1272 | 1270 | 1248 | 1760.40 |
| 42 | 1444 | 1343 | 1343 | 1340 | 1312 | 1847.26 |
| 43 | 1521 | 1396 | 1396 | 1400 | 1377 | 1936.26 |
| 44 | 1600 | 1480 | 1480 | 1480 | 1448 | 2027.38 |
| 45 | ...... | 1518 | 1518 | 1520 | 1512 | 2120.53 |
| 46 | 1764 | 1587 | 1587 | 1590 | 1581 | 2215.87 |
| 47 | ...... | 1656 | 1656 | 1660 | 1652 | 2312.20 |
| 48 | 1936 | 1728 | 1728 | 1730 | 1724 | 2412.74 |
| .... | ...... | ...... | ...... | ...... | ...... | ........ |

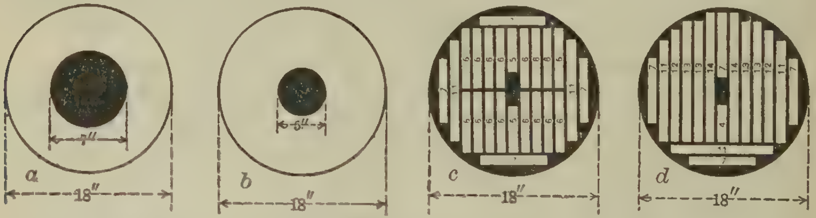


FIG. 1. Comparison of Yield from 18-in. Log. See Table 2

**Table 2.—Comparison of Yield of Veneer and Sawn Lumber by Various Log Scales**

(See Fig. 1)

| | Veneer | | Sawn Lumber | |
|---|---|---|---|---|
| | Fig. 1a | Fig. 1b | Fig. 1c | Fig. 1d |
| Cylindrical Measure: | | | | |
| Total content, board feet | 339 | 339 | 339 | 339 |
| Core or waste, board feet | 51 | 26 | 131 | 114 |
| " " " percent | 15 | 8 | 39 | 34 |
| Net product, board feet | 288 | 313 | 208 | 225 |
| " " percent | 85 | 92 | 61 | 66 |
| Net Product by Log Scales: | | | | |
| Doyle, board feet | 196 | 196 | 196 | 196 |
| " percent | 147 | 160 | 106 | 115 |
| Scribner, board feet | 213 | 213 | 213 | 213 |
| " percent | 135 | 147 | 98 | 105 |
| Scribner Decimal, board feet | 210 | 210 | 210 | 210 |
| " " percent | 137 | 149 | 99 | 107 |
| Spaulding, board feet | 216 | 216 | 216 | 216 |
| " percent | 133 | 145 | 96 | 104 |

Log scales are used both for the purchase of logs, and to estimate timber stands for valuation purposes. An experienced timber cruiser should be employed for the latter. The Spaulding and Doyle scales and their combinations are extensively used in the Midwest and the South; the Spaulding and Scribner Decimal scales are more usual on the Pacific Coast. In general, log scales favor the purchaser, rather than the seller of logs. A good grade of logs should produce the scale footage of lumber and substantial over-runs of veneer.

## Lumber

**SAW LOGS**, suitable for conversion into lumber, usually are those between the selected peelers required for veneer and the smaller logs used for poles, posts, mine props, etc. Small blocks and down-fall are dimensioned into shingle blocks or prepared for gang sawing into lath. One-inch boards usually are cut about 1 $^1/_4$ in. thick to allow for shrinkage, thickness variations, and saw marks. Corresponding allowances are made on other thicknesses. One-inch boards should *dress clean* to $^{13}/_{16}$ in., 2-in. stock to 1 $^5/_8$ in., and 4-in. stock to 3 $^1/_2$ in.

Lumber usually is marketed as *air-dried*, containing between 15% and 25% moisture, and as *kiln-dried*, approximating 10% moisture. While considerable lumber is machined or dressed green, allowance must be made for shrinkage, which may run considerably over 10%. The shrinkage and distortion varies with the position in the log from which a piece is cut. See Fig. 2. Tangential shrinkage is about twice the radial shrinkage. For data on the various species of wood, see p. 5–20.

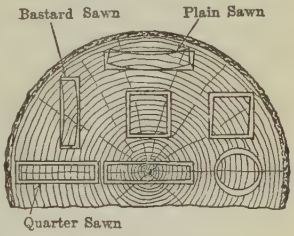


Fig. 2. Effect of Direction of Annual Rings on Shrinkage

**TYPES OF SAWING.**—Logs are sawed in various ways as follows: (See Fig. 2).

**Plain sawn** or flat grain lumber customarily is cut right through the log and nominally varies from a tangent on the growth ring to a 45 deg. angle.

**Quarter-sawn**, or vertical or edge grain, theoretically is cut on a radial line across the growth rings. In practice it may extend to 45 deg. from the radial line. In general, quarter-sawn is more carefully selected and its angular range should be considerably less than for plain sawn.

**Bastard-sawn** is a term usually applied to lumber cut between 30 deg. and 60 deg. to the radial or tangent line. Certain species like white oak, when quarter-sawn, show distinct flake markings from the wood rays (radial), which are somewhat less conspicuous in hickory, elm, sycamore, etc.

**CLASSES OF WOOD.**—The following terms are commonly used to describe different types of wood:

**Sapwood** is a name given to the several layers next to the bark, which usually are lighter in color, less dense and more easily dried than the *heartwood* that forms the center of the tree. The life processes of growth are more active near the bark, and decrease toward the heart. Heartwood is sometimes more valuable for color, as in walnut and red gum. Its density makes it more durable for use as flooring, and, in general, its greater solid content makes it more resistant to decay than sapwood. In some trees the line of color demarcation between heart and sap is not easy to detect.

**Softwood**, in general, means lumber from coniferous trees, usually evergreens.

**Hardwood** refers to lumber from broad-leaved trees. The terms have no reference to the hardness of the wood; *e.g.*, yellow pine, a softwood, is much harder than basswood, which technically is a hardwood.

**Spring Wood** is that portion of the growth ring that is formed the early part of the seasonal growth. It usually is less dense than the *Summer wood*, formed during the latter part of the yearly growth. Summer wood usually is mechanically stronger than Spring wood. A large number of growth rings per inch is desirable from a strength standpoint.

**Factory, Shop and Yard Lumber.**—In general, *factory* and *shop* lumber are intended to be cut up for use in further manufacture. *Yard* lumber is of moderate sizes and intended for general building purposes. *Structural timbers* are relatively large in size, and are selected and graded with reference to strength factors. Table 3 gives the board feet content of various sizes of dimension stock. Trade usage designates 1-in. thickness as $^4/_4$ lumber, 1 $^1/_4$-in. as $^5/_4$ lumber, etc.

**AMERICAN LUMBER STANDARDS** embody provisions for softwood lumber dealing with recognized classifications, nomenclature, basic grades, seasoning standards, sizes, uniform workings, description, measurement, tally, shipping provisions, grade markings, tally cards and inspection of lumber. These standards are in the process of becoming incorporated (1938) in the grading rules of the various lumber associations, and are making possible a more intelligent utilization of standard lumber products as well as a more accurate determination of the most desirable species and grades for any specific use.

**Table 3.—Board Measure Content of Dimension Stock**

| Thickness, in. | Width, in. | Length in inches | | | | | | | | | | Length in feet | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 4 | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 24 | 30 | 4 | 6 | 8 | 10 | 12 | 14 | 16 |
| | | Feet, Board Measure | | | | | | | | | | Feet, Board Measure | | | | | | |
| 4/4 | 1 | 0.03 | 0.04 | 0.06 | 0.07 | 0.08 | 0.10 | 0.11 | 0.13 | 0.17 | 0.21 | 0.33 | 0.50 | 0.67 | 0.83 | 1.00 | 1.17 | 1.33 |
| | 2 | .06 | .08 | .11 | .14 | .17 | .19 | .22 | .25 | .33 | .42 | .67 | 1.00 | 1.33 | 1.67 | 2.00 | 2.33 | 2.67 |
| | 3 | .08 | .13 | .17 | .21 | .25 | .29 | .33 | .38 | .50 | .63 | 1.00 | 1.50 | 2.00 | 2.50 | 3.00 | 3.50 | 4.00 |
| | 4 | .11 | .17 | .22 | .28 | .33 | .39 | .44 | .50 | .67 | .83 | 1.33 | 2.00 | 2.67 | 3.33 | 4.00 | 4.67 | 5.33 |
| | 5 | .14 | .21 | .28 | .35 | .42 | .49 | .56 | .63 | .83 | 1.04 | 1.67 | 2.50 | 3.33 | 4.17 | 5.00 | 5.83 | 6.67 |
| | 6 | .17 | .25 | .33 | .42 | .50 | .58 | .67 | .75 | 1.00 | 1.25 | 2.00 | 3.00 | 4.00 | 5.00 | 6.00 | 7.00 | 8.00 |
| | 7 | .19 | .29 | .39 | .49 | .58 | .68 | .78 | .88 | 1.17 | 1.46 | 2.33 | 3.50 | 4.67 | 5.83 | 7.00 | 8.17 | 9.33 |
| | 8 | .22 | .33 | .44 | .56 | .67 | .78 | .89 | 1.00 | 1.33 | 1.67 | 2.67 | 4.00 | 5.33 | 6.67 | 8.00 | 9.33 | 10.67 |
| | 9 | .25 | .38 | .50 | .63 | .75 | .88 | 1.00 | 1.13 | 1.50 | 1.88 | 3.00 | 4.50 | 6.00 | 7.50 | 9.00 | 10.50 | 12.00 |
| | 10 | .28 | .42 | .56 | .69 | .83 | .97 | 1.11 | 1.25 | 1.67 | 2.08 | 3.33 | 5.00 | 6.67 | 8.33 | 10.00 | 11.67 | 13.33 |
| | 11 | .31 | .46 | .61 | .76 | .92 | 1.07 | 1.22 | 1.38 | 1.83 | 2.29 | 3.67 | 5.50 | 7.33 | 9.17 | 11.00 | 12.83 | 14.67 |
| | 12 | .33 | .50 | .67 | .83 | 1.00 | 1.17 | 1.33 | 1.50 | 2.00 | 2.50 | 4.00 | 6.00 | 8.00 | 10.00 | 12.00 | 14.00 | 16.00 |
| 5/4 | 1 | .04 | .05 | .07 | .09 | 0.10 | 0.12 | 0.14 | 0.16 | 0.21 | 0.26 | 0.42 | 0.63 | 0.83 | 1.04 | 1.25 | 1.46 | 1.67 |
| | 2 | .07 | .10 | .14 | .17 | .21 | .24 | .28 | .31 | .42 | .52 | .83 | 1.25 | 1.67 | 2.08 | 2.50 | 2.92 | 3.33 |
| | 3 | .10 | .16 | .21 | .26 | .31 | .37 | .42 | .47 | .63 | .78 | 1.25 | 1.88 | 2.50 | 3.13 | 3.75 | 4.38 | 5.00 |
| | 4 | .14 | .21 | .28 | .35 | .42 | .49 | .56 | .63 | .83 | 1.04 | 1.67 | 2.50 | 3.33 | 4.17 | 5.00 | 5.83 | 6.67 |
| | 5 | .17 | .26 | .35 | .43 | .52 | .61 | .69 | .78 | 1.04 | 1.30 | 2.08 | 3.13 | 4.17 | 5.21 | 6.25 | 7.29 | 8.33 |
| | 6 | .21 | .31 | .42 | .52 | .63 | .73 | .83 | .94 | 1.25 | 1.56 | 2.50 | 3.75 | 5.00 | 6.25 | 7.50 | 8.75 | 10.00 |
| | 7 | .24 | .37 | .49 | .61 | .73 | .85 | .97 | 1.09 | 1.46 | 1.82 | 2.92 | 4.37 | 5.83 | 7.29 | 8.75 | 10.21 | 11.67 |
| | 8 | .28 | .42 | .56 | .69 | .83 | .97 | 1.11 | 1.25 | 1.67 | 2.08 | 3.33 | 5.00 | 6.67 | 8.33 | 10.00 | 11.67 | 13.33 |
| | 9 | .31 | .47 | .63 | .78 | .94 | 1.09 | 1.25 | 1.41 | 1.88 | 2.34 | 3.75 | 5.63 | 7.50 | 9.38 | 11.25 | 13.13 | 15.00 |
| | 10 | .35 | .52 | .69 | .87 | 1.04 | 1.22 | 1.39 | 1.56 | 2.08 | 2.60 | 4.17 | 6.25 | 8.33 | 10.42 | 12.50 | 14.58 | 16.67 |
| | 11 | .38 | .57 | .76 | .96 | 1.15 | 1.34 | 1.53 | 1.72 | 2.29 | 2.86 | 4.58 | 6.88 | 9.17 | 11.46 | 13.75 | 16.04 | 18.33 |
| | 12 | .42 | .63 | .83 | 1.04 | 1.25 | 1.46 | 1.67 | 1.88 | 2.50 | 3.13 | 5.00 | 7.50 | 10.00 | 12.50 | 15.00 | 17.50 | 20.00 |
| 6/4 | 1 | .04 | .06 | .08 | 0.10 | 0.13 | 0.15 | 0.17 | 0.19 | 0.25 | 0.31 | 0.50 | 0.75 | 1.00 | 1.25 | 1.50 | 1.75 | 2.00 |
| | 2 | .08 | .13 | .17 | .21 | .25 | .29 | .33 | .38 | .50 | .63 | 1.00 | 1.50 | 2.00 | 2.50 | 3.00 | 3.50 | 4.00 |
| | 3 | .13 | .19 | .25 | .31 | .38 | .44 | .50 | .56 | .75 | .94 | 1.50 | 2.25 | 3.00 | 3.75 | 4.50 | 5.25 | 6.00 |
| | 4 | .17 | .25 | .33 | .42 | .50 | .58 | .57 | .75 | 1.00 | 1.25 | 2.00 | 3.00 | 4.00 | 5.00 | 6.00 | 7.00 | 8.00 |
| | 5 | .21 | .31 | .42 | .52 | .63 | .73 | .83 | .94 | 1.25 | 1.56 | 2.50 | 3.75 | 5.00 | 6.25 | 7.50 | 8.75 | 10.00 |
| | 6 | .25 | .38 | .50 | .63 | .75 | .88 | 1.00 | 1.13 | 1.50 | 1.88 | 3.00 | 4.50 | 6.00 | 7.50 | 9.00 | 10.50 | 12.00 |
| | 7 | .29 | .44 | .58 | .73 | .88 | 1.02 | 1.17 | 1.31 | 1.75 | 2.19 | 3.50 | 5.25 | 7.00 | 8.75 | 10.50 | 12.25 | 14.00 |
| | 8 | .33 | .50 | .67 | .83 | 1.00 | 1.17 | 1.33 | 1.50 | 2.00 | 2.50 | 4.00 | 6.00 | 8.00 | 10.00 | 12.00 | 14.00 | 16.00 |
| | 9 | .38 | .56 | .75 | .94 | 1.13 | 1.31 | 1.50 | 1.69 | 2.25 | 2.81 | 4.50 | 6.75 | 9.00 | 11.25 | 13.50 | 15.75 | 18.00 |
| | 10 | .42 | .63 | .83 | 1.04 | 1.25 | 1.46 | 1.67 | 1.88 | 2.50 | 3.13 | 5.00 | 7.50 | 10.00 | 12.50 | 15.00 | 17.50 | 20.00 |
| | 11 | .46 | .69 | .92 | 1.15 | 1.38 | 1.60 | 1.83 | 2.06 | 2.75 | 3.44 | 5.50 | 8.25 | 11.00 | 13.75 | 16.50 | 19.25 | 22.00 |
| | 12 | .50 | .75 | 1.00 | 1.25 | 1.50 | 1.75 | 2.00 | 2.25 | 3.00 | 3.75 | 6.00 | 9.00 | 12.00 | 15.00 | 18.00 | 21.00 | 24.00 |
| 7/4 | 1 | .05 | .07 | 0.10 | 0.12 | 0.15 | 0.17 | 0.19 | 0.22 | 0.29 | 0.37 | 0.58 | 0.88 | 1.17 | 1.46 | 1.75 | 2.04 | 2.33 |
| | 2 | .10 | .15 | .19 | .24 | .29 | .34 | .39 | .44 | .58 | .73 | 1.17 | 1.75 | 2.33 | 2.92 | 3.50 | 4.08 | 4.67 |
| | 3 | .15 | .22 | .29 | .37 | .44 | .51 | .58 | .66 | .88 | 1.09 | 1.75 | 2.63 | 3.50 | 4.38 | 5.25 | 6.13 | 7.00 |
| | 4 | .19 | .29 | .39 | .49 | .58 | .68 | .78 | .88 | 1.17 | 1.46 | 2.33 | 3.50 | 4.67 | 5.83 | 7.00 | 8.17 | 9.33 |
| | 5 | .24 | .37 | .49 | .61 | .73 | .85 | .97 | 1.09 | 1.46 | 1.82 | 2.92 | 4.38 | 5.83 | 7.29 | 8.75 | 10.21 | 11.67 |
| | 6 | .29 | .44 | .58 | .73 | .88 | 1.02 | 1.17 | 1.31 | 1.75 | 2.19 | 3.50 | 5.25 | 7.00 | 8.75 | 10.50 | 12.25 | 14.00 |
| | 7 | .34 | .51 | .68 | .85 | 1.02 | 1.19 | 1.36 | 1.53 | 2.04 | 2.55 | 4.08 | 6.13 | 8.17 | 10.21 | 12.25 | 14.29 | 16.33 |
| | 8 | .39 | .58 | .78 | .97 | 1.17 | 1.36 | 1.56 | 1.75 | 2.33 | 2.92 | 4.67 | 7.00 | 9.33 | 11.67 | 14.00 | 16.33 | 18.67 |
| | 9 | .44 | .66 | .88 | 1.09 | 1.31 | 1.53 | 1.75 | 1.97 | 2.63 | 3.28 | 5.25 | 7.88 | 10.50 | 13.13 | 15.75 | 18.38 | 21.00 |
| | 10 | .49 | .73 | .97 | 1.22 | 1.46 | 1.70 | 1.94 | 2.19 | 2.92 | 3.65 | 5.83 | 8.75 | 11.67 | 14.58 | 17.50 | 20.42 | 23.33 |
| | 11 | .54 | .80 | 1.07 | 1.34 | 1.60 | 1.87 | 2.14 | 2.41 | 3.21 | 4.01 | 6.42 | 9.63 | 12.83 | 16.04 | 19.25 | 22.46 | 25.67 |
| | 12 | .58 | .88 | 1.17 | 1.46 | 1.75 | 2.04 | 2.33 | 2.63 | 3.50 | 4.38 | 7.00 | 10.50 | 13.00 | 17.50 | 21.00 | 24.50 | 28.00 |
| 8/4 | 1 | .06 | .08 | 0.11 | 0.14 | 0.17 | 0.19 | 0.22 | 0.25 | 0.33 | 0.42 | 0.67 | 1.00 | 1.33 | 1.67 | 2.00 | 2.33 | 2.67 |
| | 2 | .11 | .17 | .22 | .28 | .33 | .39 | .44 | .50 | .67 | .83 | 1.33 | 2.00 | 2.67 | 3.33 | 4.00 | 4.67 | 5.33 |
| | 3 | .17 | .25 | .33 | .42 | .50 | .58 | .67 | .75 | 1.00 | 1.25 | 2.00 | 3.00 | 4.00 | 5.00 | 6.00 | 7.00 | 8.00 |
| | 4 | .22 | .33 | .44 | .56 | .67 | .78 | .89 | 1.00 | 1.33 | 1.67 | 2.67 | 4.00 | 5.33 | 6.67 | 8.00 | 9.33 | 10.67 |
| | 5 | .28 | .42 | .56 | .69 | .83 | .97 | 1.11 | 1.25 | 1.67 | 2.08 | 3.33 | 5.00 | 6.67 | 8.33 | 10.00 | 11.67 | 13.33 |
| | 6 | .33 | .50 | .67 | .83 | 1.00 | 1.17 | 1.33 | 1.50 | 2.00 | 2.50 | 4.00 | 6.00 | 8.00 | 10.00 | 12.00 | 14.00 | 16.00 |
| | 7 | .39 | .58 | .78 | .97 | 1.17 | 1.36 | 1.56 | 1.75 | 2.33 | 2.92 | 4.67 | 7.00 | 9.33 | 11.67 | 14.00 | 16.33 | 18.67 |
| | 8 | .44 | .67 | .89 | 1.11 | 1.33 | 1.56 | 1.78 | 2.00 | 2.67 | 3.33 | 5.33 | 8.00 | 10.67 | 13.33 | 16.00 | 18.67 | 21.33 |
| | 9 | .50 | .75 | 1.00 | 1.25 | 1.50 | 1.75 | 2.00 | 2.25 | 3.00 | 3.75 | 6.00 | 9.00 | 12.00 | 15.00 | 18.00 | 21.00 | 24.00 |
| | 10 | .56 | .83 | 1.11 | 1.39 | 1.67 | 1.94 | 2.22 | 2.50 | 3.33 | 4.12 | 6.67 | 10.00 | 13.33 | 16.67 | 20.00 | 23.33 | 26.67 |
| | 11 | .61 | .92 | 1.22 | 1.53 | 1.83 | 2.14 | 2.44 | 2.75 | 3.67 | 4.58 | 7.33 | 11.00 | 14.67 | 18.33 | 22.00 | 25.67 | 29.33 |
| | 12 | .67 | 1.00 | 1.33 | 1.67 | 2.00 | 2.33 | 2.67 | 3.00 | 4.00 | 5.00 | 8.00 | 12.00 | 16.00 | 20.00 | 24.00 | 28.00 | 32.00 |

**Lumber Grading Rules** are promulgated by the various lumber manufacturing associations. They usually are revised from year to year, so that it is important to use the latest edition. Most of the hardwood lumbermen have combined their rules in the publications of the National Hardwood Lumber Assoc. They also make available an inspection service, under the same rules, for all controversial points of interpretation. The softwood lumbermen are grouped in several associations, usually according to species and locality. The more important lumber producers' associations, from whom current grade rule data may be obtained, are:

*Hardwood:* American Walnut Manufacturers Assoc., Chicago; Mahogany Assoc., Chicago; National Hardwood Lumber Assoc., Chicago.

*Hardwood and Softwood:* National Lumber Manufacturers Assoc., Washington; Northeastern Lumber Manufacturers Assoc., New York; Northern Hemlock and Hardwood Manufacturers Assoc., Oshkosh, Wis.

*Softwood:* Northern Pine Manufacturers Assoc., Minneapolis; Short Leaf Yellow Pine Assoc., Columbus, Miss.; Southern Cypress Manufacturers Assoc., Jacksonville, Fla.; Southern Pine Assoc., New Orleans; U. S. Red Cedar Shingle Industry, Seattle; West Coast Lumbermen's Assoc., Seattle; Western Pine Assoc., Portland, Ore.; Western Red and Northern White Cedar Assoc., Minneapolis.

## Veneer, or Thin Lumber

**VENEERS** are thin sheets of wood, which are, excepting a minor amount of sawn veneer, knife cut by various processes. See below. There is no saw kerf waste in knife cutting, rotary veneer being cut continuously from the log (See Fig. 3), and sheets of sliced veneer being cut from flitches, *i.e.*, log sections, in widths as they come, without edging (See Fig. 5). Hence the conversion of logs into veneer utilizes much more of the log than does sawing into lumber. See Fig. 1 and Table 2. The assembly of veneers into plywood distributes wood strength more effectively than does solid lumber. Hence, thinner veneers (in plywood form) provide equivalent strength factors as compared with thicker solid lumber. The four principal types of veneer, distinguished by different cutting processes, are:

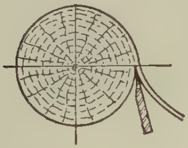


Fig. 3. Rotary Veneer Cut on a Lathe

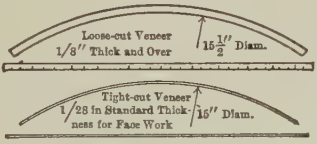


Fig. 4. Tight and Loose Cutting of Veneer

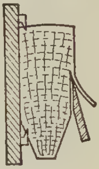


Fig. 5. Sliced Veneer Cut Quartered from a Segment Flitch

**Rotary Veneer**, cut on lathes, accounts for approximately 90% of all veneer. As very little face veneer is made by this process, with such exceptions as birdseye and quilted maple, rotary cut veneer usually is called commercial veneer. It is used for cross bands, cores and faces where strength is more important than appearance. Rotary veneer may be as thin as 0.01 in. It is difficult to cut thicker than $^1/_4$ in., although $^5/_{16}$-in. and $^3/_8$-in. veneer sometimes is cut from softer woods like basswood and poplar. Few veneer lathes are over 8 ft. long (some are 10 ft.) because of the difficulty of cutting accurately across the unsupported distance between lathe centers. Fig. 3 shows rotary cutting, the veneer being unrolled over a knife, much as paper is unrolled. The convex side is the *tight* side and often is marked with crayon during lathe cutting.

Fig. 4 shows tight and loose cut veneer. The cutting checks on the loose side may result from inadequate pressure. They cannot be eliminated entirely, and frequently cause the veneer to flatten better. In some stock, however, they sometimes are objectionable. The process of flattening veneer from its original curved shape need not result in checks or splits on the concave side in veneer under $^1/_8$ in. thick. Proper pressure produces tight cut veneer, and avoids cutting checks. Both sides of the veneer then can be used, as in matched faces. See Fig. 28 for book-matching.

**Sliced Veneer.**—Fig. 5 shows the method customarily used in cutting valuable face veneers. Much of the most attractive face figure, such as mahogany and walnut, is developed from slicing on the "quarter" as shown. Adjacent sheets can be "book-matched" by opening a flitch like the pages of a book, to produce balanced symmetrical figure. Sliced veneer does not have the cutting checks characteristic of rotary veneer. It may be as thin as $^1/_{300}$ in., but seldom can be thicker than $^1/_8$ in. It is regularly cut up to 16 ft. long.

**Half-round Veneer** cutting is an intermediate process between rotary and sliced. Two

**Table 4.—Veneer Dimensions in Inches Reduced to Square Feet**
Fractions of 1/2 or over carried to next higher decimal

| Length, in. | Width, in | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 30 | 32 | 34 | 36 |
| | Square Feet | | | | | | | | | | | | |
| 1 | 0.08 | 0.10 | 0.11 | 0.13 | 0.14 | 0.15 | 0.17 | 0.18 | 0.19 | 0.21 | 0.22 | 0.24 | 0.25 |
| 2 | .17 | .19 | .22 | .25 | .28 | .31 | .33 | .36 | .39 | .42 | .44 | .47 | .50 |
| 3 | .25 | .29 | .33 | .38 | .42 | .46 | .50 | .54 | .58 | .63 | .67 | .71 | .75 |
| 4 | .33 | .39 | .44 | .50 | .56 | .61 | .67 | .72 | .78 | .83 | .89 | .94 | 1.00 |
| 5 | .42 | .49 | .56 | .63 | .69 | .76 | .83 | .90 | .97 | 1.04 | 1.11 | 1.18 | 1.25 |
| 6 | .50 | .58 | .67 | .75 | .83 | .92 | 1.00 | 1.08 | 1.17 | 1.25 | 1.33 | 1.42 | 1.50 |
| 7 | .58 | .68 | .78 | .88 | .97 | 1.07 | 1.17 | 1.26 | 1.36 | 1.46 | 1.56 | 1.65 | 1.75 |
| 8 | .67 | .78 | .89 | 1.00 | 1.11 | 1.22 | 1.33 | 1.44 | 1.56 | 1.67 | 1.78 | 1.89 | 2.00 |
| 9 | .75 | .88 | 1.00 | 1.13 | 1.25 | 1.38 | 1.50 | 1.63 | 1.75 | 1.88 | 2.00 | 2.13 | 2.25 |
| 10 | .83 | .97 | 1.11 | 1.25 | 1.39 | 1.53 | 1.67 | 1.81 | 1.94 | 2.08 | 2.22 | 2.36 | 2.50 |
| 11 | .92 | 1.07 | 1.22 | 1.38 | 1.53 | 1.68 | 1.83 | 1.99 | 2.14 | 2.29 | 2.44 | 2.60 | 2.75 |
| 12 | 1.00 | 1.17 | 1.33 | 1.50 | 1.67 | 1.83 | 2.00 | 2.17 | 2.33 | 2.50 | 2.67 | 2.83 | 3.00 |
| 13 | 1.08 | 1.26 | 1.44 | 1.63 | 1.81 | 1.99 | 2.17 | 2.35 | 2.53 | 2.71 | 2.89 | 3.07 | 3.25 |
| 14 | 1.17 | 1.36 | 1.56 | 1.75 | 1.94 | 2.14 | 2.33 | 2.53 | 2.72 | 2.92 | 3.11 | 3.31 | 3.50 |
| 15 | 1.25 | 1.46 | 1.67 | 1.88 | 2.08 | 2.29 | 2.50 | 2.71 | 2.92 | 3.13 | 3.33 | 3.54 | 3.75 |
| 16 | 1.33 | 1.56 | 1.78 | 2.00 | 2.22 | 2.44 | 2.67 | 2.89 | 3.11 | 3.33 | 3.56 | 3.78 | 4.00 |
| 17 | 1.42 | 1.65 | 1.89 | 2.13 | 2.36 | 2.60 | 2.83 | 3.07 | 3.31 | 3.54 | 3.78 | 4.01 | 4.25 |
| 18 | 1.50 | 1.75 | 2.00 | 2.25 | 2.50 | 2.75 | 3.00 | 3.25 | 3.50 | 3.75 | 4.00 | 4.25 | 4.50 |
| 19 | 1.58 | 1.85 | 2.11 | 2.38 | 2.64 | 2.90 | 3.17 | 3.43 | 3.69 | 3.96 | 4.22 | 4.49 | 4.75 |
| 20 | 1.67 | 1.94 | 2.22 | 2.50 | 2.78 | 3.06 | 3.33 | 3.61 | 3.89 | 4.17 | 4.44 | 4.72 | 5.00 |
| 21 | 1.75 | 2.04 | 2.33 | 2.63 | 2.92 | 3.21 | 3.50 | 3.79 | 4.08 | 4.38 | 4.67 | 4.96 | 5.25 |
| 22 | 1.83 | 2.14 | 2.44 | 2.75 | 3.06 | 3.36 | 3.67 | 3.97 | 4.28 | 4.58 | 4.89 | 5.19 | 5.50 |
| 23 | 1.92 | 2.24 | 2.56 | 2.88 | 3.19 | 3.51 | 3.83 | 4.15 | 4.47 | 4.79 | 5.11 | 5.43 | 5.75 |
| 24 | 2.00 | 2.33 | 2.67 | 3.00 | 3.33 | 3.67 | 4.00 | 4.33 | 4.67 | 5.00 | 5.33 | 5.67 | 6.00 |
| 25 | 2.08 | 2.43 | 2.78 | 3.13 | 3.47 | 3.82 | 4.17 | 4.51 | 4.86 | 5.21 | 5.56 | 5.90 | 6.25 |
| 26 | 2.17 | 2.53 | 2.89 | 3.25 | 3.61 | 3.97 | 4.33 | 4.69 | 5.06 | 5.42 | 5.78 | 6.14 | 6.50 |
| 27 | 2.25 | 2.63 | 3.00 | 3.38 | 3.75 | 4.13 | 4.50 | 4.88 | 5.25 | 5.63 | 6.00 | 6.38 | 6.75 |
| 28 | 2.33 | 2.72 | 3.11 | 3.50 | 3.89 | 4.28 | 4.67 | 5.06 | 5.44 | 5.83 | 6.22 | 6.61 | 7.00 |
| 29 | 2.42 | 2.82 | 3.22 | 3.63 | 4.03 | 4.43 | 4.83 | 5.24 | 5.64 | 6.04 | 6.44 | 6.85 | 7.25 |
| 30 | 2.50 | 2.92 | 3.33 | 3.75 | 4.17 | 4.58 | 5.00 | 5.42 | 5.83 | 6.25 | 6.67 | 7.08 | 7.50 |
| 31 | 2.58 | 3.01 | 3.44 | 3.88 | 4.31 | 4.74 | 5.17 | 5.60 | 6.03 | 6.46 | 6.89 | 7.32 | 7.75 |
| 32 | 2.67 | 3.11 | 3.56 | 4.00 | 4.44 | 4.89 | 5.33 | 5.78 | 6.22 | 6.67 | 7.11 | 7.56 | 8.00 |
| 33 | 2.75 | 3.21 | 3.67 | 4.13 | 4.58 | 5.04 | 5.50 | 5.96 | 6.42 | 6.88 | 7.33 | 7.79 | 8.25 |
| 34 | 2.83 | 3.31 | 3.78 | 4.25 | 4.72 | 5.19 | 5.67 | 6.14 | 6.61 | 7.08 | 7.56 | 8.03 | 8.50 |
| 35 | 2.92 | 3.40 | 3.89 | 4.38 | 4.86 | 5.35 | 5.83 | 6.32 | 6.81 | 7.29 | 7.78 | 8.26 | 8.75 |
| 36 | 3.00 | 3.50 | 4.00 | 4.50 | 5.00 | 5.50 | 6.00 | 6.50 | 7.00 | 7.50 | 8.00 | 8.50 | 9.00 |
| 37 | 3.08 | 3.60 | 4.11 | 4.63 | 5.14 | 5.65 | 6.17 | 6.68 | 7.19 | 7.71 | 8.22 | 8.74 | 9.25 |
| 38 | 3.17 | 3.69 | 4.22 | 4.75 | 5.28 | 5.81 | 6.33 | 6.86 | 7.39 | 7.92 | 8.44 | 8.97 | 9.50 |
| 39 | 3.25 | 3.79 | 4.33 | 4.88 | 5.42 | 5.96 | 6.50 | 7.04 | 7.58 | 8.13 | 8.67 | 9.21 | 9.75 |
| 40 | 3.33 | 3.89 | 4.44 | 5.00 | 5.56 | 6.11 | 6.67 | 7.22 | 7.78 | 8.33 | 8.89 | 9.44 | 10.00 |
| 41 | 3.42 | 3.99 | 4.56 | 5.13 | 5.69 | 6.26 | 6.83 | 7.40 | 7.97 | 8.54 | 9.11 | 9.68 | 10.25 |
| 42 | 3.50 | 4.08 | 4.67 | 5.25 | 5.83 | 6.42 | 7.00 | 7.58 | 8.17 | 8.75 | 9.33 | 9.92 | 10.50 |
| 43 | 3.58 | 4.18 | 4.78 | 5.38 | 5.97 | 6.57 | 7.17 | 7.76 | 8.36 | 8.96 | 9.56 | 10.15 | 10.75 |
| 44 | 3.67 | 4.28 | 4.89 | 5.50 | 6.11 | 6.72 | 7.33 | 7.94 | 8.56 | 9.17 | 9.78 | 10.39 | 11.00 |
| 45 | 3.75 | 4.38 | 5.00 | 5 63 | 6.25 | 6.88 | 7.50 | 8.13 | 8.75 | 9.38 | 10.00 | 10.63 | 11.25 |
| 46 | 3.83 | 4.47 | 5.11 | 5.75 | 6.39 | 7.03 | 7.67 | 8.31 | 8.94 | 9.58 | 10.22 | 10.86 | 11.50 |
| 47 | 3.92 | 4.57 | 5.22 | 5.88 | 6.53 | 7.18 | 7.83 | 8.49 | 9.14 | 9.79 | 10.44 | 11.10 | 11.75 |
| 48 | 4.00 | 4.67 | 5.33 | 6.00 | 6.67 | 7.33 | 8.00 | 8.67 | 9.33 | 10.00 | 10.67 | 11.33 | 12.00 |
| 49 | 4.08 | 4.76 | 5.44 | 6.13 | 6.81 | 7.49 | 8.17 | 8.85 | 9.53 | 10.21 | 10.89 | 11.57 | 12.25 |
| 50 | 4.17 | 4.86 | 5.56 | 6.25 | 6.94 | 7.64 | 8.33 | 9.03 | 9.72 | 10.42 | 11.11 | 11.81 | 12.50 |
| 51 | 4.25 | 4.96 | 5.67 | 6.38 | 7.08 | 7.79 | 8.50 | 9.21 | 9.92 | 10.63 | 11.33 | 12.04 | 12.75 |
| 52 | 4.33 | 5.06 | 5.78 | 6.50 | 7.22 | 7.94 | 8.67 | 9.39 | 10.11 | 10.83 | 11.56 | 12.28 | 13.00 |
| 53 | 4.42 | 5.15 | 5.89 | 6.63 | 7.36 | 8.10 | 8.83 | 9.57 | 10.31 | 11.04 | 11.78 | 12.51 | 13.25 |
| 54 | 4.50 | 5.25 | 6.00 | 6.75 | 7.50 | 8.25 | 9.00 | 9.75 | 10.50 | 11.25 | 12.00 | 12.75 | 13.50 |
| 55 | 4.58 | 5.35 | 6.11 | 6.88 | 7.64 | 8.40 | 9.17 | 9.93 | 10.69 | 11.46 | 12.22 | 12.99 | 13.75 |
| 56 | 4.67 | 5.44 | 6.22 | 7.00 | 7.78 | 8.56 | 9.33 | 10.11 | 10.89 | 11.67 | 12.44 | 13.22 | 14.00 |
| 57 | 4.75 | 5.54 | 6.33 | 7.13 | 7.92 | 8.71 | 9.50 | 10.29 | 11.08 | 11.88 | 12.67 | 13.46 | 14.25 |
| 58 | 4.83 | 5.64 | 6.44 | 7.25 | 8.06 | 8.86 | 9.67 | 10.47 | 11.28 | 12.08 | 12.89 | 13.69 | 14.50 |
| 59 | 4.92 | 5.74 | 6.56 | 7.38 | 8.19 | 9.01 | 9.83 | 10.65 | 11.47 | 12.29 | 13.11 | 13.93 | 14.75 |
| 60 | 5.00 | 5.83 | 6.67 | 7.50 | 8.33 | 9.17 | 10.00 | 10.83 | 11.67 | 12.50 | 13.33 | 14.17 | 15.00 |
| 63 | 5.25 | 6.13 | 7.00 | 7.88 | 8.75 | 9.63 | 10.50 | 11.38 | 12.25 | 13.13 | 14.00 | 14.88 | 15.75 |
| 66 | 5.50 | 6.42 | 7.33 | 8.25 | 9.17 | 10.08 | 11.00 | 11.92 | 12.83 | 13.75 | 14.67 | 15.58 | 16.50 |
| 69 | 5.75 | 6.71 | 7.67 | 8.63 | 9.58 | 10.54 | 11.50 | 12.46 | 13.42 | 14.38 | 15.33 | 16.29 | 17.25 |
| 72 | 6.00 | 7.00 | 8.00 | 9.00 | 10.00 | 11.00 | 12.00 | 13.00 | 14.00 | 15.00 | 16.00 | 17.00 | 18.00 |
| 78 | 6.50 | 7.58 | 8.66 | 9.75 | 10.83 | 11.92 | 13.00 | 14.08 | 15.17 | 16.25 | 17.33 | 18.42 | 19.50 |
| 84 | 7.00 | 8.17 | 9 33 | 10.50 | 11.67 | 12.83 | 14.00 | 15.17 | 16.33 | 17.50 | 18.67 | 19.83 | 21.00 |
| 90 | 7.50 | 8.75 | 10.00 | 11.25 | 12.50 | 13.75 | 15.00 | 16.25 | 17.50 | 18.75 | 20.00 | 21.25 | 22.50 |
| 96 | 8.00 | 9.33 | 10.67 | 12.00 | 13.33 | 14.67 | 16.00 | 17.33 | 18.67 | 20.00 | 21.33 | 22.67 | 24.00 |

distinct types are cut, *regular* and *back-cut.* See Fig. 6. Cutting is done in a lathe in which the log or flitch is eccentrically swung between centers on a *stay log*, the radius of knife cut being much greater than in ordinary rotary cutting.

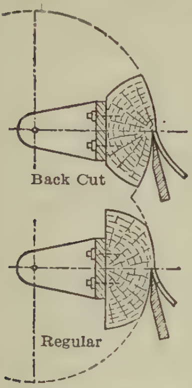


FIG. 6. Half-round Veneer

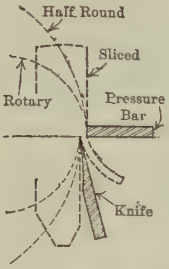


FIG. 7. Positions of Pressure Bar and Knife in Veneer Cutting

Skillful cutting of these three types of veneer depends principally on proper adjustment and manipulation of the relations between the knife and the pressure bar. See Fig. 7.

**Sawn Veneer.**—A few face veneers are sawn on a large diameter segment saw, particularly to reveal the flake of woods like quartered white oak. The minimum thickness for sawn veneer is $^1/_{20}$ in., when the saw kerf waste is equal to the veneer; the maximum may be $^3/_8$ in. or more.

**Standard Thicknesses** for veneer and cutting tolerances are in the process of establishment (1937) by the various associations and the U. S. Bureau of Standards. For description of grade rules and face figure nomenclature, consult the publications of the veneer associations, the more prominent of which are: American Walnut Manufacturers Assoc., Chicago; Mahogany Assoc., Chicago; National Hardwood Lumber Assoc., Chicago; Pacific Veneer Package Assoc., Tacoma, Wash.; Rotary Cut Lumber Assoc., New Orleans; Veneer Assoc., Chicago; Veneer Package Assoc., Washington.

## 2. PRIMARY CONVERSION MACHINERY

### Saw Mills

Saw mills are either permanent or portable. The former are used to saw the timber from large tracts, where the life of the operation will be 10 years or more. The latter are used in small tracts. They are so constructed that they easily can be moved from place to place, and are driven either by steam, gasoline or Diesel engine. Both types of mill use either band saws or circular saws. The former are higher in first cost but the waste in saw kerf is much less.

**ARRANGEMENT OF SAW MILLS.**—Saw mills may be either single or double, *i.e.*, the equipment of a double mill largely is in duplicate, and arranged symmetrically on either side of the center line of the mill. Modern (1938) practice tends to the use of band saws; a mill with one such saw is known as a *single-band mill*; one with two saws is a *double-band mill.*

Fig. 8 shows the arrangement of a typical double-band mill designed to cut 100,000 board feet of lumber per day, with a steam gang saw for sawing baulks of timber into boards. In many mills a horizontal or vertical band-resaw replaces the gang saw. The live roll conveyor arrangement then is somewhat different from that shown. The machinery floor of the mill is elevated 15 to 20 ft. above ground level. Line-shafts or other drives are located below the floor.

Logs are drawn from the log pond or other storage and elevated to the mill floor by a *log slip, i.e.*, an inclined chain conveyor, which may be single or in duplicate. At the head of the log slip, the logs are cut into convenient lengths by a deck saw. This is a horizontal or vertical swing saw, 6 ft. or more diam., belt or motor driven, moved to and fed through the log by steam or air cylinders. Steam-actuated *kickers, i.e.*, steam cylinders whose piston rods terminate in large forked ends, kick the sawed logs from the log slip to the *log deck*, whence they are transferred to the saw carriages. The log deck slopes downward to the carriages, and the logs are moved across it either by gravity or by a power-driven chain-haul. Steam *niggers* transfer the logs from the log deck to the saw carriages, and also turn over the partially sawed logs to present new surfaces to the saw.

The carriages carry the logs back and forth past the band saws. An operator riding on the carriage, by means of appropriate *set-works*, advances the log toward the saw after each cut to obtain the desired thickness of lumber.

The lumber as it comes from the saw is moved by the live roll conveyors and chain transfers to any one of a series of operations. Lumber that is sawed to correct widths moves on the first roller conveyor to the transfer chain to the trimming saws, which cut it to proper length. Lumber which is of irregular width is moved by chain transfers to the double-edgers which cut it to width, whence it passes to the trimming saws. Large baulks of timber are moved by the chain transfers to pass through the steam gang saw which saws the baulk into planks, which then move to the trimming saws.

If a band resaw is used instead of a gang saw, the baulks travel on the first live roll bed to the far end of the mill. A chain transfer then moves them to the middle live roll bed, which returns them in the reverse direction to the band resaw, where they are sawed into boards. The chain transfer moves these boards again to the first live roll bed, which delivers them to the trimming saw.

Refuse from the various operations is disposed of in several ways. Sawdust from each operation drops into a conveyor which moves it either to a fuel storage or direct to the boiler plant. Trimmings, edgings, and other pieces that are large enough are placed on conveyors to transport them to the lath and shingle mills for conversion into marketable product. Other waste is diverted to the slasher which saws it into pieces short enough to be used as fuel. In some mills the slasher delivers it to a fuel hog for further preparation for the boiler furnaces (see Kent's Mechanical Engineers' Handbook—Power, p. 443, forming Vol. 2 of this series).

Finished lumber from the trimming saws is placed on the sorting chain. This is a two- or three-strand chain conveyor from 400 to 800 feet long, on which the lumber is graded and sorted into convenient stacks for transfer to the lumber yard or dry kilns.

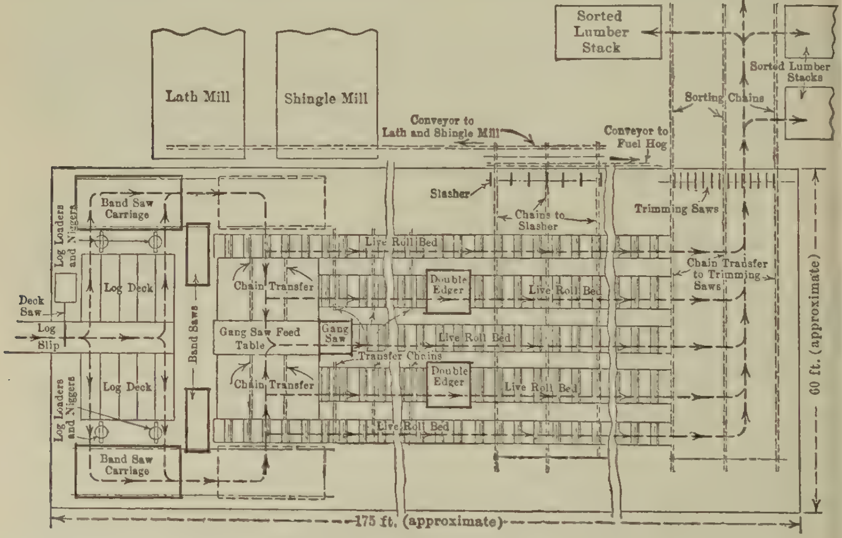


FIG. 8. Double-band Saw Mill

**SAW MILL MACHINERY.**—The following paragraphs briefly describe the various items of equipment commonly used in large saw mills. Manufacturers of equipment should be consulted for detailed specifications and advice concerning the equipment to use for any particular set of conditions.

**Band Mills.**—A band mill comprises a heavy cast-iron frame and column carrying an upper and lower band wheel on which runs a band saw. The upper wheel usually has a cast-iron rim with steel spokes cast in place. The lower wheel sometimes is of the same construction and sometimes is made entirely of cast iron. The shaft and bearings of the upper wheel are movable in a vertical plane for the purpose of putting the proper tension on the saw. Various types of mechanism or *straining devices* are used by different manufacturers to maintain uniform tension on the saw. The mill is driven by a belt pulley on the shaft of the lower wheel.

Band mills commonly are made with band wheels of from 6 to 10 ft. diameter. The diameter of the wheel designates the size of the mill. The saws range from 10 in. wide, No. 16 gage, on 6-ft. mills, up to 18 in., No. 12 gage, on 10-ft. mills. The length of the saw varies with the class of work to be done. For yellow pine, and similar hardwood, a short stiff saw is preferable to one that is more flexible. Fig. 9 and Table 5 give the overall dimensions of Allis-Chalmers band mills. Table 6 gives the capacity and horsepower requirements of these mills.

**Saw Carriages** are massive timber frames carried on 8 to 12 wheels, riding on T-rails. Rails weigh from 60 to 90 lb. per yard. The headblocks, which carry movable knees, are mounted on the carriage. The knees each carry two sets of dogs with teeth pointing upward and downward, respectively. These dogs grip the baulk of partially-sawed timber, and hold it firmly against the knee while the carriage moves it past the saw. Hook dogs on each knee are used to hold round logs against the knee. The number of headblocks depends on the length of logs sawed. Up to 30-ft.

**Table 5.—Dimensions of Band Mills.** (See Fig. 9)
(Allis-Chalmers Co., Milwaukee, Wis.)

| Size of Mill, *D*, ft. | *A* | *B* | *C*, in. | *E*, in. | *F*, in. | *G*, in. | *H*, in. | *J*, in. | *K* | *L*, in. | *M*, in. | *N*, in. | *P*, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 8′ 4″ | 9′ 9″ | 12 | 48 | 45 | 21 | 25 7/8 | 28 5/8 | 8′ 4 1/2″ | 15 | 38 | 9 | 22 |
| 7 | 9′ 4″ | 10′ 3″ | 12 | 52 | 47 | 21 | 27 7/8 | 30 5/8 | 8′ 9 1/2″ | 18 | 44 | 9–11 | 25 |
| 8 | 10′ 8″ | 11′ 9″ | 14 | 62 | 51 | 27 1/2 | 28 7/8 | 31 5/8 | 9′ 11 3/4″ | 18 | 50 | 11–13 | 27 |
| 9 | 11′ 8″ | 12′ 4″ | 14 | 67 | 53 | 27 1/2 | 30 7/8 | 33 5/8 | 11′ 7″ | 18 | 56 | 11–15 | 31 1/2 |
| 9* | 11′ 8″ | 12′ 4″ | 14 | 67 | 53 | 27 1/2 | 30 7/8 | 33 5/8 | 10′ 3″ | 18 | 56 | 11–15 | 31 1/2 |
| 10 | 13′ 0″ | 12′ 11″ | 16 | 68 | 55 | 31 1/4 | 35 7/8 | 38 5/8 | 13′ 7″ | 23 | 62 | 14–16 | 31 1/2 |
| 10* | 13′ 0″ | 12′ 11″ | 16 | 68 | 55 | 31 1/4 | 35 7/8 | 38 5/8 | 12′ 3″ | 23 | 62 | 14–16 | 31 1/2 |

* Low mill.

**Table 6.—Saw Dimensions, Capacity and Horsepower of Band Mills**
(Allis-Chalmers Co., Milwaukee, Wis.)

| Size of Mill, ft. | Saws | | | Logs, Max. Diam., in. | | | | Surface Speed of Saw, ft. per min. | | | Hp.† Rating of Motor |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Length | | Gage No. | Split | | Slabbed | | 10,000 | 9000 | 8500 | |
| | Max. | Min. | | Round | Square | 4-in. slab | 8-in. slab | R. p. m. | | | |
| 6 | 38′ 1″ | 35′ 7″ | 16 | 42 | 34 | 53 | ...... | 531 | 477 | 451 | 75 |
| 7 | 42′ 7″ | 39′ 7″ | 15 | 45 | 36 | 65 | ...... | 455 | 410 | 385 | 100 |
| 8 | 48′ 0″ | 45′ 0″ | 14 | 48 | 38 | 72 | ...... | 400 | 360 | 340 | 150 |
| 9 | 54′ 5″ | 51′ 5″ | 13 | 62 | 50 | 96 | 88 | 353 | 318 | 300 | 200 |
| 9* | 51′ 9″ | 48′ 9″ | 13 | 46 | 37 | 72 | 64 | 353 | 318 | 300 | 200 |
| 10 | 62′ 5″ | 58′ 7″ | 12 | 75 | 62 | 124 | 116 | 318 | 286 | 270 | 250 |
| 10* | 59′ 9″ | 55′ 11″ | 12 | 61 | 49 | 101 | 92 | 318 | 286 | 270 | 250 |

* Low mill. † With babbitted bearings; deduct 20% for roller bearings.

logs, 2-block carriages are usual. For logs over 30 ft. long, three headblocks generally are used. After each saw cut, the log or baulk is advanced the desired amount for the next cut, by means of *set-works* which move the knees forward. The set-works are of varying design, and may be actuated by air or steam cylinders, or by hand. Appropriate mechanism limits the travel of the set-works to a predetermined amount, thereby enabling exact thicknesses of lumber to be cut. Set-works usually are designed to advance the knees by multiples of 1/32 in. The operator rides on the carriage and manipulates the set-works in accordance with directions from the sawyer. Overall width of carriages as made by the Filer and Stowell Co. range from 6 ft. 8 in. to 10 ft. 2 1/2 in. The length depends on the length of logs cut.

Two general methods are in use for traversing the carriage past the saw, *steam* or "*shot gun*" feed and *twin-engine feed*. The former is a steam cylinder from 10 to 16 in. diameter, and of stroke equal to the total travel of the carriage. It is located with its axis on the center line of the carriage. Total travel is determined by the length of the longest log, plus 15 to 20 ft. The piston rod is attached to the carriage. Steam admitted to one end of the cylinder causes the carriage to move the log against the saw. Admitted to the other end, it returns the carriage to its original position.

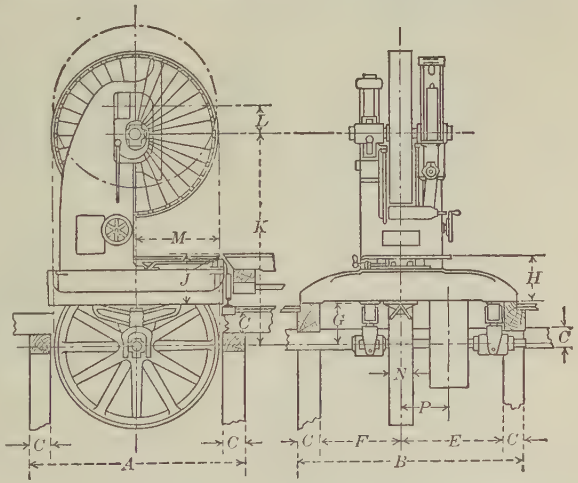


Fig. 9. Band Mill

The twin-engine feed comprises a pair of steam engines, with cranks set at an angle to each other, driving a winding drum either directly or through gears. An endless cable on the winding drum runs over a sheave at the far end of the carriage track and is attached to the carriage. The carriage is moved back and forth by winding or unwinding the cable on the drum. One widely-used type of twin-engine feed has oscillating engine cylinders. The speed and direction of rotation are controlled by valves operated by the sawyer. Cylinder sizes range from 9 × 12 in. to 16 × 20 in., and winding drums range from 27 in. diam. by 11 in. face, using 3/4-in. rope, to 48 in. diam. by 23 1/2 in. face, using 1 1/8-in. rope. Carriage travel up to 200 ft. can be obtained with this type of feed. The twin-engine feed is more economical of steam than the shot-gun feed, but is slower in operation.

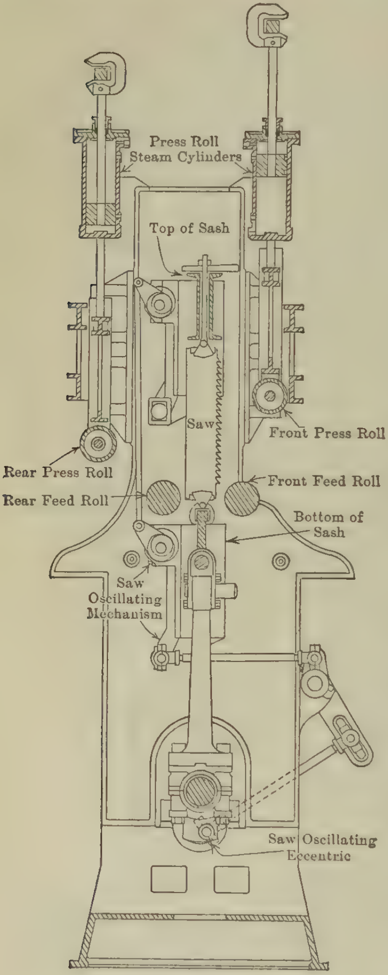


Fig. 10. Gang Saw

**Gang Saws** are reciprocating saws, mounted in gangs to saw large baulks of lumber into a number of planks in one operation. They are either belt- or engine-driven. From 25 to 54 saws are mounted parallel to each other, the requisite distance apart to saw planks of the desired thickness. They are carried in a steel sash which is reciprocated vertically by a pitman driven by a crankshaft below the sash. In the Wickes gang saw, Fig. 10, the saw is oscillated on the down stroke, to vary the depth of cut in accordance with the variation in linear speed of the saw due to the change in position of the crank. This is accomplished by means of an eccentric on the crank-shaft which operates linkwork to move the lower guides of the sash forward at about 5/8 of the upward stroke of the saw, continuing the forward motion until 5/8 of the down stroke has been completed. The guides then move backward, and free the saw from the cut on the return stroke. The path of the saw teeth on the down stroke is an oval, the depth of cut being greatest at the point in the stroke where the linear speed is highest. The amount of oscillation is increased as feed increases, so that the saw always works at maximum efficiency. Work is fed to the saws by fluted feed rolls, driven by a friction feed, whose speed can be varied by the operator at will. Pressure rolls, actuated by steam cylinders, hold the work firmly in place during sawing. Table 7 gives data on gang saws.

**Band Resaws.**—Band resaws are of both horizontal and vertical types. The vertical type is, in general, similar to the band mill, with the exception that it has adjustable feed rolls on either side of the saw to provide for resawing various thicknesses of stock. The feed rolls usually are power operated, and can be set to saw thicknesses varying by 1/32 in.

The horizontal band resaw comprises a massive column supporting on either side of it bearings for the band wheels. The saw blade is horizontal. Underneath the upper run of the saw and at right angles to it are one or two adjustable feed tables which can be raised or lowered to vary the thickness of the stock sawed. Above the feed table, and before and behind the saw, are sets of pressure rolls, independently adjustable, to permit two or more different thicknesses of slab to be sawed at the same time. If but a single thickness of work is to be sawed the feed table is made stationary.

Filer and Stowell Co. gives the following data concerning single-table horizontal band resaws:

| Size of Bed | | Pulley | | | Saw | | Lift of Press, Rolls, in. | Drop of Table, in. | Horsepower |
|---|---|---|---|---|---|---|---|---|---|
| Resaw | Width, in. | Diam., in. | Face, in. | R.p.m. | Max. Width, in. | Gage No. | | | |
| 6 ft. | 36 | 32 | 17 | 420–450 | 10 | 16 | 13 | 4 1/2 | 75 |
| 8 ft. | 36 | 44 | 25 | 320–340 | 12 | 14 | 13 | 4 1/2 | 100 |

Double-table resaws have the same dimensions, excepting that the tables are 16 and 24 in. wide.

**Edgers.**—Double edgers consist of one or more circular saws so mounted on the saw arbor that they can be moved on it in an axial direction. The position of the saws is controlled by the operator, who sets them to cut whatever width of stock may be desired. Double edgers are used to remove the rough or wany edges from slabs to form a single wide plank, or to saw a wide slab into several narrower planks. Saws range from 20 to 24 in. diameter, and usually are direct motor driven through a flexible coupling, although belt drive also is used. Double edgers are made ranging

**Table 7.—Sizes and Capacities of Gang Saws**

(Wickes Bros., Saginaw, Mich.)

| Saw No. | Width between Stiles, in. | Cut, in. | Stroke, in. | Pulley Size, in. | R.p.m., max. | Hp., average | Saws | | Feed, in. | | Capacity per 8 hr., M Board Feet |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Size, in. | No. | Max. | Min. | |
| 10 | 44 | 18 | 20 | 72 × 25 | 240 | 200 | 8 × 47 | 36 | 1 1/28 | 5/16 | 140–175 |
| 10 | 48 | 18 | 20 | 72 × 30 | 240 | 250 | 8 × 47 | 40 | 1 1/28 | 5/16 | 160–200 |
| 10 | 52 | 18 | 20 | 72 × 30 | 240 | 250 | 8 × 47 | 44 | 1 1/28 | 5/16 | 180–215 |
| 10 | 60 | 18 | 20 | 72 × 33 | 240 | 300 | 8 × 47 | 52 | 1 1/28 | 5/16 | 200–240 |
| 15 | 32 | 8 | 10 | 36 × 8 | 275 | 40 | 4 5/8 × 24 | 27 | 1/2 | 1/8 | 20–30 |
| 16 | 40 | 14 | 15 | 48 × 12 | 250 | 125 | 8 × 37 | 33 | 3/4 | 1/4 | 75–85 |
| 17 | 32 | 10 | 12 | 42 × 10 | 275 | 50 | 5 × 28 | 27 | 1/2 | 1/8 | 25–40 |
| 18 | 48 | 14 | 15 | 48 × 14 | 250 | 150 | 8 × 37 | 39 | 27/32 | 9/32 | 80–100 |
| 19 | 32 | 12 | 13 | 42 × 10 | 275 | 50 | 6 × 31 | 27 | 1/2 | 1/8 | 30–50 |
| 22 | 54 | 14 | 15 | 54 × 14 | 250 | 175 | 8 × 37 | 44 | 15/16 | 5/16 | 90–115 |

from 54 to 72 in. wide in the saw space, and single edgers range in width from 36 to 54 in. wide. Edgers are provided with pressure and feed rolls both before and behind the saws.

**Slashers.**—A slasher consists of a series of saws spaced at equal intervals on an arbor together with feed chains to convey waste material to the saws. Fig. 11 is a diagrammatic arrangement of a slasher. The chains *A* extend across the mill, as shown in Fig. 8, to convey waste from any set of live rolls. The short feeder chains *B* insure that the waste is squarely presented to the saw *C*. The discharge chains *D* normally lie below the plane of the roller conveyor, but when a piece is to be transferred to the slasher these chains are elevated to lift the material from the roller bed and discharge it to the slasher chain. Sawed slashings drop into the conveyor *E* which transfers them either to the lath and shingle mills, or to the fuel hog.

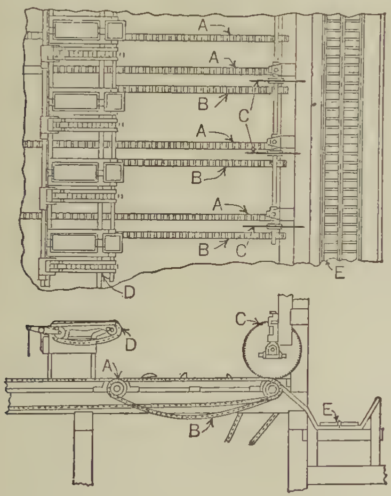
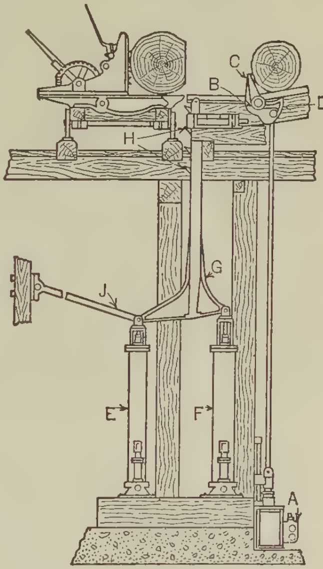


Fig. 11. Diagrammatic Arrangement of Slasher



*Filer & Stowell Co.*

Fig. 12. Nigger and Log Loader

**Trimming Saws.**—Trimming saws comprise a series of swing saws suspended over transfer chains somewhat similar to those used in the slasher. The saws are spaced at such intervals that any two of them will saw stock to commercial lengths. In the more modern types of trimming saws, compressed air is used to swing the saw into the path of the work. Each saw is individually controlled by one operator, who, as a piece of stock approaches the saws, depresses those two which are nearest to the ends of the stock. The trimming saw conveyor delivers the sawed stock to the sorting chain.

**Niggers and Log Loaders.**—Fig. 12 shows, diagrammatically, a steam nigger and log loader. The log loader comprises the steam cylinder *A* connected to the loader arm *B*. The nigger consists of the oscillating steam cylinders *E* and *F*, the head *G* carrying dog-teeth *H*, and the radius rod *J*. Steam to the cylinders is controlled by suitable valves, operated by the sawyer. The log is held at the edge of the log deck by the arm *C*. When steam is admitted to cylinder *A*, arm *C* is rotated downward and arm *D* kicks the log against the knee of the saw carriage. The head *G* of the nigger then is raised by the cylinders *E* and *F* until the teeth *H* engage the log. The direction of motion of the head is controlled by the combination of the radius rod *J* and the variable stroke possible in the two cylinders *E*, *F*, to hold the log in place until it is gripped by the carriage dogs or to turn it through any desired angle.

Log loader cylinders range from 8 to 18 in. diameter, with stroke varied to conform to log sizes. Nigger cylinders range from 8 to 14 in. diameter, with stroke of 5 to 6 ft.

**Manufacturers.**—Saw mill machinery is built by Allis-Chalmers Co., Milwaukee, Wis.; American Saw Mill Machinery Co., Hackettstown, N. J.; F. L. Estes Saw and Machinery Co., Nashville, Tenn.; Filer and Stowell Co., Milwaukee, Wis.; Gainesville Iron Works, Gainesville, Ga.; Standard Saw Mill Machinery Co., Erie, Pa.; Taylor Iron Works and Supply Co., Macon, Ga.; Wheland Co., Chattanooga, Tenn.; J. A. Vance Co., Winston-Salem, N. C.

**SAW MILL DRIVES.**—Saw mills are, in general, driven either from one or more line-shafts extending the length of the mill, underneath the machinery floor, or by electric motors driving machines individually or in groups. In line-shaft drives, the main shaft is belted to a steam engine or electric motor. Sometimes a rope drive is used to transmit power from the engine. Where the driven machine is at an angle to the line-shaft, mule pulley drives (see p. 24–26) or large bevel gears with wood teeth, or bevel friction gears (see p. 14–03) are used to change the direction of the power supply. Final drive to the individual machines usually is by belt. Line-shaft speeds are from 150 to 250 r.p.m. Line-shafts usually are carried in rigid post bearings. In the more modern mills, all bearings are force lubricated from a central point by an oil pump, reservoir, and oil filter.

Steam engines used for line-shaft drive are of various types. In Southern mills, Corliss engines are widely used. Slide valve engines and uniflow engines also are in use. Steam economy seldom is given consideration in saw mill practice, as wood waste is used for fuel, and usually there is more available than can be burned for steam making. However, it should be remembered that wasteful engines require a larger boiler plant, with consequently greater first cost and overhead and operating charges. Many mill engines exhaust direct to atmosphere and use live steam in the dry kilns, but this practice also requires greater investment in boilers, than where exhaust steam is used in the kilns. A few mills, which have ample water available, operate condensing, and use live steam in the kilns.

**Power Requirements.**—The power required to drive a saw mill with line-shafts varies widely with the type of equipment, the character of maintenance and other factors. A mill of the type shown in Fig. 8 usually would have installed engine capacity of about 1000 hp. Indicator cards from engines driving several mills of this type showed a maximum demand of 850 to 900 hp.

Where the mill is to be motor-driven, manufacturers of equipment will advise as the size of the motors to be installed. For line-shaft drive, the power must be estimated from previous performances of similar mills. The following figures were used in estimating the requirements of a double-band mill, similar to that shown in Fig. 8. They are given as a general guide only, and should be checked against local conditions. The figures are indicated horsepower except as otherwise stated. Deck saw, 30 Hp.; 8-ft. band saw, 225 Hp.; 48-in. gang saw, 225 Hp.; 72-in. double edger, 150 Hp.; slasher, 50 Hp.; trimming saw, 40 Hp.; chain transfer to gang saw, 5 Hp.; chain transfer to trimming saw, 15 Hp.; nigger, 40 boiler Hp.; shot-gun feed for carriage, 14 in., 120 boiler Hp., 12 in., 80 boiler Hp. The load factor generally is taken as 80%.

## Veneer Lathes and Slicers

The accuracy requirements of veneer cutting demand more careful log preparation than for lumber sawing. Logs for rotary cutting are cut to length on a *drag saw*, blade, circular or band. The short lengths, or *bolts*, are sorted by length and specie, and placed in vats for cooking or steaming. To produce sliced, half-round and sawn veneer, logs are sawn into flitches, partly to obtain face figure and partly to facilitate clamping on the machine bed. Every veneer mill producing these latter types requires a small band (band-saw) mill both for flitching and to salvage lumber from unsuitable and defective logs that will not make veneer. Flitching technique varies widely according to log shape and veneer character. Fig. 13 shows a suitable method of flitching oak logs, 24 to 30 in. diameter, for quartering. In crotches, stumps and burls, only experience and judgment can solve the widely varying individual problems of flitching. Flitches, except for the sawn veneer, go to the vats.

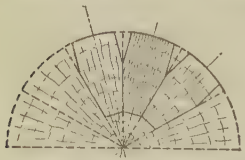


Fig. 13. Flitching for Quarter-sawed Veneer

**COOKING OR STEAMING.**—A few species of veneer woods, as basswood, poplar and similar woods, are best cut "cold," without cooking. All logs are improved for veneer cutting by soaking in a log pond. The best authorities advocate cooking logs in hot water. Temperature and duration of cooking vary widely with the species, size, texture and fresh-ness from the woods. A minimum is 4 to 6 hours at 200° F. up for coniferous woods, extending to 15 to 20 hours at 150° to 175° F. for hardwoods. Flitches of fragile figure are best soaked for longer periods at 100° to 125° F. Too much cooking produces "fuzzy" veneer. Logs are barked before mounting and should not be more than 30 minutes from vat to machine. Some operators prefer steaming, a shorter and more intense process that is too harsh for fragile woods, and one which does not properly penetrate others.

A cooking vat should be built of concrete liberally reinforced, especially at corners, to withstand severe temperature strains. It usually is made about 6 ft. deep by 10 ft. wide; the length is varied to suit capacity. Covers are preferably of steel plates with central rings, and are handled by the crane used for bolts and flitches.

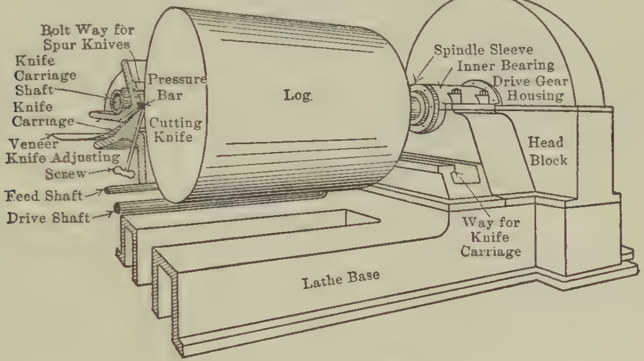


FIG. 14. Veneer Lathe

**VENEER LATHES.**—Fig. 14 shows a typical rotary veneer lathe with one end omitted. Details of the cutting mechanism are shown in Fig. 16. Logs are mounted in the lathe from the back by crane. Standard operating speed, traditionally, is 25 to 30 r.p.m. to accommodate 30-in. logs. This is slow for small logs. Spindle chucks or centers usually are 5- or 6-in. diameter for 60- to 100-in. bolts, leaving a 7-in. core, that is defective in many species, but which sometimes can be cut into crating. Lathes usually are equipped with 50- to 75-hp. motors for 60- to 100-in. lengths and logs up to 30-in. diameter. For larger logs, as on the Pacific Coast, power demand will be greater. An efficient set-up would be mechanical speed control permitting speeds to be changed without stopping lathes, giving 48, 36, 30 and 24 r.p.m., and permitting linear veneer delivery speeds of 100 (for thick veneer) to 200 ft. per min. Electrical speed controls are to be avoided, as motors incline to race under wide load variations, caused by thick and thin cuts, and large and small logs. Table 8 shows the resulting delivery speeds in such a set-up. Rotary box veneer often is cut at 60 r.p.m., but quality requirements are low. Fig. 17 shows a typical lathe lay out, with delivery tables, clipper, and crew location. In

**Table 8.—Actual Veneer Delivery Speeds, Feet per Minute**

| Log Diam., in. | R.p.m. of Spindle | | | | Log Diam., in. | R.p.m. of Spindle | | | | Log Diam., in. | R.p.m. of Spindle | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 24 | 30 | 36 | 48 | | 24 | 30 | 36 | 48 | | 24 | 30 |
| | Lineal ft. per min. of lathe delivery | | | | | Lineal ft. per min. of lathe delivery | | | | | Lineal ft. per min. of lathe delivery | |
| 8 | ...... | ...... | 75.4 | 100.5 | 19 | 119.4 | 149.2 | 179.1 | 238.8 | 30 | 188.5 | 235.6 |
| 9 | ...... | ...... | 84.8 | 113.1 | 20 | 125.7 | 157.1 | 188.5 | 251.3 | 31 | 194.8 | 243.5 |
| 10 | ...... | 78.5 | 94.2 | 125.7 | 21 | 131.9 | 164.9 | 197.9 | ...... | 32 | 201.1 | 251.3 |
| 11 | ...... | 86.4 | 103.7 | 138.2 | 22 | 138.2 | 172.8 | 207.3 | ...... | 33 | 207.3 | 259.2 |
| 12 | 75.4 | 94.2 | 113.1 | 150.8 | 23 | 144.5 | 180.6 | 216.7 | ...... | 34 | 213.6 | ...... |
| 13 | 81.7 | 102.1 | 122.5 | 163.4 | 24 | 150.8 | 188.5 | 226.2 | ...... | 35 | 219.9 | ...... |
| 14 | 88.0 | 110.0 | 131.9 | 175.9 | 25 | 157.1 | 196.4 | 235.6 | ...... | 36 | 226.2 | ...... |
| 15 | 94.2 | 117.8 | 141.4 | 188.5 | 26 | 163.4 | 204.2 | 245.0 | ...... | 37 | 232.5 | ...... |
| 16 | 100.5 | 125.7 | 150.8 | 201.1 | 27 | 169.6 | 212.1 | 254.5 | ...... | 38 | 238.8 | ...... |
| 17 | 106.8 | 133.5 | 160.2 | 213.6 | 28 | 175.9 | 219.9 | ...... | ...... | 39 | 245.0 | ...... |
| 18 | 113.1 | 141.4 | 169.6 | 226.2 | 29 | 182.2 | 227.8 | ...... | ...... | 40 | 251.3 | ...... |

species with sound cores, as birch, a supplementary high-speed (50 r.p.m.) lathe sometimes is provided with 3-in. chucks to cut $^1/_{20}$-in. veneer from 7- and 8-in. cores, 42 in. long or less.

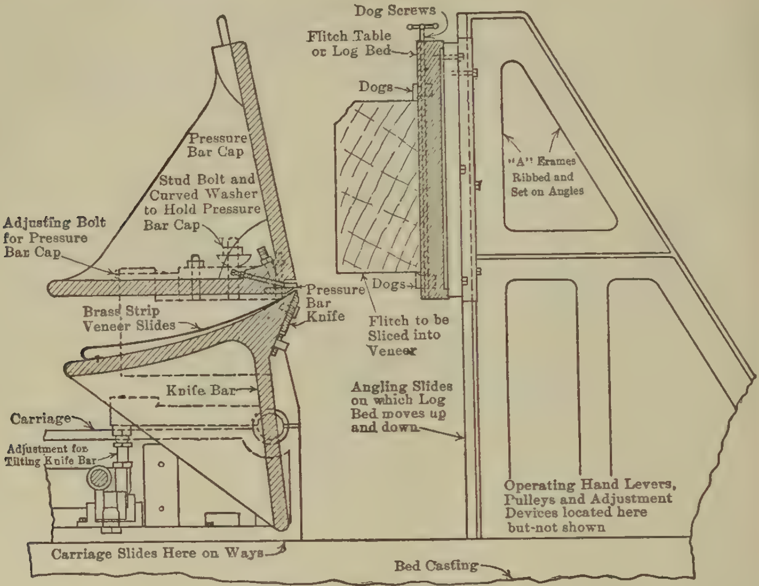

Dog Screws
Flitch Table or Log Bed
Dogs
"A" Frames Ribbed and Set on Angles
Pressure Bar Cap
Stud Bolt and Curved Washer to Hold Pressure Bar Cap
Adjusting Bolt for Pressure Bar Cap
Pressure Bar Knife
Dogs
Brass Strip Veneer Slides
Flitch to be Sliced into Veneer
Knife Bar
Angling Slides on which Log Bed moves up and down
Carriage
Adjustment for Tilting Knife Bar
Operating Hand Levers, Pulleys and Adjustment Devices located here but not shown
Carriage Slides Here on Ways
Bed Casting

FIG. 15. Veneer Slicer

**VENEER SLICER.**—Fig. 15 shows a typical veneer slicer. Flitches are mounted on the table or bed by dogs, the table moving up and down on angling ways to produce a shear cut. The cutting stroke is about 3 ft. long. The draw or lateral movement of the knife is about 1 1/2 ft. Cutting is done on the down stroke, and the log is fed forward on the return stroke of the knife. Horizontal slicers are used more in Europe than in the U.S., and afford greater latitude in range of angle cuts, often desirable in highly figured flitches. Veneer delivery equipment for slicers is simpler than for lathes. Flitch-cut veneer always is kept in sequence to permit matching.

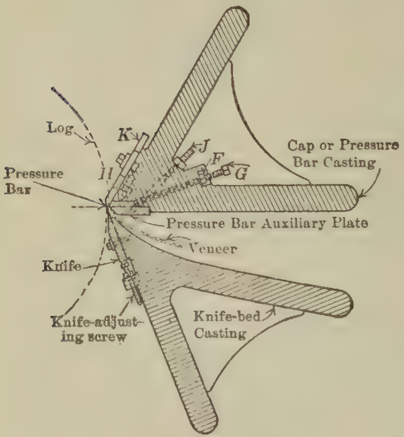


FIG. 16. Details of Veneer Lathe Cutting Mechanism

**HALF-ROUND VENEER** is an intermediate method between rotary and sliced veneer. The log is flitched or sawn, as in slicing, and cooked. It is cut on a lathe provided with a *stay-log* swung from the lathe centers, with radial adjustment and provided with a table or bed to which flitches are clamped with bolts. See Fig. 6. The purpose of half-round veneer is to obtain a veneer with more regular grain than that of rotary veneer.

**CUTTING MECHANISM**, Fig. 16, is essentially the same in both lathes and slicer Its proper adjustment and manipulation is the critical factor in good veneer cutting. The feed mechanism in both lathes and slicers is mounted on two screws, moving the lathe carriage with the cutting mechanism into the log or flitch a distance equal to the veneer thickness at each revolution or stroke.

Rotary veneers usually are cut to widths, providing whole, half, third and quarter parts of the desired sizes. Flitch veneers are kept together after numbering and taking several sample swatches from top, center and bottom of each flitch.

**VENEER DRYERS** comprise rollers, traveling plates or mesh belts, on which veneer passes progressively through drying chambers. Controlled temperature and humidity establish the necessary moisture deficit to dry various thicknesses at a single pass, the speed for thicker veneers being reduced. Veneer usually is dried to 10% moisture content. It must be dried within a few hours after cutting in order to prevent discoloration.

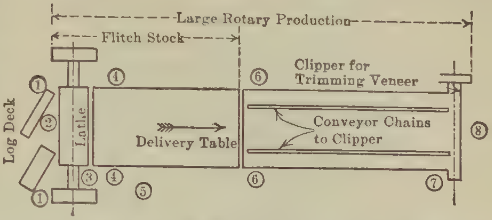


Fig. 17. Standard Veneer Layout and Crew. Standard Crew is: (1) 2 log peelers. (2) 1 log hanger. (3) 1 lathe operator. (4) 2 off bearers or breakers. (5) 1 waste collector. (6) 2 off bearers for large production. (7) 1 clipper operator. (8) 1 clipper off bearer.

## 3. PROCESSING

### Lumber Dry Kilns *

**EQUILIBRIUM MOISTURE CONTENT.**—Wood products, in service and storage, should have, as nearly as possible, an equilibrium moisture content in the wood parts, to keep at a minimum dimensional changes (shrinkage and swelling) due to moisture changes. Under ordinary room conditions this is about 6%–8% moisture content; in normal outdoor weather from 10%–15%. Lumber is stronger and stiffer when dry than when wet. Dry wood (to the extent mentioned above) will retain paint and finishes, whereas wet lumber will not hold surface coatings. Table 9 gives conditions for maintaining moisture content equilibrium in lumber. Correct atmospheric conditions are particularly important in raw lumber storage rooms.

**LUMBER DRY KILNS** are of two principal types. In the single charge (*box* or *compartment*) kiln, the lumber charge remains stationary during the drying cycle; in the *progressive* kiln the lumber moves in the kiln through the various stages in the drying cycle, from the initial low-temperature-high-humidity condition to the final high-temperature-low-humidity stage. Fig. 18 is a comparison of the two types.

**Table 9.—Variation of Equilibrium Moisture Content of Wood**

| Wet-bulb Depression, deg. F. | Dry-bulb Temperature, deg. F. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 | 100 |
| | Equilibrium Moisture Content, percent | | | | | | | | | | |
| 4 | 15.7 | 16.2 | 17.0 | 17.3 | 17.6 | 17.9 | 18.1 | 18.1 | 18.2 | 18.2 | 18.3 |
| 5 | 13.0 | 14.5 | 15.0 | 15.3 | 15.6 | 15.9 | 16.2 | 16.4 | 16.6 | 16.8 | 17.0 |
| 6 | 12.5 | 13.0 | 13.5 | 13.0 | 14.2 | 14.5 | 14.8 | 15.0 | 15.1 | 15.3 | 15.5 |
| 7 | 11.3 | 11.9 | 12.4 | 12.8 | 13.1 | 13.4 | 13.7 | 13.9 | 14.1 | 14.3 | 14.4 |
| 8 | 10.2 | 10.8 | 11.4 | 11.8 | 12.2 | 12.5 | 12.8 | 13.1 | 13.3 | 13.5 | 13.7 |
| 9 | 9.2 | 9.9 | 10.4 | 10.9 | 11.3 | 11.7 | 12.0 | 12.2 | 12.4 | 12.6 | 12.7 |
| 10 | 8.3 | 9.9 | 9.6 | 10.1 | 10.6 | 10.9 | 11.2 | 11.5 | 11.8 | 12.0 | 12.2 |
| 11 | 7.3 | 8.1 | 8.7 | 9.3 | 9.9 | 10.3 | 10.6 | 10.8 | 11.0 | 11.2 | 11.3 |
| 12 | 6.4 | 7.3 | 8.1 | 8.6 | 9.1 | 9.6 | 10.0 | 10.2 | 10.4 | 10.6 | 10.8 |
| 13 | 5.5 | 6.4 | 7.2 | 7.8 | 8.4 | 8.8 | 9.2 | 9.6 | 9.9 | 10.1 | 10.2 |
| 14 | 4.6 | 5.7 | 6.6 | 7.3 | 7.9 | 8.3 | 8.7 | 9.1 | 9.4 | 9.7 | 9.9 |
| 15 | ....... | 5.0 | 5.8 | 6.6 | 7.3 | 7.8 | 8.2 | 8.5 | 8.8 | 9.1 | 9.3 |
| 16 | ....... | ....... | 5.2 | 6.0 | 6.7 | 7.2 | 7.7 | 8.0 | 8.3 | 8.6 | 8.8 |
| 17 | ....... | ....... | ....... | 5.4 | 6.1 | 6.7 | 7.2 | 7.5 | 7.8 | 8.1 | 8.3 |
| 18 | ....... | ....... | ....... | 4.8 | 5.6 | 6.2 | 6.7 | 7.1 | 7.4 | 7.7 | 7.9 |
| 19 | ....... | ....... | ....... | ....... | 5.0 | 5.6 | 6.2 | 6.6 | 7.0 | 7.3 | 7.5 |
| 20 | ....... | ....... | ....... | ....... | ....... | 5.2 | 5.7 | 6.2 | 6.6 | 6.9 | 7.1 |
| 21 | ....... | ....... | ....... | ....... | ....... | ....... | 5.3 | 5.8 | 6.2 | 6.5 | 6.7 |
| 22 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | 5.3 | 5.8 | 6.1 | 6.3 |
| 23 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | 5.4 | 5.7 | 6.0 |
| 24 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | 5.1 | 5.4 | 5.7 |
| 25 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | 5.1 | 5.5 |

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* See also Kent's Mechanical Engineers' Handbook—Power, pp. 3–52 to 3–54, forming Vol. 2 of this series.

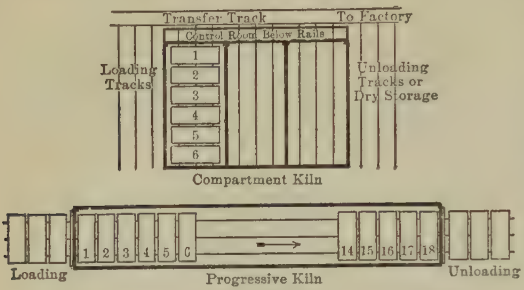


FIG. 18. Dry Kilns with Equal Capacity

**Drying Technique** is of three principal forms, shown in Fig. 19: 1. Natural circulation or *ventilated kilns*, where air movement is accomplished by gravity; 2. Forced circulation, or *fan kilns*, in which air is mechanically forced through the lumber (circulation frequently reversible); 3. *Condensing kilns*, where moisture-laden air passes over condensing coils (usually circulating cold water pipes), is reheated, and again passes through the lumber. Most kilns have steam or water sprays to maintain the humidity. Temperature and humidity controls, usually with recording attachments, are essential to maintain schedule requirements. In kiln drying programs, *humidity* refers to air conditions, and *moisture content* means the moisture contained in the lumber. The latter is computed by weight, weight of dry wood being divided into the weight of moisture removed.

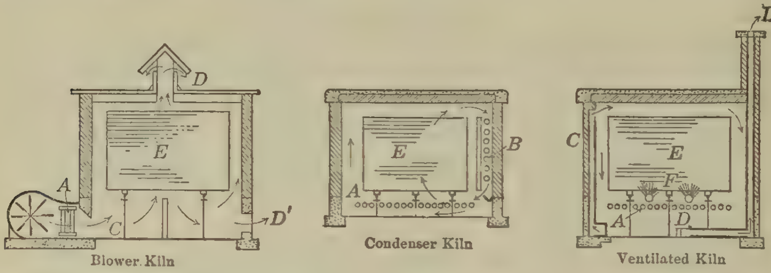


FIG. 19. Types of Kilns. *A*, heating unit. *B*, condensing unit. *C*, fresh air inlet. *D*, *D'*, used air outlets. *E*, lumber charge. *F*, humidifying unit

**Table 10.—Time Required to Dry Hardwoods in Ventilated Kilns to 5% Moisture Conten**

| Wood | 4/4 in. Thick | | | | 6/4 in. Thick | | | | 8/4 in. Thick | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Moisture Content of Wood Entering Dry Kiln, percent | | | | | | | | | | | |
| | 15–20 | 21–30 | 31–40 | 41–50 | 15–20 | 21–30 | 31–40 | 41–50 | 15–20 | 21–30 | 31–40 | 41–5 |
| | Days | | | | | | | | | | | |
| Hardwood: | | | | | | | | | | | | |
| Ash | 6 | 7 | 10 | 14 | 9 | 12 | 15 | 21 | 12 | 14 | 20 | 28 |
| Beech | 8 | 10 | 15 | 20 | 12 | 15 | 22 | 30 | 16 | 20 | 30 | 40 |
| Birch | 7 | 8 | 12 | 16 | 10 | 12 | 18 | 24 | 14 | 16 | 24 | 32 |
| Cherry | 8 | 10 | 15 | 20 | 12 | 15 | 22 | 30 | 16 | 20 | 30 | 40 |
| Chestnut | 6 | 7 | 10 | 14 | 9 | 12 | 15 | 21 | 12 | 14 | 20 | 28 |
| Elm, soft | 7 | 8 | 12 | 16 | 10 | 12 | 18 | 24 | 14 | 16 | 24 | 32 |
| " rock | 8 | 10 | 15 | 20 | 12 | 15 | 22 | 30 | 16 | 20 | 30 | 40 |
| Gum, red | 8 | 10 | 15 | 20 | 12 | 15 | 22 | 30 | 16 | 20 | 30 | 40 |
| " sap | 7 | 8 | 12 | 16 | 10 | 12 | 18 | 24 | 14 | 16 | 24 | 32 |
| " Tupelo | 6 | 7 | 10 | 14 | 9 | 12 | 15 | 21 | 12 | 14 | 20 | 28 |
| Hickory | 10 | 12 | 18 | 24 | 15 | 18 | 27 | 36 | 20 | 24 | 36 | 48 |
| Mahogany | 8 | 9 | 14 | 18 | 12 | 15 | 21 | 27 | 16 | 18 | 28 | 36 |
| Maple, hard | 7 | 8 | 12 | 16 | 10 | 12 | 18 | 24 | 14 | 16 | 24 | 32 |
| " soft | 6 | 7 | 10 | 14 | 9 | 12 | 15 | 21 | 12 | 14 | 20 | 28 |
| Oak | 8 | 10 | 15 | 20 | 12 | 15 | 22 | 30 | 16 | 20 | 30 | 40 |
| Sycamore | 8 | 10 | 15 | 20 | 12 | 15 | 22 | 30 | 16 | 20 | 30 | 40 |
| Walnut | 10 | 12 | 18 | 24 | 15 | 18 | 27 | 36 | 20 | 24 | 36 | 48 |
| Semi-hardwood: | | | | | | | | | | | | |
| Basswood | 5 | 6 | 7 | 8 | 8 | 9 | 10 | 12 | 10 | 12 | 14 | 16 |
| Cottonwood | 5 | 6 | 7 | 8 | 8 | 9 | 10 | 12 | 10 | 12 | 14 | 16 |
| Poplar | 5 | 6 | 7 | 8 | 8 | 9 | 10 | 12 | 10 | 12 | 14 | 16 |
| Willow | 5 | 6 | 7 | 8 | 8 | 9 | 10 | 12 | 10 | 12 | 14 | 16 |

In general, temperatures of 200° F. safely can be used on 1-in. (4/4) softwoods; for hardwoods the temperature range is 150° to 160° F. With thicker and greener lumber these temperatures must be reduced. Fig. 20 is a convenient chart for atmospheric humidity.

Example.—Determine the relative humidity of air when the dry-bulb temperature is 140° F., and the wet-bulb, 120° F. Enter the chart on the 120° wet-bulb line and follow it until it intersects the curve of 140° dry-bulb temperature. The ordinate through the intersection intersects the relative humidity scale at 54%.

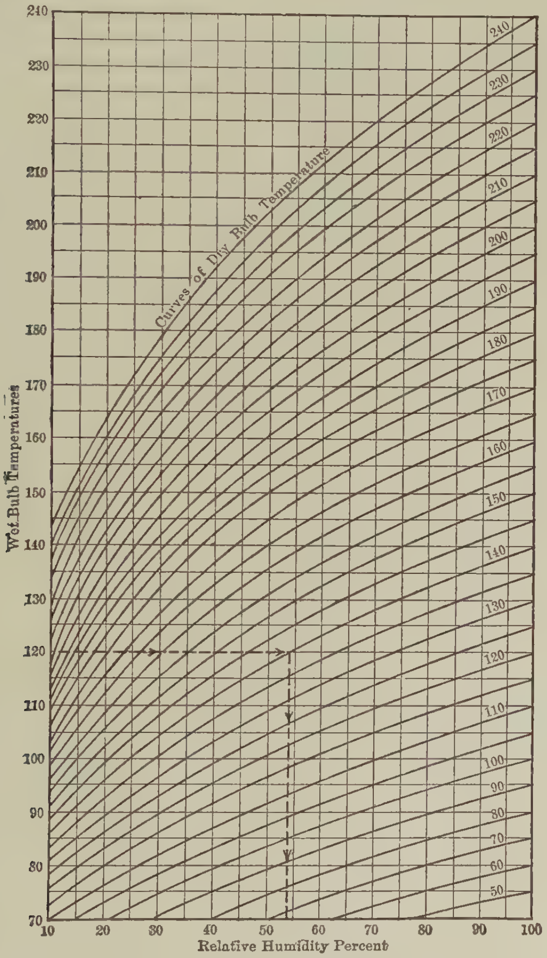


Fig. 20. Relative Humidity Chart

**TIME REQUIRED FOR KILN DRYING** of pine and similar softwoods usually is from 3 to 5 days, from the original tree condition of greenness to about 10% moisture content, usually considered shipping dry. Hardwoods require varying times, according to thickness, original and desired moisture content, and types of kilns available. A general survey of these drying times is given for ventilated kilns in Table 10.

It is highly important that lumber be uniformly kiln dried to the required moisture content, and be maintained at that point during the entire product manufacturing procedure. In fact, most kiln authorities advocate kiln drying slightly below required conditions, and allowing reabsorption to equilibrium point in dry storage.

**Tests for Moisture Content** usually are made by weighing samples. When tests are required, samples are cut from test boards whose locations in the kiln load are shown in

## Table 11.—Moisture Content of Veneer and Lumber

| Oven Dry Weight | Excess of Initial Weight over Oven Dry Weight | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 1/2 | 2 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 | 5 1/2 | 6 | 7 | 8 | 9 | 10 | 12 | 14 | 16 | 18 | 20 |
| | Moisture Content, percent | | | | | | | | | | | | | | | | | | |
| 41 | 3.7 | 4.9 | 6.1 | 7.3 | 8.5 | 9.8 | 11.0 | 12.2 | 13.4 | 14.6 | 17.1 | 19.5 | 22.0 | 24.4 | 29.3 | 34.1 | 39.0 | 43.9 | 48.8 |
| 42 | 3.6 | 4.8 | 6.0 | 7.1 | 8.3 | 9.5 | 10.7 | 11.9 | 13.1 | 14.3 | 16.7 | 19.0 | 21.4 | 23.8 | 28.6 | 33.3 | 38.1 | 42.9 | 47.6 |
| 43 | 3.5 | 4.7 | 5.8 | 7.0 | 8.1 | 9.3 | 10.5 | 11.6 | 12.8 | 14.0 | 16.3 | 18.6 | 20.9 | 23.3 | 27.9 | 32.6 | 37.2 | 41.9 | 46.5 |
| 44 | 3.4 | 4.5 | 5.7 | 6.8 | 8.0 | 9.1 | 10.2 | 11.4 | 12.5 | 13.6 | 15.9 | 18.2 | 20.5 | 22.7 | 27.3 | 31.8 | 36.4 | 40.9 | 45.5 |
| 45 | 3.3 | 4.4 | 5.6 | 6.7 | 7.8 | 8.9 | 10.0 | 11.1 | 12.2 | 13.3 | 15.6 | 17.8 | 20.0 | 22.2 | 26.7 | 31.1 | 35.6 | 40.0 | 44.4 |
| 46 | 3.3 | 4.3 | 5.4 | 6.5 | 7.6 | 8.7 | 9.9 | 10.9 | 12.0 | 13.0 | 15.2 | 17.4 | 19.6 | 21.7 | 26.1 | 30.4 | 34.8 | 39.1 | 43.5 |
| 47 | 3.2 | 4.3 | 5.3 | 6.4 | 7.4 | 8.5 | 9.5 | 10.6 | 11.7 | 12.8 | 14.9 | 17.0 | 19.2 | 21.3 | 25.5 | 29.8 | 34.0 | 38.3 | 42.6 |
| 48 | 3.1 | 4.2 | 5.2 | 6.2 | 7.3 | 8.3 | 9.4 | 10.4 | 11.5 | 12.5 | 14.6 | 16.7 | 18.7 | 20.8 | 25.0 | 29.2 | 33.3 | 37.5 | 41.7 |
| 49 | 3.1 | 4.1 | 5.1 | 6.1 | 7.1 | 8.2 | 9.2 | 10.2 | 11.2 | 12.2 | 14.3 | 16.3 | 18.4 | 20.4 | 24.5 | 28.6 | 32.7 | 36.7 | 40.8 |
| 50 | 3.0 | 4.0 | 5.0 | 6.0 | 7.0 | 8.0 | 9.0 | 10.0 | 11.0 | 12.0 | 14.0 | 16.0 | 18.0 | 20.0 | 24.0 | 28.0 | 32.0 | 36.0 | 40.0 |
| 51 | .... | 3.9 | 4.9 | 5.9 | 6.9 | 7.8 | 8.8 | 9.8 | 10.8 | 11.8 | 13.7 | 15.7 | 17.6 | 19.6 | 23.5 | 27.5 | 31.4 | 35.3 | 39.2 |
| 52 | .... | 3.8 | 4.8 | 5.8 | 6.7 | 7.7 | 8.7 | 9.6 | 10.6 | 11.5 | 13.5 | 15.4 | 17.3 | 19.2 | 23.1 | 26.9 | 30.8 | 34.6 | 38.5 |
| 53 | .... | 3.8 | 4.7 | 5.7 | 6.6 | 7.5 | 8.5 | 9.4 | 10.4 | 11.3 | 13.2 | 15.1 | 17.0 | 18.9 | 22.6 | 26.4 | 30.2 | 34.0 | 37.7 |
| 54 | .... | 3.7 | 4.6 | 5.6 | 6.5 | 7.5 | 8.3 | 9.3 | 10.2 | 11.1 | 13.0 | 14.8 | 16.7 | 18.5 | 22.2 | 25.9 | 29.6 | 33.3 | 37.0 |
| 55 | .... | 3.6 | 4.5 | 5.5 | 6.4 | 7.3 | 8.2 | 9.1 | 10.0 | 10.9 | 12.7 | 14.5 | 16.4 | 18.2 | 21.8 | 25.5 | 29.1 | 32.7 | 36.4 |
| 56 | .... | 3.6 | 4.5 | 5.4 | 6.3 | 7.1 | 8.0 | 8.9 | 9.8 | 10.7 | 12.5 | 14.3 | 16.1 | 17.9 | 21.4 | 25.0 | 28.6 | 32.1 | 35.7 |
| 57 | .... | 3.5 | 4.4 | 5.3 | 6.1 | 7.0 | 7.9 | 8.8 | 9.6 | 10.5 | 12.3 | 14.0 | 15.8 | 17.5 | 21.0 | 24.6 | 28.1 | 31.6 | 35.1 |
| 58 | .... | 3.4 | 4.3 | 5.2 | 6.0 | 6.9 | 7.8 | 8.6 | 9.5 | 10.3 | 12.1 | 13.8 | 15.5 | 17.2 | 20.7 | 24.1 | 27.6 | 31.0 | 34.5 |
| 59 | .... | 3.4 | 4.2 | 5.1 | 5.9 | 6.8 | 7.6 | 8.5 | 9.3 | 10.2 | 11.9 | 13.6 | 15.3 | 17.0 | 20.3 | 23.7 | 27.1 | 30.5 | 33.9 |
| 60 | .... | 3.3 | 4.2 | 5.0 | 5.8 | 6.7 | 7.5 | 8.3 | 9.2 | 10.0 | 11.7 | 13.3 | 15.0 | 16.7 | 20.0 | 23.3 | 26.7 | 30.0 | 33.3 |
| 61 | .... | 3.3 | 4.1 | 4.9 | 5.7 | 6.6 | 7.4 | 8.2 | 9.0 | 9.8 | 11.5 | 13.1 | 14.8 | 16.4 | 19.7 | 22.9 | 26.2 | 29.5 | 32.8 |
| 62 | .... | 3.2 | 4.0 | 4.8 | 5.6 | 6.5 | 7.3 | 8.1 | 8.8 | 9.7 | 11.3 | 12.9 | 14.5 | 16.1 | 19.4 | 22.6 | 25.8 | 29.0 | 32.3 |
| 63 | .... | 3.2 | 4.0 | 4.8 | 5.6 | 6.3 | 7.1 | 7.9 | 8.7 | 9.5 | 11.1 | 12.7 | 14.3 | 15.9 | 19.0 | 22.2 | 25.4 | 28.6 | 31.7 |
| 64 | .... | 3.1 | 3.9 | 4.7 | 5.5 | 6.3 | 7.0 | 7.8 | 8.6 | 9.4 | 10.9 | 12.5 | 14.1 | 15.6 | 18.8 | 21.9 | 25.0 | 28.1 | 31.3 |
| 65 | .... | 3.1 | 3.8 | 4.6 | 5.4 | 6.2 | 6.9 | 7.7 | 8.5 | 9.2 | 10.8 | 12.3 | 13.8 | 15.4 | 18.5 | 21.5 | 24.6 | 27.7 | 30.8 |
| 66 | .... | 3.0 | 3.8 | 4.5 | 5.3 | 6.1 | 6.8 | 7.6 | 8.3 | 9.1 | 10.6 | 12.1 | 13.6 | 15.2 | 18.2 | 21.2 | 24.2 | 27.3 | 30.3 |
| 67 | .... | 3.0 | 3.7 | 4.5 | 5.2 | 6.0 | 6.7 | 7.5 | 8.2 | 9.0 | 10.5 | 11.9 | 13.4 | 14.9 | 17.9 | 20.9 | 23.9 | 26.9 | 29.9 |
| 68 | .... | ... | 3.7 | 4.4 | 5.1 | 5.9 | 6.6 | 7.4 | 8.1 | 8.8 | 10.3 | 11.8 | 13.2 | 14.7 | 17.7 | 20.6 | 23.5 | 26.5 | 29.4 |
| 69 | .... | ... | 3.6 | 4.3 | 5.1 | 5.8 | 6.5 | 7.2 | 8.0 | 8.7 | 10.1 | 11.6 | 13.0 | 14.5 | 17.4 | 20.3 | 23.2 | 26.1 | 29.0 |
| 70 | .... | ... | 3.6 | 4.3 | 5.0 | 5.7 | 6.4 | 7.1 | 7.9 | 8.6 | 10.0 | 11.4 | 12.9 | 14.3 | 17.1 | 20.0 | 22.9 | 25.7 | 28.6 |
| 71 | .... | ... | 3.5 | 4.2 | 4.9 | 5.6 | 6.3 | 7.0 | 7.7 | 8.4 | 9.9 | 11.3 | 12.7 | 14.1 | 16.9 | 19.7 | 22.5 | 25.3 | 28.2 |
| 72 | .... | ... | 3.5 | 4.2 | 4.9 | 5.6 | 6.3 | 6.9 | 7.6 | 8.3 | 9.7 | 11.1 | 12.5 | 13.9 | 16.7 | 19.4 | 22.2 | 25.0 | 27.8 |
| 73 | .... | ... | 3.4 | 4.1 | 4.8 | 5.5 | 6.2 | 6.9 | 7.5 | 8.2 | 9.6 | 11.0 | 12.3 | 13.7 | 16.4 | 19.2 | 21.9 | 24.7 | 27.4 |
| 74 | .... | ... | 3.4 | 4.1 | 4.7 | 5.4 | 6.1 | 6.8 | 7.4 | 8.1 | 9 5 | 10.8 | 12.2 | 13.5 | 16.2 | 18.9 | 21.6 | 24.3 | 27.0 |
| 75 | .... | ... | 3.3 | 4.0 | 4.7 | 5.3 | 6.0 | 6.7 | 7.3 | 8.0 | 9.3 | 10.7 | 12.0 | 13.3 | 16.0 | 18.6 | 21.3 | 24.0 | 26.7 |
| 76 | .... | ... | 3.3 | 3.9 | 4.6 | 5.3 | 5.9 | 6.6 | 7.2 | 7.9 | 9.2 | 10.5 | 11.8 | 13.2 | 15.8 | 18.4 | 21.7 | 23.7 | 26.3 |
| 77 | .... | ... | 3.2 | 3.9 | 4.5 | 5.2 | 5.8 | 6.5 | 7.1 | 7.8 | 9.1 | 10.4 | 11.7 | 13.0 | 15.6 | 18.2 | 20.8 | 23.4 | 26.0 |
| 78 | .... | ... | 3.2 | 3.8 | 4.5 | 5.1 | 5.7 | 6.4 | 7.1 | 7.7 | 9.0 | 10.3 | 11.5 | 12.8 | 15.4 | 17.9 | 20.5 | 23.1 | 25.6 |
| 79 | .... | ... | 3.2 | 3.8 | 4.4 | 5.1 | 5.7 | 6.3 | 7.0 | 7.6 | 8.9 | 10.1 | 11.4 | 12.7 | 15.2 | 17.7 | 20.3 | 22.8 | 25.3 |
| 80 | .... | ... | 3.1 | 3.8 | 4.4 | 5.0 | 5.6 | 6.3 | 6.9 | 7.5 | 8.8 | 10.0 | 11.3 | 12.5 | 15.0 | 17.5 | 20.0 | 22.5 | 25.0 |
| 81 | .... | ... | 3.1 | 3.7 | 4.3 | 4.9 | 5.6 | 6.2 | 6.8 | 7.4 | 8.6 | 9.9 | 11.1 | 12.4 | 14.8 | 17.3 | 19.8 | 22.2 | 24.7 |
| 82 | .... | ... | 3.1 | 3.7 | 4.3 | 4.9 | 5.5 | 6.1 | 6.7 | 7.3 | 8.5 | 9.8 | 11.0 | 12.2 | 14.6 | 17.1 | 19.5 | 22.0 | 24.4 |
| 83 | .... | ... | 3.0 | 3.6 | 4.2 | 4.8 | 5.4 | 6.0 | 6.6 | 7.2 | 8.4 | 9.6 | 10.8 | 12.1 | 14.5 | 16.9 | 19.3 | 21.7 | 24.1 |
| 84 | .... | ... | 3.0 | 3.6 | 4.2 | 4.7 | 5.4 | 6.0 | 6.5 | 7.1 | 8.3 | 9.5 | 10.7 | 11.9 | 14.3 | 16.7 | 19.0 | 21.4 | 23.8 |
| 85 | .... | ... | 3.0 | 3.5 | 4.1 | 4.7 | 5.3 | 5.9 | 6.5 | 7.1 | 8.2 | 9.4 | 10.6 | 11.8 | 14.1 | 16.5 | 18.8 | 21.2 | 23.5 |
| 86 | .... | ... | .... | 3.5 | 4.1 | 4.7 | 5.2 | 5.8 | 6.4 | 7.0 | 8.1 | 9.3 | 10.5 | 11.6 | 14.0 | 16.3 | 18.6 | 20.9 | 23.3 |
| 87 | .... | ... | .... | 3.4 | 4.0 | 4.6 | 5.2 | 5.7 | 6.3 | 6.9 | 8.0 | 9.2 | 10.3 | 11.5 | 13.8 | 16.1 | 18.4 | 20.7 | 23.0 |
| 88 | .... | ... | .... | 3.4 | 4.0 | 4.5 | 5.1 | 5.7 | 6.2 | 6.8 | 8.0 | 9.1 | 10.2 | 11.4 | 13.6 | 15.9 | 18.2 | 20.4 | 22.7 |
| 89 | .... | ... | .... | 3.4 | 3.9 | 4.5 | 5.1 | 5.6 | 6.2 | 6.7 | 7.9 | 9.0 | 10.1 | 11.2 | 13.5 | 15.7 | 18.0 | 20.2 | 22.5 |
| 90 | .... | ... | .... | 3.3 | 3.9 | 4.4 | 5.0 | 5.6 | 6.1 | 6.7 | 7.8 | 8.9 | 10.0 | 11.1 | 13.3 | 15.6 | 17.8 | 20.0 | 22.2 |
| 91 | .... | ... | .... | 3.3 | 3.8 | 4.4 | 4.9 | 5.5 | 6.0 | 6.6 | 7.7 | 8.8 | 9.9 | 11.0 | 13.2 | 15.4 | 17.6 | 19.8 | 22.0 |
| 92 | .... | ... | .... | 3.3 | 3.8 | 4.3 | 4.9 | 4.4 | 6.0 | 6.5 | 7.6 | 8.7 | 9.8 | 10.9 | 13.0 | 15.2 | 17.4 | 19.6 | 21.7 |
| 93 | .... | ... | .... | 3.2 | 3.8 | 4.3 | 4.8 | 5.4 | 5.9 | 6.5 | 7.5 | 8.6 | 9.7 | 10.8 | 12.9 | 15.1 | 17.2 | 19.4 | 21.5 |
| 94 | .... | ... | .... | 3.2 | 3.7 | 4.3 | 4.8 | 5.3 | 5.9 | 6.4 | 7.4 | 8.5 | 9.6 | 10.6 | 12.8 | 14.9 | 17.0 | 19.2 | 21.3 |
| 95 | .... | ... | .... | 3.2 | 3.7 | 4.2 | 4.7 | 5.3 | 5.8 | 6.3 | 7.4 | 8.4 | 9.5 | 10.5 | 12.6 | 14.7 | 16.8 | 19.0 | 21.1 |
| 96 | .... | ... | .... | 3.1 | 3.6 | 4.2 | 4.7 | 5.2 | 5.7 | 6.3 | 7.3 | 8.3 | 9.4 | 10.4 | 12.5 | 14.6 | 16.7 | 18.8 | 20.8 |
| 97 | .... | ... | .... | 3.1 | 3.6 | 4.1 | 4.6 | 5.2 | 5.7 | 6.2 | 7.2 | 8.2 | 9.3 | 10.3 | 12.4 | 14.4 | 16.5 | 18.6 | 20.6 |
| 98 | .... | ... | .... | 3.1 | 3.6 | 4.1 | 4.6 | 5.1 | 5.6 | 6.1 | 7.1 | 8.2 | 9.2 | 10.2 | 12.2 | 14.3 | 16.3 | 18.4 | 20.4 |
| 99 | .... | ... | .... | 3.0 | 3.5 | 4.0 | 4.5 | 5.1 | 5.6 | 6.1 | 7.1 | 8.1 | 9.1 | 10.1 | 12.1 | 14.1 | 16.2 | 18.2 | 20.3 |
| 100 | .... | ... | .... | 3.0 | 3.5 | 4.0 | 4.5 | 5.0 | 5.5 | 6.0 | 7.0 | 8.0 | 9.0 | 10.0 | 12.0 | 14.0 | 16.0 | 18.0 | 20.0 |

Fig. 21. The locations of the test samples in the test boards are shown in Fig. 22. The board ends, which probably are over-dried, are discarded. Fig. 23 shows the best procedure for marking weighed samples for filing, the figures on the left being moisture content and those on the right shrinkage. In both cases the upper figures are values immediately after cutting test samples; the lower figures are values after oven drying to minimum obtainable conditions, *i.e.*, bone dry. Fig. 24 illustrates a test sample prepared to show case-hardening. If the prongs flare out, the sample is case-hardened but dry; if they pinch, it was case-hardened before drying, and the stresses were not relieved during drying. A further test for exterior and interior moisture content is shown in Fig. 25. The test sample is band sawn, and outer and inner portions weighed before and after oven drying to determine the moisture content of each. A normal moisture content gradient usually will show an excess of 1% to 2% for the inner section.

Table 11 gives data for determining moisture content of veneer and lumber.

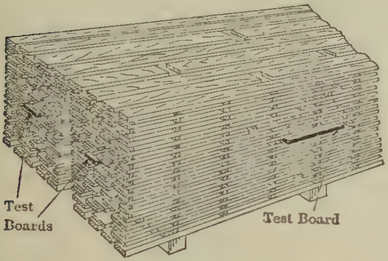


**Fig. 21.** Kiln Stack, Showing Locations of Test Boards

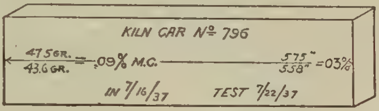


Fig. 23. Method of Marking Test Samples

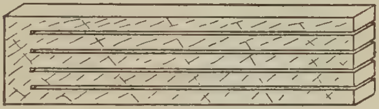


Fig. 24. Test Sample for Case-hardening

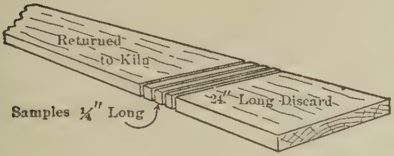


Fig. 22. Location of Sample in Test Boards

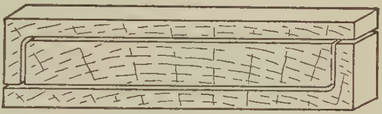


Fig. 25. Test Sample to Determine Dryness between Center and Outside

**Kiln Accessories.**—For kilns and adjoining storage tracks, 35-lb. rail generally is used, supported at 6-ft. intervals. Kiln bunks or trucks should have roller bearings, frequently lubricated because of the heat. Kiln transfer cars will facilitate lumber distribution from railroad unloading siding direct to kilns, and from kilns to dry storage and cutting rooms. They are usually about 12 in. high, rail top to rail top, and preferably are motor driven. Lumber lifts elevated by motor-driven revolving screws, in a pit below, are essential both for loading and unloading kiln cars, as well as for keeping the top of the lumber load at convenient height for serving rough planers or cut-off saws. Some mills use locomotive cranes with 40-ft. booms to transfer lumber in packets to and from kilns and lumber piles in the yard.

## Machine Operations and Lay Outs

In wood products factories, machine and department arrangement are substantially similar in major items, with considerable divergence in minor details, because of difference in character of products, and degree of pre-manufacture in raw materials. A typical example of standard procedure is given in Table 12.

For further details of plywood and finishing, see below. For kiln drying, see p. 22–17. For details of operation of the other types of wood-working machines, see pp. 22–39 to 22–48.

## Plywood and Gluing

**PLYWOOD.**—Standard plywood consists of an odd number of layers of veneer, with opposed grain in alternate layers, rigidly glued together. The along-the-grain strength reinforces the across-the-grain weakness, and distributes wood strength in both directions. The central layer (core), of veneer or lumber, has the lengthwise grain in the direction requiring strength. In 5-ply, face and core grains are parallel; in 3-ply, they are opposed.

**Table 12.—Standard Sequence of Machine Operations in Woodworking**

Lumber yards
↓
Dry kilns
↓
Dry storage and tempering
↓
Transfers and lumber lifts
↓
Rough planing (sometimes omitted)
↓
Cut-off or stock saws
↓
Rip saws
↓
Edge jointers
↓
Revolving glue clamps (for solid lumber parts or plywood cores)
↓
Finish planing to net thickness

| (Plywood parts)* | (Solid parts) |
|---|---|
| Veneer preparation | Dimensioning, *i.e.*, machining to size |
| Glue room | Moulding |
| Plywood redrying | Shaping |
| Dimensioning | Turning |
| Bending and curving | Routing |
| Routing | Special machine work |
| Moulding edges | |

↓
Boring and drilling
↓
Corner construction
↓
Sanding
↓
Stock room for machined parts
↓
Cabinet assembly
↓
Polish sanding
↓
Finishing (stain, filler, lacquer and rubbing)
↓
Trimming with hardware and glass
↓
Packing and crating

Inspection stations according to product requirements

* Omit plywood operations if plywood is purchased

Fundamentally the layers should be balanced on the core to avoid uneven stresses. See Figs. 26 and 27. The face wood is chosen for appearance; the other layers supply the strength. For most purposes $^{1}/_{4}$-in., 5-ply plywood is stronger than $^{13}/_{16}$-in. solid lumber. Plywood is unsplitable, and has more nail and screw holding power than solid wood.

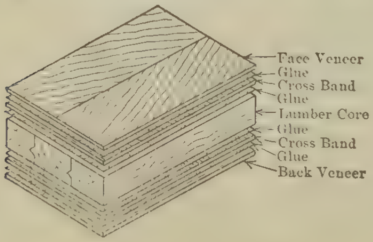


Fig. 26. 5-ply Plywood

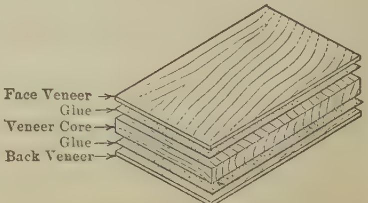
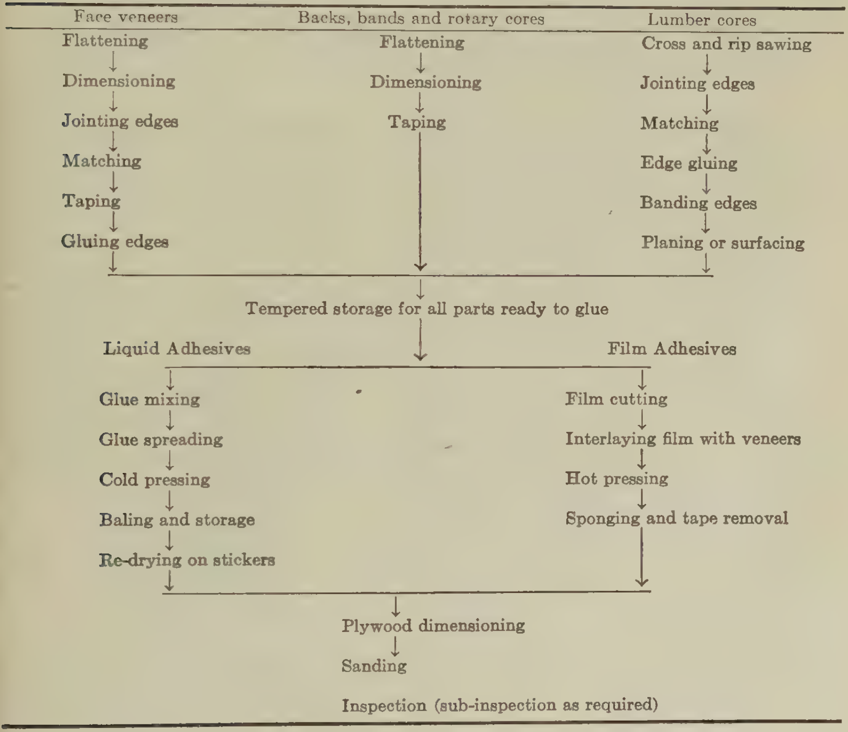


Fig. 27. 3-ply Plywood

**ARRANGEMENT OF EQUIPMENT.**—Plywood manufacturing falls in two general classes: 1. Factories assembling plywood as a semi-raw material for others; 2. Factories making the ultimate product, as furniture, with an integral plywood department, in which plywood operations are seldom complete. Plywood factories preferably are located with reference to the supply of the bulkiest material, *i.e.*, rotary commercial veneer and core lumber, the face veneer supply being far less important. Propinquity to a consumer furniture area is a doubtful advantage, because of the higher wage levels.

Table 13 shows the general sequence of plywood operations. The face veneer preparation, commercial veneer preparation and lumber core stock converge to a central tempered-air stock-room, where the gluing allocation for each day must be ready not later than the night before. The glue room is invariably the bottle-neck, has a higher man-hour-ratio per machine than any other plywood operation, and must produce at maximum efficiency.

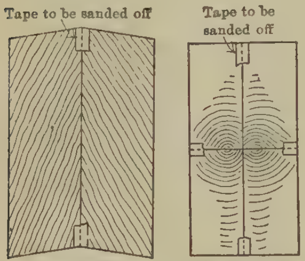
**Table 13.—General Arrangement of Plywood Operations**



**FACE VENEER.**—Sliced, half-round and sawn face veneers are received in flitches, and their suitability for the job in question demands experienced judgment. Unused portions of flitches (lay-backs) are a serious problem, often because of wrong size or figure. In general, whole flitches should be charged to a job, leaving lay-backs as a by-product; otherwise costs may be seriously distorted.

Highly-figured face veneers, if "crinkly," must be flattened before dimensioning. For less fragile species, platen veneer driers are used; it is advisable to dampen fragile veneers and apply gradual pressure with heated boards; in extreme cases, as feathered crotches, veneers are dipped in a dilute glue size to make them more leathery, and are flattened even more gradually. Moisture content should be 5% to 6% after flattening.

Longwood veneers, longer flitches in moderate figure, usually are cross cut on saws in packages of 25 sheets, and edge-clipped (sheared) with reference to figure matching. Clipped edges are *jointed* (see Veneer Jointers, p. 22–43), planed straight and square, to produce tight joints without cracks. *Stumps* (butts), *crotches* and *burls* and other highly-figured veneers usually are cut both ways in packages on a paper cutting machine, with an angling shear cut under pressure. Such veneers cannot be jointed as the edges wil tear out.



*a* Two piece Center Matched
*b* Four piece End and Side Matched

Fig. 28. Matched Veneer

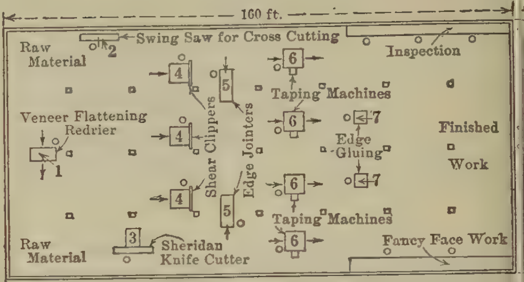
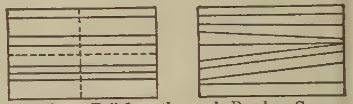


Fig. 29. Arrangement of Face Veneer Department. O indicates position of operator. Numerals show sequence o operations

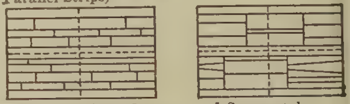
**Book-matched Veneer** comprises two or four adjacent sheets of a flitch, with uniformly (as to grain figure) trimmed edges, laid together to balance the figure as in Fig. 28. It resembles an unfolded 2- or 4-sheet booklet.

**Dimensioned Veneers** (longwood or book-matched) are *taped* (see Fig. 28) with gummed tape from edge to edge on one surface to hold the pieces in position during gluing. The tape is sanded off later. Taping machines have angling rollers that draw the edges together and press the moistened gummed tape over the adjoining edges. Intricate *combination faces*, of vari-colored species, with or without angling grains, give remarkable effects. Fig. 29 shows general arrangement of machine operations in a face veneer department.

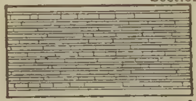
**CROSS-BANDS, ROTARY CORES AND BACKS,** unless two faces are required, are of commercial rotary veneer, flattened, edge-clipped, sometimes jointed if 3-ply, and taped as in face veneer, but without regard to figure. Cross-bands under faces, as well as cores in 3-ply plywood, should be one-piece and not taped, as tape will show through. Back cross-bands, backs and 5-ply cores may be taped. Commercial veneers, for long runs of standard sizes often are dimensioned at the veneer mill into whole, half, one-third, two-third, one-quarter and three-quarter sizes for taping. Moisture content must be kept close to 5%, except that certai layers in film bonding should be 8% to 12%. A commercial veneer department arrangement would be similar to Fig. 29, omitting the swin saw, Sheridan cutter and edge gluing.



*a.* Plain Core (Full Length Parallel Strips)
*b.* Random Core

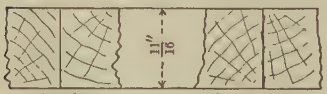


*c.* Butt Joint Strip Core
*d.* Segmental or Sectional Core

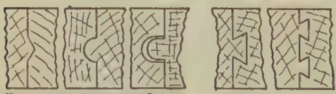


*e.* Block Core for Desk Bed

Fig. 30. Types of Cores



Planed or "Jointer" Joint
Sawed Joint without Planing



Tongue and Groove Joints
Single Double Dovetail Joints

Fig. 31. Types of Edge Joints in Plywood Cores

**LUMBER CORES** generally are used for plywood ½ in. and thicker. Five-ply construction i the more stable; face and back veneers in 3-pl usually are thin and do not effectively balance th lumber core. Core lumber is preferably of popla chestnut, gum, or other available species. Only on species should be used in a core. Chestnut doe not give a sound edge for molding or shapin To avoid warp, cores preferably should be mad of strips 2 to 3 in. wide. End joints are not objectionable and tend to eliminate twisting stresse Fig. 30 shows various core constructions, and Fig. 3 various core joints. The dotted lines in Fig. 3 indicate multiple cores to be cut apart after planin or gluing, to avoid showing endwise splices. The veneer strips used as glue carriers in co e are patented.

Fig. 32 shows the arrangement of a core department. Lumber is most economically handled on kiln cars from dry storage and placed on a lumber lift; top of load then always is at machine table height. If, before cross cutting, both sides are rough planed to reveal defects, a 2-side rough planer is placed adjacent to the lift, with the swing cut-off saws next in line. The stock then goes from cross-cut to rip-saws (see pp. 22–35 and 22–39) for trimming edges and removing wane (tapered and bark edges). If specifications are not too exacting, ripped edges may be glued together without further finishing operations, but quite frequently edges must be jointed true, straight and smooth for gluing. The sliding matching table, operation 5, is used for the purpose of arranging core strips to distribute stresses, and to rip the last strip to net core width.

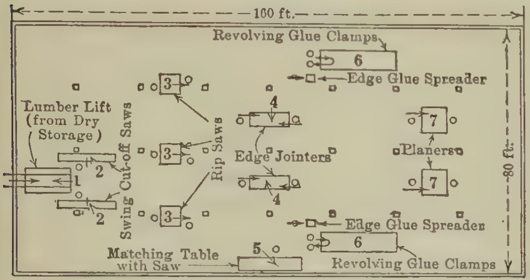
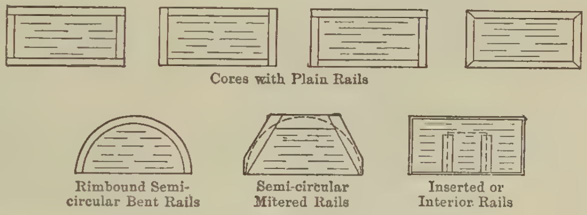


Fig. 32. Arrangement of Core Department. O indicates position of operator. Numerals show sequence of operations.

Assembled core strips are glued edge to edge in an endless chain revolving clamp, with a cycle of about 30 minutes. When the glued core returns to the loading point, it is removed and a fresh assembly inserted. The application of edge glue swells the lumber, increasing thickness at the joint, and glued cores should be seasoned, or equalized in moisture content, before planing.


Cores with Plain Rails

Rimbound Semi-circular Bent Rails

Semi-circular Mitered Rails

Inserted or Interior Rails

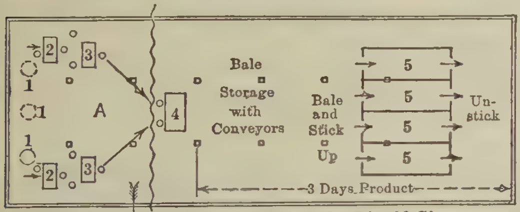
Fig. 33. Banded or Railed Cores

**Banded or Railed Cores** (Fig. 33) have a band or rail, of the same species as the face veneer, on one or more edges, to give molded edges the same appearance as the upper surface. These bands are additional core strips, glued on edges or ends, after the ends are trimmed.

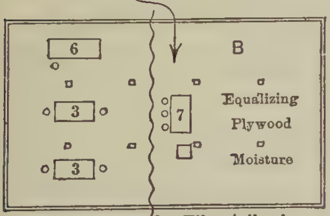
All plywood elements, faces, bands, backs and cores for an order should meet in an air-conditioned central stock room to avoid subsequent glue room delays.

**PLYWOOD GLUING DEPARTMENTS** are of two distinct types: 1. Adhesives are mechanically mixed and spread in *liquid* form as layers are assembled. 2. Adhesive films are merely interleaved between the layers of veneers. The two types of department are shown in Fig. 34.

**Liquid Glues** are: 1. *Animal* (bone and hide), prepared and spread hot. Seldom used because of cost. 2. *Vegetable* (casava or tapioca flour), cooked with caustic during mixing, and spread cold; used for cabinets and furniture, where water resistance is not required. 3. *Casein* (self-soured or acid-soured curds of milk), mixed, and spread cold with 5% to 15% of hydrated lime; water resistance increases with lime content, but similarly adds abrasive qualities that dull tools. 4. *Soya bean* (flour from soya bean residue after oil extraction), mixed with caustic, and spread cold; used



Arrangement for Liquid Glue with Cold Pressing



Arrangement for Film Adhesive with Hot Pressing

Fig. 34. Plywood Gluing Department. O indicates position of operator. Numerals show sequence of operations

chiefly for box shooks and cheaper plywood. Some minor use is made of *liquid glass* (silica) and *fish glues*. Resin adhesives are a comparatively recent development which are coming into extensive use. They are more fully described below.

*Glue Mixers*, except for animal glue which is heated without mixing, are steam- and water-jacketed, and have revolving double beaters. They usually are located above the spreaders and discharge into them.

*Glue Spreaders* usually coat both sides of veneer at one operation; they have corrugated steel rollers provided with scrapers (doctor bars or rollers), with glue reservoirs for both rolls. The spread veneer is *laid-up* with alternate layers of unspread veneer or lumber, until a 3-ply or 5-ply unit is complete. Assembled units often are separated, singly or in pairs, by *cauls* (waxed plywood or fiber sheets).

When as thick a *bale* or *package* has been accumulated, as will enter the daylight of the hydraulic press, the charge is pressed with 75 to 100 lb. per sq. in. of plywood area. While still under pressure, I-beams and clamp bolts are attached to maintain pressure until the initial edge glue set has taken place, usually from 4 to 12 hours. Clamped bales weigh 500 lb. and more, and are best handled by overhead conveyors. They usually are stored over night and opened the following morning.

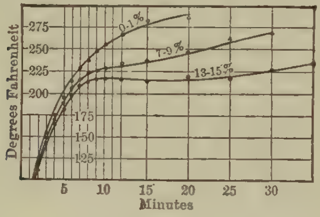


FIG. 35. Temperatures Attained with Veneers of Different Moisture Content During Different Pressure Periods. Platen temperature (P.T.), 300° F. Core thickness, 1/4 in. Specific pressure, 150–200 lb. Penetration depth, 3/8 in.

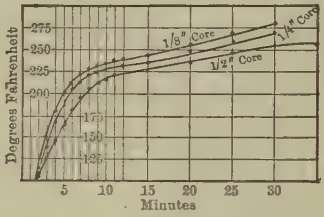


FIG. 36. Temperature Attained with Various Core Thicknesses During Different Pressure Periods. Platen temperature (P.T.), 300° F. Veneer moisture content 7–9%. Specific pressure, 150–200 lb. Penetration depth, 3/8 in.

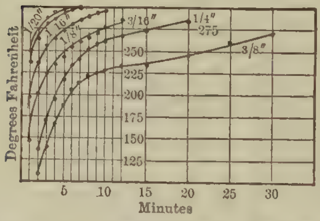


FIG. 37. Temperatures Attained with Various Penetration Depths During Different Pressure Periods. Platen temperature (P.T.), 300° F. Veneer moisture content, 7–9%. Specific pressure, 150–200 lb. Core thickness, 1/4 in.

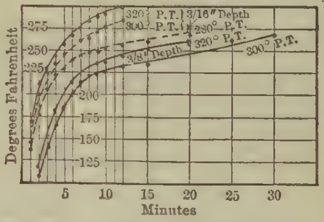


FIG. 38. Temperatures Attained with Various Platen Temperatures and Penetration Depths during Different Pressure Periods. Specific pressure, 150–200 lb. Veneer moisture content, 7–9%. Core thickness, 1/4 in. P.T. = platen temperature.

The resulting plywood has excessive moisture content, due to glue water. It should be *re-dried* in an air-conditioned room at 100° to 120° F., and 70% to 50% humidity for about two days. Re-drying plywood should be piled with alternate layers of uniform stickers to give full surface exposure for easy moisture removal, and well weighted down during re-drying to prevent warping. Cheaper grades of plywood, as plywood box shooks, where gradual evaporation of moisture content and consequent warping are unimportant, need not be re-dried.

**Resin-film Adhesives** are thin sheets of tissue paper, coated on both sides with synthetic resins. They have been successfully used in Europe for 10 years and made in the U. S. since 1935. Typical and most used of these, Tego, is a phenol formaldehyde resin. It comes in continuous rolls, is cut to veneer sizes, interleaved with veneer on a *lay-up* table, and sometimes held in position by gummed tape or wire staples. It is immediately ready for pressing, requiring simultaneous heat and pressure for setting or *polimerization*. Setting of the resin film makes the resulting joint insoluble in water, and produces the strongest plywood bond known, such as is required for airplanes and other uses exposed to weather. Required temperatures are from 280° to 300° F. Specific pressures on the plywood range from 150 lb. per sq. in. up, depending on the compressibility of the wood species. Hot presses are hydraulically operated and equipped with from 6 to 11 steam (occasionally electrically) heated platens. Each hot press opening normally will take one layer of assembled plywood; but it often is practicable to insert two or more layers, up to $^1/_8$ in. each. The time required for heat penetration usually precludes more than $^3/_8$ in. total thickness in each press opening. Plywood is conveniently placed between sheet aluminum *cauls* (No. 16 gage) to facilitate press loading. Alternate veneer layers, usually the crossbands, should have 8% to 12% moisture content to aid in the initial flow of the resin as it becomes heated. Further heat and pressure give the final set. Figs. 35 to 38 show the speed and extent of heat penetration to the furthest adhesive line to be polimerized. It is important that the effect of the variables be carefully observed to obtain the best results in film bonding. After removal from the press, plywood only requires cooling under slight pressure, to be ready to dimension. Some authorities prefer to cool and restore normal moisture content in the plywood by dipping or spraying with water. See Fig. 34.

**Liquid Resin Adhesives** are (1937) coming into accepted use. They are mixed and spread as described for liquid glues, and bonded in hot presses. While having a urea formaldehyde base, lacking the full waterproofness of film adhesives, and requiring more processing than films, they can be extended by admixtures, and thus are considerably cheaper than films. A departmental lay-out can be had by adding section A of Fig. 34 to section B, following the arrow.

**PLYWOOD DIMENSIONING** preferably is done on double cross-cut and double rip-saws of the double-end tennoner type (see p. 22–46), in which a pair of endless chains, with lugs, carry the plywood through the saws. One saw head is fixed and the other adjustable to the size required, usually up to 72 in. Many other types of saws and saw tables may be used for this work. Ripping usually precedes cross-cutting. For center matched plywood faces, pointers on the machine are used to insure proper centering of face veneer joints.

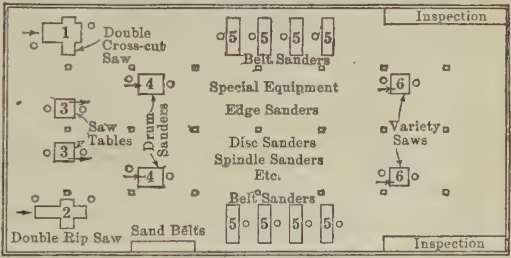


Fig. 39. Arrangement of Dimensioning and Sanding Department. O indicates position of operator. Numerals show sequence of operations

**Sanding.**—Plywood is sanded to remove the veneer tape and to prepare surfaces for filling, staining, and lacquering. Usual operations are: 1. Preliminary sanding of the back side of the plywood on a roll or endless belt sander, to clean it and remove most of the tape; other back sanding seldom is necessary; 2. Similar sanding on the face, taking care that high spots do not sand through; 3. Finish sanding on belt or stroke sanders, with very fine sandpaper. Some authorities advocate doing all sanding of plywood with highly-figured face on belt or stroke sanders, to prevent the waste of sanding-through by careful operation. Fig. 39 shows a typical arrangement of a dimensioning and sanding department.

**Re-dimensioning** sometimes is practiced when plywood is glued and sanded in multiples, as with small parts, where separate handling would greatly increase costs. See Fig. 30.

**Inspection.**—The chief inspector usually is stationed between the sanding and crating departments, with sub-inspectors only in the face veneer and lumber core departments.

## Finishing

**FINISHING.**—The term *finishing* generally includes polish *sanding*, *filling*, *staining*, *lacquering*, *waxing*, etc. The preliminary sanding operations (see p. 22–22) precede the cabinet assembly stage. Final *polish sanding*, either by hand or by mechanical devices, after cabinet assembly, should leave a very smooth surface which will not place an undue burden on the filler. Many factories sponge before polish sanding to raise the grain, and cause the slender sander shreds to become more or less erect for removal in sanding. Since sponging tends to clean out the wood pores, they must be filled even with the surface, especially in open-grain woods like oak. The *filler*, silex and other ingredients, usually is a paste, diluted with naphtha. It is brushed on, allowed to dry partially, and rubbed into the pores with a circular motion. The surplus is wiped off, without removing any filler below the finished surface. Defective filling is indicated by pock-marks, when too much filler has been wiped off, or the depressions are too open for the filler to stick. Correct results should follow the first filling; deficiencies may be remedied by repeating the process. In modern finishing technique, filler replaces the many coats of varnish formerly used.

**Stain.**—After filling, stain is applied (this sequence is sometimes reversed), usually with a spray gun, to obtain the desired color. The color may be that to which the species will normally mellow from air exposure, or may be a purely artificial color. Stain and filler make end wood appear darker than side wood, because of the volume of penetration. Care must be taken to limit the amount of stain on colored areas. Stains are distinguished by the solvent as: 1. *Water stains*, erroneously called acid stains. They are the most permanent in color, but darken a trifle as the wood ages. They raise the grain more than other stains. 2. *Spirit stains*, dissolved in alcohol. They dry more quickly, fade easily, but do not raise the grain. 3. *Oil stains*, which are not extensively used in high grade furniture, because they " bleed " with the solvents used in later lacquer applications. In cheaper finishes, filler and stain are combined and applied in one operation; results are inferior. Water stain and aniline dyes, made from coal tar, are preferred for best all-around results.

**Varnish**, formerly a widely used final finish, has largely given place to lacquers, which have much shorter drying cycles.

**Lacquer.**—Many clear lacquers, of similar characteristics, gradually are developing into high-solid, quick-drying coatings. They perform more satisfactorily than varnish in drying ovens, and can be varied in constituents to meet the conditions of exacting service. A large number of them are made from a synthetic resin base, and many are formulated to withstand, without discoloration or blister, water, heat and alcoholic solutions. The number of coats applied varies, in general, with the degree of permanent finish desired. Consult the literature of the well-known makers of finishing materials for further details.

**Wax** is one of the oldest finishes known. It is applied as a paste over about two coats of shellac. The shellac is applied on properly stained wood, which then usually is sanded. After the wax is dry, it is rubbed to a dull polish. If rewaxed occasionally, an excellent and easily maintained finish is obtained.

**Other Finishing Materials** are *colored lacquers* and *enamels*, and various trade-marked products, some of which are of excellent quality. Their application is similar to the process outlined above. Usually a less expensive undercoater is applied, followed by the finish coats. Care must be taken to prevent the solvents used in the successive coats bleeding the preceding coats, thus producing muddy colors and sticky effects.

**ARRANGEMENT OF FINISHING EQUIPMENT.**—Fig. 40 shows a typical finishing department. Finishing rooms must be dust free, and thoroughly fireproof, since many solvent fumes are highly explosive. All fan motors must be spark proof. Conveyor chains and other moving parts should be guarded against friction and sparks. Fully air-conditioned finishing rooms permit standardization of drying conditions and make them independent of climate and weather.

Many finishing rooms have overhead conveyors, from which semi-finished units are suspended, progressing successively from station to station in the finishing cycle.

Some cheaper products are finished, in whole or in part, by dipping in tanks. Dipping applies especially to stains and sealers. Provision then must be made to drain and drip, to avoid excess material forming beads in objectionable locations.

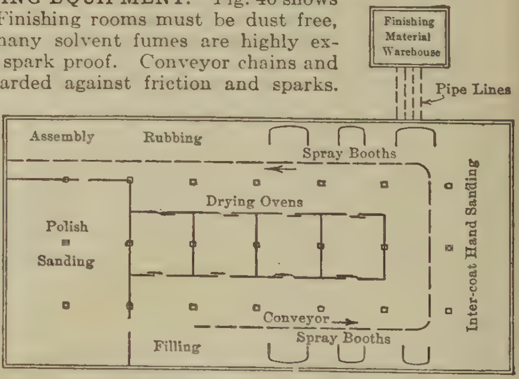


Fig. 40. Arrangement of Finishing Department.

**The Polish Sanding Department**, immediately preceding the application of finishing materials, should be separated from the spray booths by air-tight partitions to reduce dust hazards. Drying ovens should be centrally located, with several separately controlled, well-ventilated rooms to provide conditions required for drying filler, stain, undercoater and lacquer. Finishing materials are best stored in a separate fire-safe building, where all standard mixing can be done. All material should be piped and pumped to individual booth locations.

Hand sanding often is necessary between coats to remove small pimples that form around small particles of dust and undissolved solids. Many better grades of products require *rubbing*, to produce a dull, non-lustrous surface. On irregular shapes, rubbing may be by hand, or by holding the work against a cushioned revolving spindle. For table tops and similar products, mechanical stroke rubbers, with reciprocating cushion pads, are used. Rubbing down a good grade of lacquer gives a much better finish than semi-gloss lacquer, which lacks durability. The usual rubbing compounds are powdered pumice and oil, mineral oil and diatomaceous earth.

**Spray Booths** are almost universally used in finishing rooms. They are closely regulated by local ordinances, requiring direct fan connections to outdoors, to preserve health of operators and to reduce fire hazard. Spray booths generally are equipped with automatic revolving tables, with pneumatic vertical travel, to give access to legs, backs and under parts.

**Spray Guns** are atomizers, operated by compressed air at 75 to 100 lb. per sq. in. Guns giving smooth, glasslike, fan sprays are available that cover up to 60 in. width. Air volume may vary from $^1/_2$ to 50 cu. ft. per min. Dual, triple, and quadruple heads are available for attachment on spray guns for special products. Consult manufacturers' literature for details of spray booths and spray guns.

## Wood Preservation

Preservatives that lengthen the life of wood exposed to decay or attack by insects or marine borers are: 1. Preservatives of an oily nature, relatively insoluble in water; 2. Salts injected into wood in water solutions; 3. Toxic constituents in a volatile solution other than water; 4. Fire retardent wood treatments, which cover an industrial problem where municipal ordinances require fireproof construction. Intelligent selection for a particular service greatly prolongs usefulness of wood, with excellent ultimate economy.

**COAL TAR CREOSOTE,** the best known wood preservative, is a black or brownish oily distilate of coal tar. It is the most generally adaptable preservative for a wide variety of uses. The broadest field is outdoor service, with weather and underwater exposure. Its advantages are: 1. Toxicity to wood-destroying fungi and insects; 2. Relative insolubility in water and low volatility; 3. Ease of application; 4. Facility of determining depth of penetration; 5. General availability and relatively low cost. Its odor may be objectionable, but not serious, except where it may be transmitted to stored food products. The color sometimes is an objection where appearance is a factor, as creosoted wood cannot be successfully painted. Creosote does not increase the fire resistance of wood, but adds to its inflammability when freshly impregnated. The quality of coal tar creosote varies over a wide range. Grade 1 specifications of the A.W.P.A. (which are the same as those of the A.R.E.A.) should apply to creosote used for pressure treatments. For non-pressure treatments, the corresponding specifications of the A.W.P.A. should be designated. Coal tar, used alone, is too viscous for adequate penetration, but is used to dilute creosote for treating crossties. The blends most commonly used contain 20% to 30% tar.

**Creosote Petroleum** blends are extensively used for the treatment of crossties. On the average, blends contain 50% petroleum and although this decreases toxicity, it still is sufficient for the purpose. The presence of petroleum is claimed to retard checking, splitting and brooming. Creosotes of distinctly lower value for wood preservation are made from distillates of water gas tar and wood tar.

**Toxicity Tests** of wood preservatives may be defined as a " measure of extent or degree a substance acts deleteriously on wood-destroying fungi " (Schmitz).

**ZINC CHLORIDE** is a water soluble salt. Its chief advantages are: 1. General availability; 2. Relative cheapness; 3. Uniform quality; 4. Cleanliness; 5. Paintability; 6. Freedom from odor; 7. Ease of shipment; 8. Absence of fire hazard. On the other hand, it leaches out rapidly in damp soil and underwater locations, and is not as effective against decay as creosote. It is used in building construction, on roof planking, and is more adaptable for use on sawn lumber, as contrasted with creosote, on piles, posts, ties and large timbers. The preferable degree of impregnation is about 1 lb. of dry salt per cubic foot of wood, applied in 3% to 5% solutions, with subsequent evaporation of water solvent down to the moisture content required for the purpose intended.

**SODIUM FLUORIDE** is less soluble in water than zinc chloride, and leaches out of wood rather rapidly. It is approximately as effective as zinc chloride, but more expensive.

**MERCURIC CHLORIDE** has higher toxicity characteristics than zinc chloride. It is considerably more costly, and is objectionable because of its extremely poisonous character and tendency to corrosion.

**ARSENICAL PRESERVATIVES,** principally $As_2O_3$, have been used from time to time, but with questionable effectiveness.

The foregoing preservatives are no longer protected by patents. Some patented proprietary preservatives, among which are the three following, give excellent results.

**WOLMAN SALTS** are said to contain sodium fluoride as the principal ingredient in a water solution. They are of German origin, but have been used with reasonable success in the U.S. for mine timbers, roof timbers and miscellaneous lumber.

**ZINC-META-ARSENITE** is a somewhat recent American development. It is a water-soluble preservative. Experience with it so far has been encouraging.

**VOLTATILE SOLVENT SOLUTIONS,** in general, consist of a solution of beta naphthol, one of the chlor-phenols, or some other similar toxin in the light petroleum fraction commonly known as mineral spirits. These preservaties are clean, odorless and paintable, and when contrasted with water soluble salts, have the advantage of drying quickly. They are usable for treating window sash, doors, flooring, and other mill work. When properly applied, warping, twisting or raising of the grain is avoided.

**FIRE-RETARDANT TREATMENTS** are preferably by impregnation. They delay ignition and check the spread of fire over surfaces. For a thorough impregnation and a high degree of effectiveness, some authorities advocate approximately 300 lb. of chemicals per 1000 board feet of lumber, the cost of which often is prohibitive. Saturation must be as complete as possible, as many wood parts are extensively manufactured after treatment. Some fire-retardant treatments make the wood difficult to machine, requiring special cutters, frequently sharpened. Wood strength is not materially affected by fire-retardant treatments. The more usual chemical compounds for fire retarding have a predominating proportion of ammonium salts, but are rather complex and reference should be had to the literature of the industry and to that issued by the U. S. Bureau of Standards.

**EQUIPMENT AND PROCEDURE.**—All bark must be peeled from round poles and piles, to facilitate impregnation. Round pieces and sawn lumber must be well seasoned (air or mechanical), as any preservative treatment is practically useless when applied to green or wet wood. Sometimes this pre-drying is the initial step in the wood preservation process.

Complete impregnation is not essential, but penetration should be sufficiently deep to avoid subsequent checking or splitting exposing untreated areas. If possible, all cutting, framing, boring, etc., should be done before treatment. If such work is necessary after treatment, the freshly cut surfaces should be given two or three brush coats with hot preservative. Holes drilled on the job may be pressure treated by a simple device marketed for this purpose.

An approved method of increasing penetration depth and diffusion is *incising*. Sawed or hewed timbers are passed through a machine, in which teeth on horizontal and vertical rollers sink into the wood from $^1/_2$ to $^3/_4$ in. The incisions are closer on the denser and less absorptive woods.

**The Pressure Processes** are the most effective methods of treatment. The timber, separated to expose all surfaces, is placed on steel car bunks and run into a long steel cylinder, which then is closed and filled with preservative under pressure. Temperatures for efficient penetration usually are specified between 160° and 200° F., but are more advantageous toward the upper limits.

**The Full-cell Pressure Process** (Bethell) is preferable for both oil and water solutions, when retention of a maximum amount of preservative is desired. It is not so satisfactory when limited oil absorptions are specified. Limited absorptions of water-soluble preservatives are obtained by lower concentration of solids, and subsequent evaporation of the water. The steps in this process are: 1. Charging the cylinder; 2. Applying a preliminary vacuum (10 to 12 in.) to remove all possible air from the cylinder and wood pores; 3. Admission of hot preservative, without air, to the cylinder; 4. Application of pressure (150 to 175 lb. per sq. in.) until required absorption is obtained; 5. Withdrawal of preservative; 6. A short final vacuum to free the charge from dripping preservative.

**Empty Cell Pressure Processes** omit the initial vacuum step. In the Rueping process, an initial air pressure (25 to 100 lb. per sq. in.) is applied for a maximum of one-half hour with the denser timbers. The preservative is under a pressure of 150 to 200 lb. per sq. in. The Lowry process admits preservative to the cylinder under a pressure of 150 to 175 lb. per sq. in. without either initial vacuum or air pressure. Neither empty-cell process will impregnate as heavily as the full-cell process, due to the resistance of the air imprisoned in the wood cells.

**PRE-CONDITIONING.**—When natural or mechanical seasoning is impracticable, green timber is pre-conditioned, after peeling, on the treating car bunks in a closed cylinder, which may be the regular treating cylinder. The process most used for yellow pine is to steam the charge at 20 lb. per sq. in. (259° F.) for several hours, followed by a like period of vacuum. The effect of the vacuum is to lower the boiling temperature in the

**Table 14.—Specifications for Wood Preservation Treatments**

| Form | Species | Use or Condition of Service | Preservative * | Specification No. | | Absorption, lb. per cu. ft., min. | | Specification No. for Treatment ‡ | Remarks |
|---|---|---|---|---|---|---|---|---|---|
| | | | | Federal | Am. Wood Preservers' Assoc. | Empty-cell | Full-cell | | |
| Crossties | All † | ................ | CT; CS | 556; 566 | 4d; 5b | 6 | 12 | 34b | Note 1 |
| Crossties | Douglas fir | Incised | CT; CS | 556; 566 | 4d; 5b | 6 | 12 | 38a | Note 1 |
| Crossties | All † | ................ | ZnCl | 576 | 17a | ....... | 1/2¶ | 34b | Note 2 |
| Crossties | Douglas fir | ................ | ZnCl | 576 | 17a | ....... | 1/2¶ | 38a | Note 2 |
| Piling | All † | Land; fresh water | CT; CS | 556; 566 | 4d; 5b | 8 | 16 | 39a | Note 3 |
| Piling | All † | Marine | CT | 556 | 4d | ....... | 16 | 39a | Note 4 |
| Piling | All † | Marine; severe | CT | 556 | 4d | ....... | 22 | 39a | Note 5 |
| Piling | Douglas fir | Land; fresh water | CT; CS | 556; 566 | 4d; 5b | 8 | 12 | 41a | Note 3 |
| Piling | Douglas fir | Marine | CT | 556 | 4d | ....... | 14 | 41a | Note 6 |
| Poles | Pine | Ordinary | CT | 556 | 4d | 8 | ..... | 36b | Note 7 |
| Poles | Pine | Severe | CT | 556 | 4d | 12 | 16 | 36b | Note 8 |
| Poles | Douglas fir | Ordinary | CT | 556 | 4d | 8 | ..... | 41a | Note 7 |
| Poles | Douglas fir | Severe | CT | 556 | 4d | ....... | 12 | 41a | Note 8 |
| Poles | Cedar | Incised § | CT | 556 | 4d | ....... | ..... | 43b | |
| Poles | Chestnut | Not incised § | CT | 556 | 4d | ....... | ..... | 44b | ........ |
| Fence posts | ........ | ................ | CT; CS | 556; 566 | 4d; 5b | 6 | ..... | 37b | Note 9 |
| Structural timber | Pine | Under 5 in. thick | CT; CS | 556; 566 | 4d; 5b | 10 | 16 | 35c | Note 10 |
| Structural timber | Pine | Over 5 in. thick | CT, CS | 556; 566 | 4d; 5b | 8 | 12 | 35c | Note 10 |
| Structural timber | Douglas fir | Under 5 in. thick | CT; CS | 556; 566 | 4d; 5b | 10 | 16 | 45a | Note 10 |
| Structural timber | Douglas fir | 5–9 in. thick | CT; CS | 556; 566 | 4d; 5b | 8 | 10 | 45a | Note 10 |
| Structural timber | Douglas fir | 10–14 in. thick | CT; CS | 556; 566 | 4d; 5b | 6 | 8 | 45a | Note 10 |
| Building lumber | All † | Painted | ZnCl | 576 | 17a | ....... | 1¶ | 35c | Note 11 |
| Building lumber | Douglas fir | Painted | ZnCl | 576 | 17a | ....... | 1¶ | 38a | Note 11 |

* CT = Coal-tar creosote. CS = creosote coal-tar solution. ZnCl = Zinc chloride. † Except Douglas fir. ‡ American Wood Preservers' Assoc. § Butt treatment. ¶ Based on dry salt.

NOTE 1.—Empty-cell treatment recommended for ordinary conditions of use; full-cell treatment for exceptional conditions.

NOTE 2.—Not recommended for arid or very wet regions.

NOTE 3.—Full-cell treatment recommended for important permanent structures where long life is of special importance, or for use in warm, humid regions.

NOTE 4.—For North Atlantic waters.

NOTE 5.—For waters where marine-borer attack is severe.

NOTE 6.—For all waters where marine borers are active.

NOTE 7.—For use in the U. S.

NOTE 8.—For use in tropical regions.

NOTE 9.—Round posts recommended.

NOTE 10.—Full-cell treatment recommended for timber used under salt water. Empty-cell treatment suitable for most ordinary purposes.

NOTE 11.—For building lumber not in contact with ground, where painting is desirable, or creosoted material is otherwise unsuitable.

cylinder and facilitate evaporation of moisture from the wood. The Boulton process is a preferred, and less harsh treatment for Douglas fir and hardwoods. This is a boiling-under-vacuum treatment in which the standard cylinder charge is heated in the preservative oil at 180° to 210° F. under vacuum. As compared with steaming, the lower temperatures are less likely to injure the wood. The Boulton process removes more moisture from the outer surfaces (1 to 2 in. deep), but makes little change in the original moisture content below this depth.

**ABSORPTION AND PENETRATION.**—The absorptive capacity of wood varies considerably with different densities of wood species, and the varying cell capacities of sap and heart wood, and spring and summer wood. The amount absorbed is less per cubic foot in large timbers than in small poles, because of the larger proportion of center wood that is a negligible absorbing factor. A summary of Federal specifications for preservative treatment of wood is given in Table 14.

**NON-PRESSURE PROCESSES** do not impregnate as thoroughly as the pressure processes, but are the only practical methods for butt ends of poles, posts and other items that require treatment only on the sub-soil end. They sometimes are used on submerged wood, where low costs are imperative or when pressure tanks are not available.

**The Hot and Cold Bath Treatment** (Hunt process) is the most effective non-pressure process. The wood is heated in the preservative in an open tank for several hours, and then cooled in preservative for a like period. This may be accomplished by: 1. Transferring the wood from a tank of hot oil to one of cold oil; 2. Draining the hot preservative from a single tank, and quickly refilling with cold oil; 3. Shutting off the heat, allowing preservative and wood to cool together gradually.

Coal-tar creosote is the medium generally used in this process, as it is preferred for under-soil exposures. The hot temperature is about 230° F.; cooling should be to about 100° F. Higher temperatures penetrate better, but are subject to evaporative losses. The length of bath depends on the absorptive characteristics of the wood and the degree of penetration required. It is best determined by the conditions on each job. If penetration is insufficient, lengthen the time; if too much oil is absorbed, shorten the cold bath.

**The Steeping Process** is old, but not extensively used. Timbers are soaked in a cold solution of preservative for several days. It is mostly used with water-soluble compounds on sawn lumber, which should be separated to maintain complete face exposure. Only fair results can be expected, especially with treatments of 2 or 3 days duration. If time is not a factor, 10 to 15 days is much better.

**Miscellaneous Processes.**—In the *dipping* process, wood is submerged in preservative at 200 to 230° F. for 15 minutes. The effectiveness is limited. *Brushing* and *spraying* are even more superficial. Their use sometimes is unavoidable in emergencies, and on parts that have to be cut on the job.

## Wood Bending

Several processes of bending are in use. The technique differs between solid wood and plywood, as the fiber strain usually is much greater and more unbalanced in solid wood. Fundamentally, in bending wood strips the individual wood fibers tend to slide by each other slightly; outside fibers are stretched under tension, and inside fibers are compressed.

**WET BENDING IN SOLID WOOD.**—Any wood part is more easily bent when soaked or steamed to a point of thorough saturation. Consequently, bending is better and breakage less with steamed or cooked semi-dried lumber than with kiln-dried lumber, which is brash. Pieces to be bent should be dressed as nearly as possible to final sizes before bending.

The following process is applicable to pieces of approximately square cross-section; for wider or flat pieces, see Dry Bending below. Three factors are important in wet bending: 1. A smooth rigid form without abrupt changes in curvature on the concave side of the curve; 2. A thin flexible metal strip, on the convex side, drawn tight, to reduce breakage; 3. Sufficient end pressure to eliminate danger of shear on any fiber line. See Fig. 41. The cycle of bending and drying is much shortened if forming is done in a press under heat.

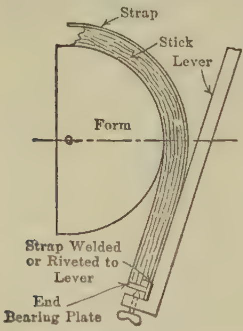


FIG. 41. Fixture for Wet Bending

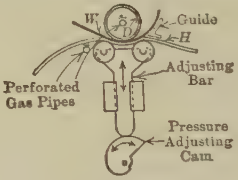


FIG. 42. Arrangement for Dry Bending Solid Wood

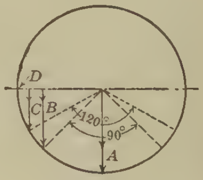


FIG. 43

**DRY BENDING IN SOLID WOOD.**—A preferable method of bending wider and thinner pieces of solid wood is a series of three rollers, as in Fig. 42. Heat is applied to the upper roller *D*, and below the endless steel flexible conveyor belt *H*. The wood *W* is rocked back and forth until the desired degree of curvature is obtained. Temperatures may be as high as possible without charring wood surfaces. This method can be used for such items as curved slats for chair backs, and for plywood up to $^5/_{16}$ in. thick, if the adhesive is heat resistant (resin or casein). The bent pieces, both of solid wood or plywood, should be placed in skeleton curved forms until thoroughly cooled and normalized as to moisture content.

**BENDING PLYWOOD DURING GLUING ASSEMBLY.**—As a practical manufacturing problem, only such curves can be economically bent in plywood assembly operations as can be formed in hydraulic presses with vertical pressure. There is a definite limit to the extent of curved segments that can be glued effectively. Fig. 43 shows that, with a given direct pressure *A*, a 90-deg. segment will have a pressure of approximately $^2/_3$ *A* at its extremity *B*, while a 120-deg. segment will have a pressure of $^1/_2$ *A* at the extremity *C*. It is possible to bend a 120-deg. segment successfully, and it will include enough more than a quadrant to produce grooved and well-fitted corners. A 180-deg. segment would have no vertical pressure component at *D*; pressure would be derived only from the snugness of the forms. The limit of 120-deg. segmental angle applies to both simple and compound curved surfaces.

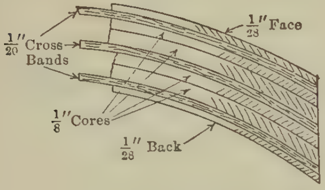
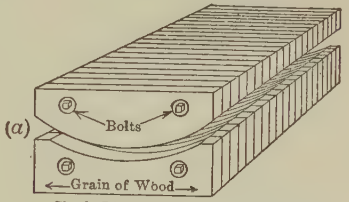


FIG. 44. Typical Curved Construction, Thick Plywood



Single Wood Forms for Cold Pressing
(For Hot Pressing made integral of Aluminum)

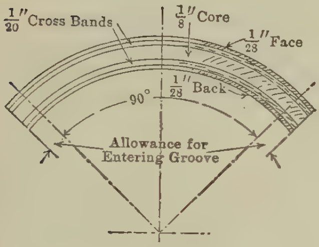
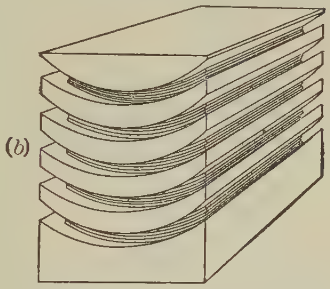


FIG. 45. Typical Curved Construction, Thin Plywood



Multiple Aluminum Forms for Cold Pressing

FIG. 46. Platens for Bending Plywood

Figs. 44 and 45 show typical curved segment plywood constructions. The thickest layers of veneer are bent with the grain, and thin sheets only are bent against the grain. Increasing the number of plies (Fig. 44) often facilitates the bending operation, and reduces the thickness of the layers.

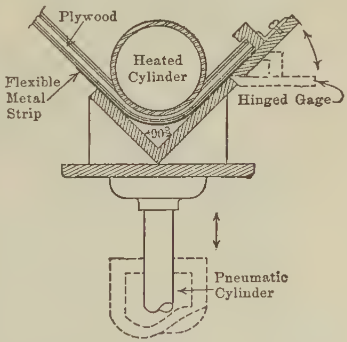


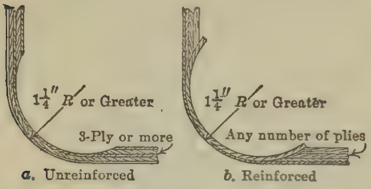
FIG. 47. Plywood Bending Press

**Cold Bending.**—Plywood bending operations can be done with liquid adhesives, vegetable, casein and animal glues, in unheated hydraulic presses, utilizing the usual I-beam clamps on one or a series of pairs of forms (Fig. 46). A quick-setting (3 to 4 hr.) glue should be used to reduce investment in forms. After removal from gluing forms, the curved plywood should be racked in curved skeleton forms, and weighted during re-drying. See p. 22–26.

**Hot Bending.**—Better results are obtained by making curved plywood with film or liquid resin in a hot press. Single assemblies of plywood are bonded, in one pair of male and female forms similar to Fig. 46*a*, but made of aluminum. A pair of forms is placed between each pair of heated platens. Upon completion of bonding, requiring about 10 minutes, the forms are again available and may be used as many as 50 times per day. The plywood does not require re-drying to remove glue water, nor is curved hot-pressed plywood inclined to warp or lose shape.

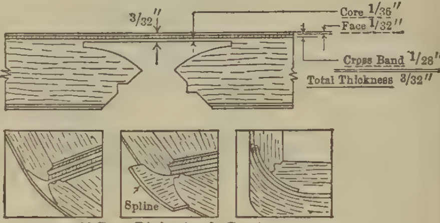
**BENDING PLYWOOD AFTER BONDING.**—Plywood with water-proof resin bonding can be steamed and softened after bonding to facilitate bending. This has opened up a new technique for curved plywood, allowing sharper bends and curvatures up to nearly a complete circle. These bends can be made in 1/8-in. plywood and thinner without recessing or cutting away on the concave side. Plywood thicker than 1/8 in. and up to approximately 1/2 in. is recessed on the concave side, leaving about 3/32 in. of plywood in the bend. The plywood is thoroughly steamed at the point to be bent, placed in a bending press, Fig. 47, and the lower female form pressed up against a heated cylinder of the required diameter. The effect of the heat is to set the bend permanently. If the resulting bend is to be framed in a groove no reinforcement is necessary (Fig. 48*a*). If the curved plywood is unsupported, a sheet of veneer should be added (Fig. 48*b*), with its grain running around the curve, and bonded to the inside curve by a liquid or film resin. This latter construction is patented.

The process of bending plywood with lumber cores is somewhat different. The back is recessed, as in Fig. 49*a*, down to the face crossband, leaving only the crossing and face to be flexed. The wood core is cut to such a miter as will meet when the curve is made. A sufficient slot is provided, next to the face crossband, into which a glue-covered spline can be inserted, the grain of the spline being parallel to the axis of the curve. When the glue is set on the curved spline, the joint will be firm. Glue setting usually is done while the plywood is clamped in curved position.

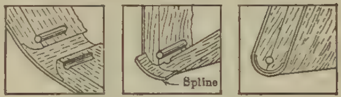


*a.* Unreinforced *b.* Reinforced

FIG. 48. Bending Plywood After Bonding



*a.* 90-Deg. Right Angle Bend



*b.* Full 180-Deg. Bend

FIG. 49. Bending Plywood with Lumber Cores

**FLEXIBLE PLYWOOD** may be made 3-ply with a very thin core, and thick loosely-cut outsides. It is suitable only for rougher uses that are to be painted, and is not adaptable to figured face woods.

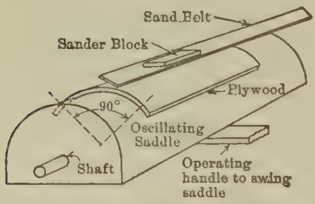


FIG. 50. Method of Sanding Convex Plywood

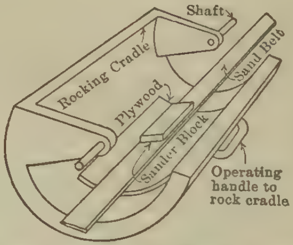


FIG. 51. Method of Sanding Concave Plywood

**SANDING OF CURVED SHAPES** presents some unusual problems. Good results may be obtained by attaching either the saddle or cradle, Figs. 50 and 51, to a belt sander to facilitate sanding either the inside or outside face of curved plywood.

## 4. SMALL TOOLS

Small tools for woodworking can, in general, be used practically unchanged for working the plastic and phenolic materials, and also rubber.

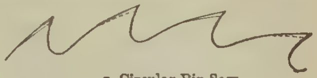
**THEORY OF CUTTING WOOD.**—Wood is made up of parallel fibers. Cutting lengthwise is simply peeling a layer of fibers off from those below, as with a splitting axe or carpenter's plane. In cutting across the fibers, every fiber must be severed. A knife or chisel driven forcibly across grain is the simplest form of cross cutting. Both examples represent " line " cutting without waste.

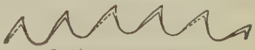
Saws divide the fibers, lengthwise or crosswise, by cutting out a swath or *kerf*. In a cross-cut kerf the fiber first is severed at both sides of the swath; the short cut fiber then is removed by a lengthwise cutting at the bottom of the kerf. In ripping a kerf with the grain, the fiber in the swath to be removed first is cross cut and then peeled loose from the sides of the kerf. Cross cutting requires a knife point cut at each side of the kerf. This is made by teeth alternately high at the sides and pointed and beveled on the front. *Raker* teeth may be used to peel or split the severed fiber and remove it. Ripping requires chisel-pointed teeth, set or swaged to the width of the kerf, and with enough hook to sever the fiber with an adzing cut. It simultaneously is split loose at both sides and removed.

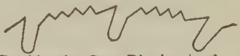
Saws are set, or hollow ground, to reduce friction. Surface tension or case-hardening of the lumber may cause boards to pinch or close the kerf during a cut. This is one of the principal reasons for wide clearance and wide kerfs. See Fig. 52*d*.

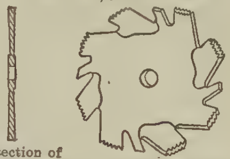
With revolving *cutter heads* having knives or bits, the cut generally is lengthwise. It is not exactly a splitting cut, since only at the finish line of straight grain woods is the cut parallel with the grain. At other parts of the cut, the knives are chiseling or adzing, and cross-cutting the grain.

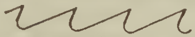
**SAWS** may be classified generally as: 1. Saws for the *primary conversion* of logs into lumber and timber. Being used on live or green logs, there is no danger of overheating and damaging saw temper. Such saws should be designed to cut smooth and to close dimensions, as subsequent shrinkage necessitates machining to final size and shape. Cutting of tough, green wood requires a different tooth than does the cutting of stiffer and stronger, but more brittle, dry wood. 2. Saws for the *secondary conversion* of the board after seasoning, into intermediate or final products. Saws may overheat and lose temper in dry wood, and may scorch or discolor it. Accurate, smooth cuts must be made, to produce final dimensions with maximum smoothness and minimum waste. Tooth shape must be such as will cut rapidly without heating, and accurately, with minimum power requirements. The thickness of saws is designated by Stubs gage. See p. 6–23.

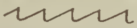

*a.* Circular Rip Saw


*b.* Circular Cut-off Saw


*c.* Combination Saw. Ripping teeth with short rake, and cross-cut teeth


*d.* Cross-section of combination saw hollow ground
*e.* Huther Patent-tooth Solid-plate Grooving Saw


*f.* Band-saw Teeth Showing Hook


*g.* Narrow Band-saw Teeth

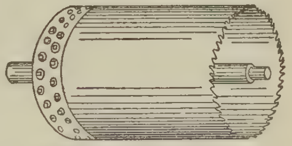

*h.* Cylinder Saw for Barrel Staves

FIG. 52. Forms of Saw Teeth

These two general classifications may be still further subdivided as below. Fig. 52 shows the various forms of saw teeth.

Saws for primary production (converting green logs into timber and lumber):
- Circular saws
  - Solid discs
    - Parallel-to-grain or rip cut (Fig. 52*a*)
    - Perpendicular-to-grain or cross cut (Fig. 52*b*)
  - Inserted teeth
  - Shingle saws, thin edges with thick centers
- Band saws, continuous or endless (Fig. 52*f*)
  - Parallel-to-grain for cutting logs into lumber and timber
  - Perpendicular-to-grain, drag saws for log lengths
- Band saws, reciprocating, mostly gang resaws
- Special purpose saws
  - Cylindrical, for cooperage stock (Fig. 52*h*)
  - Concave

Saws for secondary production (converting dry boards into diversified wood products):
- Circular saws
  - Ripping (Fig. 52*a*)
  - Cross cutting (Fig. 52*b*)
  - Combination (Fig. 52*c*)
  - Special purpose (Fig. 52*e*)
- Band saws, continuous or endless
  - Ripping or resawing, wide (Fig. 52*f*)
  - Curved and special shape, narrow (Fig. 52*g*)
- Band saws, reciprocating
  - Jig and scroll saws
- Special purpose saws
  - Dadoing (Fig. 52*e*)
  - Tennoning and mortising

**KNIVES AND CUTTERS.**—The function of a knife or cutter on a planer, moulder or shaper is to reduce the surface or edge of a part to the desired shape and dimensions without marks or scratches, and with a smoothness that can be finished. A roughly-sawed surface or edge requires more subsequent machining than one smoothly sawed. Close correlation between the roughness of the saw cut and the depth of the planer cut is necessary. Often, as in jointing plywood cores, a saw can be made to cut smooth enough to eliminate a separate jointing operation. A slightly rough cut is considered better for gluing than one with too smooth an edge.

The basic machines using knives and cutters are: 1. Planers or jointers using straight cutting edges to produce plane or level surfaces. They cut continuously along a straight face, not on the side, and parallel or nearly parallel to the grain. 2. Moulders or stickers, which are, in effect, multiple planers, usually designed to cut straight or irregular surfaces on all four sides of a piece simultaneously. Cutting may be with either face or side of the knife, but substantially parallel with the grain. 3. Shapers, which are particularly designed to produce irregular and curved shapes or edges both in vertical and horizontal planes. They use a variety of solid, sectional, and laminated cutters, often cutting at an abrupt angle to the grain.

On relatively large diameter wheels, the method of attachment of knives and cutters for planers, moulders and shapers, determines the classification. This involves both the provision made for knife attachment, and the corresponding design of head to which knives are attached. Rigid mounting is imperative because of high operating speeds, strong centrifugal forces, and the deep cuts often required.

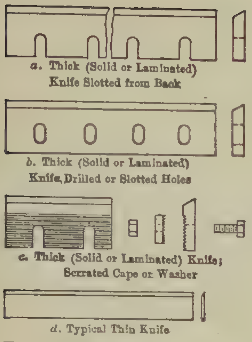
Knives are distinguished, in general, by thickness as: 1. *Thick knives*, usually comprising a tempered steel face and cutting edge, welded to a softer steel back used for clamping. Narrow knives of the chisel type may be entirely of tool-steel, tempered principally at the cutting edge. 2. *Thin knives* which require clamp bars and plates, or attachments. Such tool-steel knives seldom are recessed for attachment. They require stiffer bars to provide rigidity and firmness. A knife that chatters will not cut smoothly.

Knives and cutters may be classified by machines and by thickness, as follows:

**Planers and Jointers** for cutting straight surfaces, single and double, parallel to wood grain.
- *Thick rigid knives*, parallel to axis of head (Fig. 53*a*)
  - Square heads
    - Two-knife (Fig. 54*a*)
    - Four-knife (Fig. 54*c*)
  - Face clamping
    - Oval holes in knife with bolts adjustable in slotted head (Figs. 53*b*, 54*c*)
    - Slotted holes in knife, with tapped holes in head (Figs. 53*a*, 54*a*)
    - Ribbed-surface knives for rigid clamping (Fig. 53*c*)
  - Round heads
    - Clamped with countersunk bolt heads
    - Internal bolt clamping devices
- *Thin knives*, semi-flexible, requiring support and clamp bars (Fig. 53*d*)
  - Knives parallel to axis of head
    - External clamp bars and bolts
      - Square heads (Fig. 54*d*)
      - Round heads (Fig. 54*g*)
    - Internal clamp bars and bolts (Figs. 54*e*)
    - Wedge and taper clamps (Fig. 54*f*)
  - Knives spiral to axis of head
    - External clamp bars and bolts
    - Internal clamp bars and bolts

**Moulders and stickers** for cutting multiple straight and irregular surfaces parallel to wood grain (Figs. 56*a*, 56*b*, 56*c*)
- *Thick rigid knives* (straight, not spiral)
  - Angular heads
    - Two-knife (Fig. 54*a*)
    - Three-knife (Fig. 54*b*)
    - Four-knife (Fig. 54*c*)
  - Face clamping
    - Tapped holes in head (Fig. 54*a*)
    - Bolt slots in head (Figs. 54*b*, 54*c*)
    - Ribbed-surface knives for rigid clamping (Fig. 53*c*)
  - Edge clamping
    - External bolts (Figs. 55*a*, 55*b*)
    - Vise or jaw clamps (Figs. 55*c*, 55*d*)
    - Slotted bolts with adjustable cutting angles (Fig. 55*c*)
  - Round heads
    - Clamped with countersunk bolt heads
    - Internal bolt clamping device
    - Wedge and taper clamps (Fig. 55*f*)
- *Thin knives*, semi-flexible; requiring support and clamp bars
  - Knives, parallel to axis of head
    - External clamp plates and bolts (Fig. 54*d*)
    - Internal clamp plates and bolts (Fig. 54*e*)
    - Wedge and taper clamps (Fig. 54*f*)
  - Knives spiral to axis of head
    - External clamp plates and bolts
    - Internal clamp plates and bolts

**Shapers** for cutting irregular surfaces in two planes and frequently at abrupt angle to wood grain
- *Knife blades clamped to head*
  - Face clamped to head
    - Slotted hole, tapped hole in head (Fig. 54*a*)
    - Slotted knife and bolt in head (Figs. 54*b*, 54*c*)
  - Edge clamped
    - External bolts (Figs. 55*a*, 55*b*)
    - Vise or jaw clamps (Figs. 55*d*, 55*e*)
    - Slotted bolts with adjustable cutting angles (Fig. 55*c*)
  - Clamped with pressure bars or plates (thin knives)
    - External clamped with countersunk bolt heads
    - Internal clamped
    - Wedge and taper clamps
  - Thin knives of slightly irregular shapes (Figs. 56*a*, 56*b*, 56*c*)
- *Solid knife heads mounted on arbor*
  - Thick knives of a wide range of shapes (Figs. 56*d*, 56*e*)

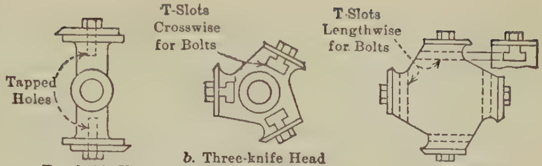

*a.* Thick (Solid or Laminated) Knife Slotted from Back

*b.* Thick (Solid or Laminated) Knife, Drilled or Slotted Holes

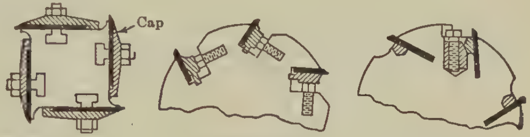
*c.* Thick (Solid or Laminated) Knife; Serrated Caps or Washer

*d.* Typical Thin Knife

FIG. 53. Forms of Cutter Knives



*a.* Two-knife Head *b.* Three-knife Head *c.* Four-knife Head



*d.* Four knife Square Head for Thin Steel Bits *e.* Six-knife Solid Round Head *f.* Six-knife Solid Round Head

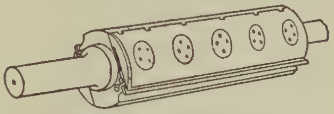
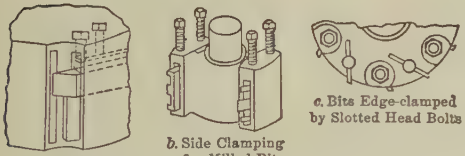
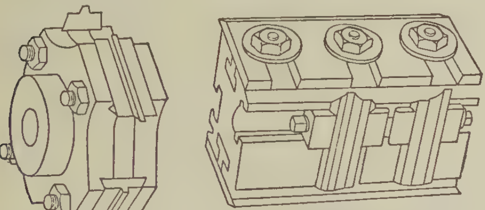

*g.* Round Cylinder and Safety Cap; Two Thin High-speed Steel Knives

FIG. 54. Forms of Cutter Heads

**BORING TOOLS.**—The principle underlying a boring tool or bit is similar to that of a shaper tool, except that the boring tool advances into the wood. Mechanical means must be provided for this movement, as well as for the removal of chips from the enclosing hole along the spiral shank of the bit. The two basic types of boring tools are: 1. Boring into *side wood,* where the bit requires side spurs to sever the wood fibers on the circumference of the hole, before the cutting and lifting edge of the bit separates the chips along the fiber line. 2. Boring into *end wood,* where the bit shears off the fibers as it advances, the cutting of the circumference being merely a separation of the fibers. See Fig. 57.

**Bits.**—Fig. 58 shows four main types of bits. The *double-spur* bit is a general utility tool, usually made with a single-thread screw point. The *extension lip* bit usually has a double-thread point. It is used to produce smoother holes, with less liability to split hard


*a.* Edge-grip Head *b.* Side Clamping for Milled Bits *c.* Bits Edge-clamped by Slotted Head Bolts


*d.* Four-bit Vise-grip Head *e.* Head with Slotted Knife and Clamped Bits Held by Clips

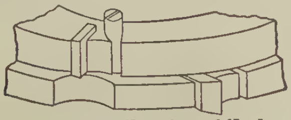
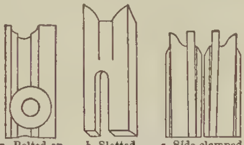

*f.* Taper-Pin Face-clamped Head

FIG. 55. Methods of Clamping Cutter Knives and Bits


*a.* Bolted-on Type *b.* Slotted Type *c.* Side-clamped Type

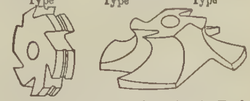

*d.* Thin Solid Knife Head *e.* Thick Solid Knife Head

FIG. 56. Bits and Solid Knife Heads

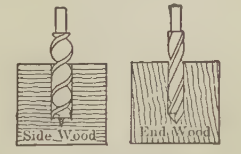


FIG. 57. Boring into Side and End Wood

wood. Both types are used to bore holes in side wood. The *taper head drill* is designed for boring end wood. It may be used for side wood if a rough hole is not objectionable. The *spur machine drill* is a modification of the taper head drill. It has spurs and a brad point, to adapt it to better boring service in side wood. It is used in gang boring where holes may be at various angles to the grain.

The kind of spiral groove or twist for chip removal, depends on bit size, depth of hole, and direction of grain. Fig. 59 shows the more important types. The *double twists* are usual for medium and large regular boring; left-hand is standard for a bit turning clockwise. Some gang boring machines require right-hand twist bits, rotating counter-clockwise. The *solid center twist,* with a round stem or shank in the center, is used for small bits. The *ship auger twist* has the largest body of metal on the edges, the center being used for more adequate chip removal. It is most suitable for deep boring in large timbers. The *drill twist* has a much more rapidly rising spiral than others, its function being the removal of smaller end wood cuttings, rather than side wood chips. It is so ground that the margin is wide enough to resist wear, thus maintaining the diameter.

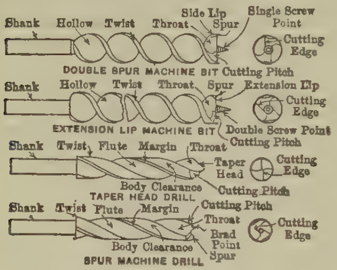


FIG. 58. Types of Boring Bits

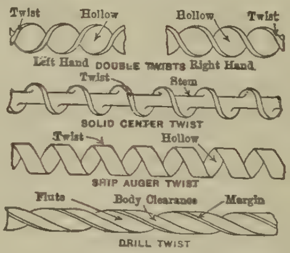
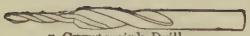
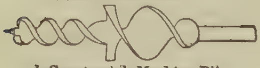


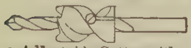
FIG. 59. Types of Chip Removal Grooves



*a.* Countersink Drill

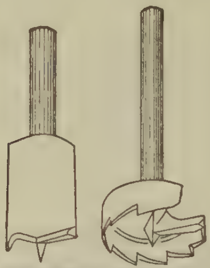


*b.* Countersink Machine Bit



*c.* Adjustable Cutter with Taper Head Drill

FIG. 60. Counterbore Bits



*a.* For Solid Wood *b.* For Plywood

FIG. 61. Bits for Large Diameter Shallow Holes

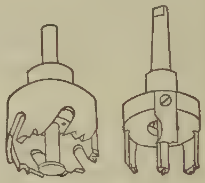
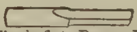
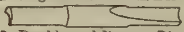


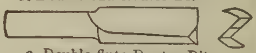
FIG. 62. Plug Cutter



*a.* Single-flute Router Bit



*b.* Double-end Router Bit



*c.* Double-flute Router Bit Made Right and Left Hand

FIG. 63. Router Bits

Fig. 60 shows bits for boring concentric holes, sometimes called counterbores. The countersink *a* is for flat-head screws, while recessing, as for washers or bolt heads, is done with one of the forms of counterbore of which *b* and *c* are examples. Many other combinations may be made for both side and end wood.

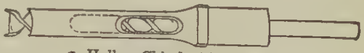
Large diameter shallow holes do not require bits with spiral shanks, as chips seldom clog. Fig. 61 shows two types, *a* for solid wood, and *b*, with saw teeth on the circumference, for plywood. Both have brad points.

Fig. 62 shows *plug cutters* with ejector plungers, one of which has inserted high-speed steel teeth for abrasive plywood adhesives, as casein.

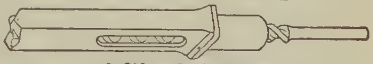
Router Bits, Fig. 63, while similar in appearance to boring tools, have different functions. A router is chiefly used to cut recessed grooves, or to cut out, by means of grooves, regular or irregular outlines. A router bit has a cutting lip throughout its length to produce smooth-edge grooves. It rotates at much greater speeds than a boring bit. The function of side cutting is much more important than end cutting. The advance cut of a spur is not needed, and the fluted clearance slot, for chip removal, may be straight instead of spiral. Router bits may be designed for various shapes, like T-slots, ogee grooves and the like. In this form they may resemble shaper knives and heads.

**MORTISING TOOLS,** Fig. 64, are duplex boring and chisel cutting tools. A central standard bit bores a round hole, and, surrounding the bit, a square four-side chisel removes or broaches out the round fillet corners. Chips from both cuts are lifted out by the spiral shank of the bit and discharged through clearance clots in the square chisel shank (one for soft wood, two for hardwood). The slots are located above the wood surface at maximum cutting depth. Chisel points are longer at the corner than at the center, to insure a shearing cut in removing corner fillets.

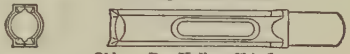
Oblong holes, up to 1 $1/_2 \times 1$ dimension ratios, can be made with a rectangular oblong chisel, as in Fig. 64*b*. If a 2 × 1 ratio is required, and the shape of hole is concealed by shoulders on the tenon, the result can be obtained at a single stroke by the hollow chisel, Fig. 64*c*. The cutting edges of the chisel are arranged for shear cuts on all four sides. Otherwise, oblong holes are made by multiple strokes of a square mortising tool. Proper clearances must be provided between internal bits and external chisels to avoid friction, heating and burning the wood. In setting mortising tools, the chisel is set first, and then the bit, with the bit cutting lips extending a little beyond the extreme cutting tips of the chisel. Mortising tools are not suited for cutting end wood.



*a*. Hollow Chisel with Chisel Bit



*b*. Oblong Hollow Chisel



*c*. Oblong Bar Hollow Chisel

FIG. 64. Mortising Tools

**Mortising Chains** are available for oblong holes, being a series of cutters attached to an endless chain. They are not boring tools, but produce a similar result. The hole has a rough edge, but may be satisfactory for such products as mortise-and-tenon doors, although the joint cannot be very snug.

## 5. MACHINE TOOLS FOR WOOD *

### Saw Machines

The five principal types of saw machines are: 1. *Rip-saws*, circular cutting; 2. *Cross-cut saws*, circular cutting; 3. *Variety saw tables*, circular cutting; 4. *Band saws*, continuous band cutting; 5. *Jig* and *scroll saws*, reciprocating band cutting. In many machines, rip, cross-cut or combination-tooth saw discs may be used interchangeably, while some are single purpose machines. See p. 22–35 for saw disc types and teeth.

Saw cuts are required on a variety of machines, where the saw cut is auxiliary to the general purpose of the machine. Examples are double-end tenoners, dado heads or grooving cuts on moulders, trim cuts on veneer jointers, or a cut-off on a dowel machine. In general, the best cutting speed for either a circular saw or band saw is approximately 10,000 ft. per min.

**RIP SAWS.**—Machines designed exclusively for ripping are of two main classes: 1. Single or multiple ripping of full length boards, often the preliminary dimensioning operation in a planing mill, and sometimes in a furniture factory. In gang set-ups, to avoid a winding cut, saws must be of equal diameter and approximately the same degree of set and sharpness. Saws usually are mounted on the lower arbor, with pressure or feed rollers above. 2. Single ripping of cross-cut stock (often called straight-line rip saws). The saw usually is mounted above, with an endless feed chain below. Such saws often give a sufficiently straight and true edge for gluing. Arbors are sometimes below the work, the saws extending upward between a divided feed chain, above which are mounted idling pressure or feed rollers.

**CROSS-CUTTING SAWS** (also called stock-cutting saws) may be of the following types: 1. Swing saw (an older type) hung from the ceiling and swinging into the work on an arc; 2. Clip saw, swinging from below, with the motor drive pulley concentric with the pivot; 3. Saws on sliding ways, or other mechanical device, to move them horizontally into the work and out. Cross-cutting usually is the first dimensioning operation in a furniture factory.

**VARIETY SAW TABLES** may be equipped interchangeably with rip, cross-cut or combination (rip and cross-cut) saw discs, depending on the work. For grooving or tenoning, dado heads or *wobble saws* may be used. Wobble saws are angle mounted, with wedge-shaped collars, between two perpendicular saws for slotting or rabbeting. Variety saw tables generally are equipped with side gages, back gages, and angle adjustments for miter sawing. For angle cutting, the table may tilt or the arbor may be pivoted. In size, they range from small bench saws up to floor machines. Speed of floor units usually is 3400 to 3600 r.p.m., requiring 5 Hp.

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* For machines for primary saw mill conversion operations, see p. 22–09.

Since all circular saws cut against the feed, the operator should take great care to avoid dangerous " kick backs." Most modern saws have mechanical devices to protect against both kick backs and hand hazards from the saw discs.

**BAND SAWS.**—Resaws for such operations as ripping special thicknesses, usually are about 6 in. wide, with heavy corrugated feed rollers on vertical shafts. Regular band saws may be equipped with a variety of blades, seldom wider than 1 in. for straight work, and down to $^1/_4$ in. for curved work. Band saw wheels range from 30 to 36 in. diameter, with direct-connected motor on the lower wheel. Band saw throats take work up to 36 in. wide by 18 in. thick. Motors are 3 to 7 $^1/_2$ Hp.; wheel speeds are 600 to 1200 r.p.m. Small bench models also are made.

**JIG AND SCROLL SAWS** are used chiefly for cut-outs that cannot be made on band saws, and are much less costly than band saws for small shops. The development of the high-speed router, producing smoother edges, has greatly lessened the utility of the jig saw.

## Planing Machines

The underlying principles of cutting wood surfaces in the planer also are used in the jointer, moulder (sticker) and flooring machine. The following explanation of these principles in the planer, applies also to these other types of machines. See p. 22–37 for the forms, shapes and methods of attachment of knives.

**CUTTING MECHANISM.**—Fig. 65 shows a section of a typical planer cutting mechanism. The cutting head revolves against the infeed of the stock, and the grooved *infeed rolls* should be powerful enough to withstand the double load of feeding stock and holding it against the action of the cutters. Infeed rolls often are made in sections to bear more uniformly on thick and thin stock. Large single surfacers and all double surfacers have two pairs of infeed rolls. Pressure of the feed rolls tends to straighten crooked lumber, and the ordinary planer cannot be relied upon to remove wind and warp entirely. A special type (straitoplane) is used for this purpose. See p. 22–42. The *chip-breaker* prevents lifting and tearing of the grain, and reduces all chips to a size that will not clog the cutting head, nor be difficult to move as shavings in the dust collector system. The steel tipped chip-breaker shoes ride the top surface of the incoming lumber, and are substantially concentric with the cutter head. They are suspended from a sturdy cross bar, as a heavy cut places a severe strain on chip-breaker mechanism. The smooth *outfeed roll* and the *pressure bar* require exact adjustment to the final planed thickness. They must be rigid enough to prevent any chatter or end clipping in the board during the planer cut.

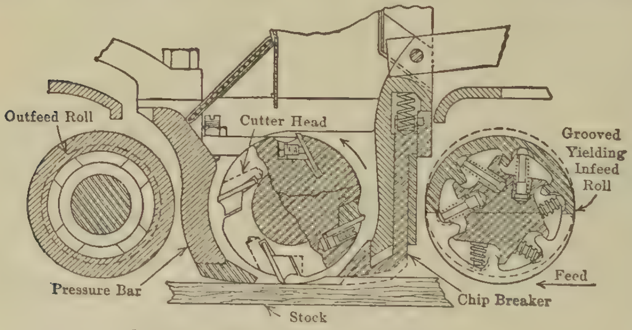


Fig. 65. Planer Cutting Mechanism.

The *cutter head* in Fig. 65 is of the round type with four thin knives, clamped in slots. While it is possible to set the several knives in approximate alignment, it is necessary to joint or grind all knives, at operating speed, to a true cutting line, after they have been set and clamped in the head. This preferably is done with a special motor-driven traveling grinder mounted over the main bearings of the planer. In some cases (as an enclosed lower knife head), the head is partially or wholly removed, with the knives clamped in place and ground in that position.

*Planer cutting circles* usually are about 6 in. diameter but vary with the different makers and shaft lengths. Since the standard speed of planer shafts usually is 3600 r.p.m. (7200 when a frequency changer is used), it is important that they be in perfect mechanical and static balance, requiring careful attention to the individual weights of knives and clamping devices. One of the most difficult problems of planer design is to overcome shaft whip or deflection at high speeds, especially in the wider machines, tending to produce uneven thicknesses and rough planing.

Proper *cutting angles* are important. They vary with the species and dryness of the wood. Table 15 gives their dimensions.

**Knife Cuts per Inch.**—A definite relation exists between the rate of feed, the peripheral speed of the cutter and the number of knives in the shaft. At best, any planer cut is a series of miniature waves in the planed surface, the degree of smoothness depending on the shortness of the individual waves. This relation is clearly indicated in Table 16.

**TYPES OF PLANERS.**—The principal types of planing machines are: 1. General-purpose planers; 2. Cabinet planers, for high quality work; 3. Mill-type planers, for surfacing full length boards.

**General-purpose Planers,** ordinarily used in so-called planing mills and job shops, have a bed (thickness) adjustment in vertical ways by means of two geared screws. With

**Table 15.—Cutting Angles and Knife Cuts per Inch for Planers and Similar Machines**

(Copyright, S. A. Woods Machine Co., Boston)

| Kind of Wood | Dry | | Air Dry | | Wet | | Green | |
|---|---|---|---|---|---|---|---|---|
| | Cutting Angle | Knife Cuts per in. | Cutting Angle | Knife Cuts per in. | Cutting Angle | Knife Cuts per in. | Cutting Angle | Knife Cuts per in. |
| Cedar | 20–25 | 8–12 | 25–28 | 7–10 | 28–35 | 7–10 | 28–35 | 7–10 |
| Redwood | 20–25 | 8–12 | 25–28 | 7–10 | 28–35 | 7–10 | 28–35 | 7–10 |
| Cypress | 20–25 | 8–12 | 25–28 | 7–10 | 28–35 | 7–10 | 28–35 | 7–10 |
| White Pine | 15–20 | 9–13 | 20–25 | 9–12 | 28–35 | 8–12 | 28–35 | 8–12 |
| Spruce | 15–20 | 9–12 | 20–25 | 9–12 | 28–35 | 8–10 | 28–35 | 8–10 |
| Norway Pine | 9–18 | 9–12 | 18–23 | 9–12 | 23–25 | 8–10 | 25–28 | 8–10 |
| Hemlock | 9–18 | 9–12 | 18–23 | 9–12 | 23–25 | 8–10 | 25–28 | 8–10 |
| Douglas Fir | 9–18 | 9–12 | 18–23 | 9–12 | 23–25 | 8–10 | 25–28 | 8–10 |
| Larch | 9–18 | 9–12 | 18–23 | 9–12 | 23–25 | 8–10 | 25–28 | 8–10 |
| Poplar | 9–18 | 9–12 | 18–23 | 9–12 | 23–25 | 8–10 | 25–28 | 8–10 |
| Yellow Pine | 9–13 | 9–12 | 13–18 | 8–11 | 18–23 | 7–10 | 23–28 | 7–10 |
| Gum | 9–13 | 10–16 | 9–18 | 10–16 | 18–25 | 9–12 | 25–28 | 9–12 |
| Oak | 9–13 | 10–14 | 13–18 | 10–14 | 25–28 | 10–12 | 25–28 | 10–12 |
| Ash | 9–13 | 10–14 | 13–18 | 10–14 | 25–28 | 10–13 | 25–28 | 10–12 |
| Mahogany | 4–9 | 12–14 | 9–13 | 12–14 | 13–18 | 10–13 | 18–25 | 10–13 |
| Beech | 4–9 | 12–14 | 9–13 | 12–14 | 13–18 | 10–13 | 13–18 | 10–13 |
| Birch | 4–9 | 12–14 | 9–13 | 12–14 | 13–18 | 10–13 | 13–18 | 10–13 |
| Maple | 4–9 | 12–15 | 9–13 | 12–14 | 13–18 | 12–14 | 13–18 | 12–14 |

Above figures apply to stock 1/2 in. thick or thicker. For stock less than 1/2 in. thick, increase the knife cuts per inch 10–50%.

To prevent tearing out grain on dry stock, decrease the cutting angle or increase the knife cuts per inch.

To prevent roughing up or raising the grain on wet or green stock, increase the cutting angle and decrease the knife cuts per inch.

**Table 16.—Knife Cuts per Inch with 3500 R.p.m. Spindle Speed**

(R.p.m. × Number of Knives)/(Ft. per Min. × 12) = Cuts per inch

| Feed of Work, ft. per min. | Number of Knives per Head | | | | | | | | Feed of Work, ft. per min. | Number of Knives per Head | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 6 | 8 | 10 | 12 | | 4 | 6 | 8 | 10 | 12 |
| 15 | 19.4 | ...... | ...... | ...... | ...... | ...... | ...... | ...... | 160 | 7.3 | 10.9 | 14.6 | 18.2 | 21.8 |
| 20 | 14.6 | 29.2 | ...... | ...... | ...... | ...... | ...... | ...... | 170 | 6.9 | 10.3 | 13.8 | 17.1 | 20.6 |
| 25 | 11.7 | 23.4 | ...... | ...... | ...... | ...... | ...... | ...... | 180 | 6.5 | 9.7 | 13.0 | 16.2 | 19.4 |
| 30 | 9.7 | 19.4 | 29.2 | ...... | ...... | ...... | ...... | ...... | 190 | 6.1 | 9.2 | 12.2 | 15.3 | 18.2 |
| 35 | 8.3 | 16.6 | 25.0 | ...... | ...... | ...... | ...... | ...... | 200 | 5.8 | 8.7 | 11.6 | 14.6 | 17.4 |
| 40 | 7.3 | 14.6 | 22.0 | 29.2 | ...... | ...... | ...... | ...... | 220 | ...... | 7.9 | 10.6 | 13.2 | 15.8 |
| 45 | 6.5 | 13.0 | 19.4 | 26.0 | ...... | ...... | ...... | ...... | 240 | ...... | 7.3 | 9.7 | 12.1 | 14.6 |
| 50 | 5.8 | 11.6 | 17.5 | 23.2 | ...... | ...... | ...... | ...... | 260 | ...... | 6.7 | 9.0 | 11.2 | 13.4 |
| 60 | ...... | 9.7 | 14.6 | 19.4 | 29.2 | ...... | ...... | ...... | 280 | ...... | 6.2 | 8.3 | 10.4 | 12.4 |
| 70 | ...... | 8.3 | 12.5 | 16.6 | 25.0 | ...... | ...... | ...... | 300 | ...... | 5.8 | 7.8 | 9.7 | 11.7 |
| 80 | ...... | 7.3 | 10.9 | 14.6 | 22.0 | 29.2 | ...... | ...... | 325 | ...... | ...... | 7.2 | 9.0 | 10.8 |
| 90 | ...... | 6.5 | 9.7 | 13.0 | 19.4 | 26.0 | ...... | ...... | 350 | ...... | ...... | 6.7 | 8.3 | 10.0 |
| 100 | ...... | 5.8 | 8.7 | 11.6 | 17.5 | 23.2 | 29.2 | ...... | 375 | ...... | ...... | 6.2 | 7.8 | 9.3 |
| 110 | ...... | ...... | 7.9 | 10.6 | 15.9 | 21.2 | 26.5 | ...... | 400 | ...... | ...... | 5.8 | 7.3 | 8.7 |
| 120 | ...... | ...... | 7.3 | 9.7 | 14.6 | 19.4 | 24.3 | 29.2 | 425 | ...... | ...... | ...... | 6.8 | 8.2 |
| 130 | ...... | ...... | 6.7 | 9.0 | 13.5 | 18.0 | 22.4 | 27.0 | 450 | ...... | ...... | ...... | 6.5 | 7.8 |
| 140 | ...... | ...... | 6.2 | 8.3 | 12.5 | 16.6 | 20.8 | 25.0 | 475 | ...... | ...... | ...... | 6.1 | 7.4 |
| 150 | ...... | ...... | 5.8 | 7.8 | 11.7 | 15.6 | 19.4 | 23.4 | 500 | ...... | ...... | ...... | 5.8 | 7.0 |

this construction a slight rocking motion is unavoidable, when stock is entering the infeed or leaving the outfeed rolls, which may be unimportant in many rough planing operations that do not require precision. Since the average cost of this type is less than that of the cabinet planer, it usually lacks the better attachments and more accurate adjustments of the latter.

**Cabinet Planers** owe much of their precision to bed adjustment by motor-actuated sliding wedges, giving both accuracy and rigidity. The range of adjustment is less than that of the general purpose planers, but tendency to rock is entirely eliminated. In cabinet planer design, the point of the chip-breaker is placed as close as possible to the actual point of cutting, to reduce the size of chips, and lessen the tendency to tear out ahead of the cut. The chip-breaker usually is of sectional or flexible construction, to ride equally well on thick, thin and rough spots in the stock. High-grade planer work demands a large number of knife cuts per inch, which is obtainable either by increasing the number of knives (up to six, rarely more) in the head, or stepping up the speed of the shaft, a point on which opinion (1937) is divided. Straight knives generally are used, but spiral knives occasionally are used to distribute the load and reduce pull-outs at the back end of the stock.

**Motor Equipment** for planers varies with the service. Standard power ratings for single surface cabinet planers are:

| Planer width, in. | 26 | 30 | 36 | 40 | 44 | 50 | 60 | 64 |
|---|---|---|---|---|---|---|---|---|
| Horsepower | 10–15 | 15–20 | 20–25 | 20–25 | 20–25 | 20–25 | 25–30 | 25–30 |

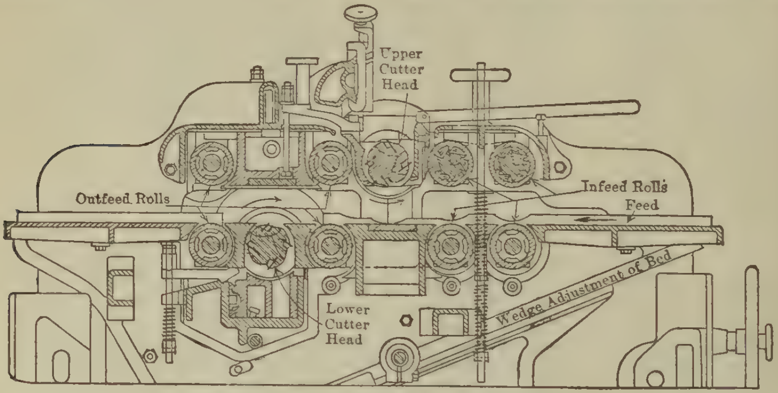


Fig. 66. Feed Mechanism of Cabinet Planer

**Feed Mechanism** may be driven direct from the motor on the cutter shaft by belt, often a severe handicap when using frequency changers, or by separate multiple-speed constant-torque motors. Speeds usually are up to 60 ft. per min., occasionally 100 ft. per min., with considerably slower speeds for exacting work.

Production cabinet planers often are fitted with *hopper feed*, the infeed bed or table being equipped with an endless rubber-faced conveyor chain, with slats or side rollers for random lengths. Hopper feeds usually are set on an angle to feed pieces diagonally.

Cabinet planers are made for double surfacing, the lower cutter shaft then being arranged to be pulled out for adjustment and jointing. See Fig. 66. Double surfacing planers usually are built from 30 to 44 in. wide with larger power requirements than single surfacers.

**The Straitoplane** is an unusual type of double surfacer in which the lower cut is made first. The freshly-cut under surface of the board then is advanced on to the rigid center bed, against which a second cut is made by the upper cutter head. During the first cut, the stock is held by an overhead flexible-fingered feed-chain, thus providing one true surface. The second cut can be adjusted for accurate thickness. This machine operates efficiently on stock as short as 10 in., and is particularly adapted to furniture work where most pieces are relatively small. Each cutter head is driven by a 15-Hp. motor at 3500 r.p.m. A 5-Hp., 2-speed motor drives the feed. The machine is standard at 36 in. wide. Feeds up to 60 ft. per min. are practicable.

**Mill Planers**, Fig. 67, primarily are designed for full-length lumber. They have been developed, and are principally used, on the Pacific Coast. Feed mechanism comprises endless chains with cross slats of ribbed cast iron, giving a more powerful and positive feed than is possible with rolls, but lacking somewhat in accuracy of cut. The cutter head is above, with the adjustments of a cabinet planer. Bed adjustment is of the sliding wedge type. Capacity is stock 8 in. thick, 30 to 36 in. wide. Feeds of 100 ft. per min. are practicable.

**JOINTERS** are classified as:

**Standard Jointers.**—The cutting mechanism of the jointer is quite similar to that of the planer, but the feed mechanism is omitted, since edged jointing is the predominant function of the jointer. Jointers are provided with a high *fence* (side gage) to permit them both to square and true the edge. Portable bench jointers are made and extensively used in preference to the hand plane. Jointers seldom are made over 16 in. wide, and sometimes are hand fed to take the wind out of dimension stock. This is particularly important in producing a true bed surface for moulder or sticker work.

**Lumber Jointer.**—The continuous feed lumber jointer has an automatic return feed for edge jointing strips for plywood core stock. This is a duplex or return cycle machine, with an endless feed-chain in the center, mounted on vertical shafts. The two cutter heads revolve in opposite directions. A board to be jointed on both edges is placed edgewise at the infeed end against the feed-chain, jointed on the lower edge, passed over a tipping mechanism that turns the unjointed edge down and into the reverse feed track. The board travels back to the infeed end, moved by the opposite side of the feed-chain, passing over the second cutter head. This machine is suitable only for use on ripped stock, as the cut is insufficient for rough or waney-edge stock, and it will not produce uniform width.

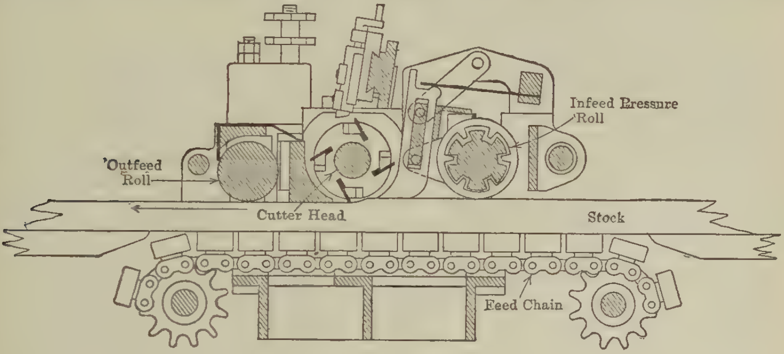


FIG. 67. Mill Planer

**Veneer Jointers** are highly specialized jointing machines. They operate on a flat *package* of sheets of veneer, aggregating 2 in. thick, which must be clamped firmly during the entire cutting period. The cutter head, usually mounted on a vertical shaft, carries as many as 12 knives to give the smooth edge necessary for matching high-grade veneer faces. In some machines the clamped package travels past the revolving knife head. In others, the clamp is stationary, and the head travels.

**MOULDERS OR STICKERS** essentially are four-sided planers, with two horizontal and two vertical shaft cutter heads, usually square 4-side heads (see Fig. 54*c*). The feed may be by pressure rolls against a rigid bed, or by endless feed-chain below. Cuts frequently made are irregular and curved faces on mouldings. Cutter heads so far as possible, have chip-breakers and the other auxiliary devices of the planer. Since all cutter shafts are short, knives and cutter heads are bolted on ends of shafts, and it is possible to use high speeds with frequency changers, up to 7200 r.p.m. Fewer knives are required at high speeds to give the same number of knife cuts per inch. This is a distinct advantage in grinding special knives as most moulder heads are 4-sided, and the uniform setting of 4 irregularly shaped knives is most difficult. It is usual to operate a moulder head at high speed with two opposed knives of the same shape and a correspondingly greater rate of feed. For long runs on standard shapes, solid cutter heads often are used, with knives jointed in a profiling machine.

Modern moulders have brackets for *cope heads*, which make re-entrant cuts that are not possible with regular heads. They also often supplement the regular heads by making deep cuts that would require impractical or very costly knives. Moulders are close coupled and compact, usually with individual motors for each head and for the feed. A popular size takes stock up to 4 × 4 in., but they are made up to 16 × 6 in. for heavy timbers.

**Flooring Machines** are specially designed moulders, the two vertical shafts making the tongue and groove, while the lower horizontal head makes the flooring face, and the upper head cuts the hollow back. A steel lettering wheel often grades and trademarks the back as it leaves the machine. Flooring machines are designed to operate on full-length lumber and at high speeds.

## Shaping Machines

**THE SHAPER** is one of the oldest types of machines. It still is important in woodworking, although many of its functions have been absorbed by more specialized machines. It is a versatile machine for small shops, as it is more easily set up for short runs than many of the specialized machines that are more efficient for long runs.

The shaper consists essentially of a large flat metal table with a vertical spindle, driven from below, extending up through the center of the table. Revolving cutter heads, mounted on the end of the spindle, accommodate a variety of knives for a wide range of cuts. (See Shaper Knives, p. 22–37.) Solid heads may be mounted for long runs. A job to which a shaper is best adapted is the shaping of the edges of table tops, especially those of other than rectangular form. The table top blank is firmly mounted, by two or more points, on a jig of the desired shape. A spindle collar is mounted at the machine table level, above which the knives are set and clamped for the required shape, ogee or otherwise. The operator revolves the combined jig and blank, holding it snug against the collar for one complete revolution of the jig, and thus producing the ogee edge around the periphery. Shaper knives cut with the grain rather than against it, as far as possible. Slow feeds are necessary when cutting at right angles to the grain. Blanks may be reversed on the jig to help this condition.

The shaper is especially useful in re-entrant curves, octagons and other odd contours not suited to machines designed for rectilinear or circular cutting. Small moulding, flutings, both straight and curved, and many edge and end cuts can be made on a shaper.

Shapers frequently are equipped with two spindles, usually rotating in opposite directions. All cuts around an irregular table top then can be made in a direction favorable to the grain, and no cuts have to " dig into the grain," leaving a rough and torn edge. The jig, with mounted work, is moved from spindle to spindle according to grain requirements. Automatic shapers are equipped with mechanically-operated jigs or revolving tables to which blanks are held with pneumatic clamps. These are suited only to long production runs.

## Routers

**THE ROUTER** is a relatively modern machine, achieving its effectiveness with the development of high spindle speeds. It is somewhat the reverse of the shaper, and usually is designed for considerably lighter cuts. The cutting head is above the table, which is provided with an upwardly projecting, jig-guiding round plug of the same diameter as, and directly under, the cutter. The cutter head carries a router bit (see p. 22–38) rotating at speeds of 20,000 r.p.m. or higher. The table may be moved up to the bit, or the bit lowered into the work. The router is admirably adapted to cutting out (with grooved cuts) grills or irregular shapes. It works much better and more quickly than jig saws. Router bits may have shoulders for shaping the edges of cut-outs, or T-heads for slotting. Straight cuts on narrow strips can be made by clamping suitable gages to the table. A straight gage, through or near the bit center, permits shaping of straight-edge table tops and similar work. Double-spindle routers are available with separate tables, facilitating work requiring two types or sizes of grooves on the same job.

Router motors, both electric and pneumatic, must be very accurately balanced for excessively high speeds. Brakes are provided to slow down the bits, which otherwise would require a long time to come to rest. Blowers, directed to the exact point of cut, are standard equipment to prevent chips and shaving from clogging the cut and heating the bits. Some routers are equipped with tilting tables. In these the bit is lowered into the work. Motors are 1 to 5 Hp. for heavy duty. Spindle speeds often are stepped up by V-belts.

**Portable Hand Routers** are built to operate electrically and pneumatically. Electric machines operate at 10,000 r.p.m. maximum; pneumatic, at 60,000 r.p.m. maximum.

## Carving Machines

**CARVING MACHINES** are of two main types, single and multiple.

**Single Carvers** may be rigidly mounted on a floor standard, the work being moved free-hand by the operator. The single carver also is available in the balanced suspension type, hung much like a dentist's drill. The work is held stationary and the drill moved over the pattern by the operator. In both types the mechanism consists of a small high-speed motor with a small spur drill mounted directly on the motor shaft. Single carvers are used for a certain amount of undercutting. Final retouching usually is by hand.

**Multiple Carvers** may carry as many as 24 blanks, all to be roughed out uniformly from a master pattern or model, usually of metal. The spur cutters are moved in unison by link levers that are fixed in their relation to each other. Cutters usually are belt-driven from one or more motors. Individual motors can be used for extra heavy cuts. Such machines can only rough out the carvings, but their efficiency consists in removing all surplus wood, even on deeply recessed carvings, leaving a minimum of hand carving of finer lines and under cuts. This type of machine has greatly extended the use of carving on classical furniture at very reasonable cost. But one operator is required, who guides the blank cutter over the pattern, carefully watching the direction of the cut to avoid tearing the wood. The hand finishing work reproduces the character of hand carving. This process is displacing pressed and overlaid imitation carved work.

## Boring Machines

**BORING MACHINES** range over a wide variety of woodworking machines, from a single bit held in clutch jaws on the end of a motor shaft, to gang or multiple machines in various arrangements, sometimes equipped for as many as 30 bits. Some bore only vertical holes, others bore horizontally, and still others will bore at various angles, depending on the complexity of the work. The maximum number of individual bits seldom exceeds 30, but special machines have been made for a large number of boring problems. The literature of the industry should be consulted for more specific information.

**Vertical Multiple Borers** carry a series of fixed or adjustably positioned spindles, usually arranged in approximately single or double vertical ranks, with a clamp-equipped work table moving up the bits. Spindles may be fixed or *rigid, i.e.*, maintain a vertical alignment with the splined drive shaft above, or *adjustable* to points at quite a horizontal distance from the plane of the drive shaft, by a telescopic universal joint (see Fig. 68).

**Horizontal Multiple Borers** differ from vertical borers principally in boring a series of horizontal holes, having a work table that moves in and out. They are used mostly for end boring. Universal-joint spindles are not generally adaptable to horizontal boring. However, adjustable multiple or cluster boring heads, to which bits are fixed in position in the head, are available.

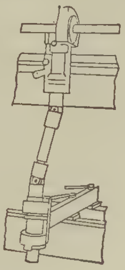
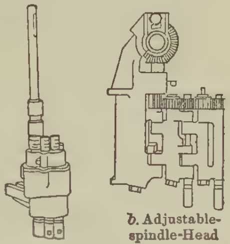


Fig. 68. Boring Mill Spindle with Universal Joint



*a*. Fixed Center Cluster

*b*. Adjustable-spindle-Head

Fig. 69. Multiple Boring Heads

**Cluster Vertical Borers** (not made in horizontal types) have universal-joint spindles, adjustable over rectangular or circular areas up to 40 in. diameter, with many combinations of bits that may be attached. Motors are above, and operate the bits through gear trains which terminate in fixed sockets for spindle connections. The universal-joint spindles permit a wide range of adjustment of bits in any horizontal direction within operating limits.

**Multiple Boring Heads** are principally: 1. Fixed cluster (Fig. 69 *a*); 2. Adjustable multi-spindle (Fig. 69 *b*). Many other types of heads are made, for use in both vertical and horizontal boring machines.

**Standard Spindle Speeds** range from about 1000 to 2000 r.p.m., varying principally with the size of hole, hardness of the wood, and the rate of feed.

## Turning Machines

**LATHES** differ widely in basic principles, developing by degrees from the plain hand lathes, which is one of the oldest woodworking machines and suited to a wide range of individual work. The principle of the high-speed cutter head has been borrowed from the shaper and adapted to many lathe cuts, especially for duplicating work. Patterns and cams have been adapted from such machines as the carver, and other devices, for various other types of cut. Only the principal types are described below.

**The Copying Lathe** works entirely from a model or pattern of the exact shape required. It can be used for gun stocks, axe handles, shoe lasts, parts of period furniture, etc. The blank to be turned and the pattern are mounted on slowly rotating gear-driven centers. The cutting member is either a high-speed revolving cutter head, or occasionally a heavy saw. Its movement is controlled by a tracing pin, or roller which traverses all portions of the pattern, producing an exact duplicate. Usually the cutter moves along the length of the blank, but in axe handle lathes, the frame, with centers, blank and pattern, move past the cutter head.

**Back Knife Lathes** first turn the blank to a rough round by a ring or die with insic cutting edges, supported on a traveling carriage. The die is moved out of the way, ove the dead center, for the turning operation. The long back knife is so set, milled to such shape, that when in an angling cutting position against the blank, it gradually cuts in, an is stopped when the final turning is of the correct size and contour. The back knife neve is the exact reverse of the turning, but a mechanically developed extension of it.

**The Shaping Lathe** is an exceedingly versatile machine, turning rounds and polygona sections in almost every conceivable combination. Both lathe centers are "alive," an oscillate slowly into and out of the cutting zone, under the operator's control. Cutte knives, similar to a series of shaper heads on a single shaft, are mounted on a cutter spindl or arbor in the rear, and revolve at high speed. All knives are mounted on the arbo whether for square, round or hexagon cuts. As a general rule, where squares and round are on one turning, flat surfaces are cut first, the lathe centers turning through a small arc and keeping the work tangential to the revolving cutters by a cam. The piece to b turned is withdrawn from the cutters, turned through approximately a 90-degree angl (for a square), and the next flat surface is cut. The angle of movement controls th number of sides for squares, hexagons, octagons, etc. After the flats are completed, th rounds are turned, with lathe centers slowly and uniformly rotating. Various modifica tions may be made in this procedure for complicated types of turning or shaping piece other than turnings. The set-up of the cutter head may be so complex as to demand lon production runs for good economy.

Many different types of lathes are built in addition to the major types described above The literature of the industry should be consulted for detailed information.

## Mortising Machines

**MORTISING MACHINES** may be classed as: 1. Hollow chisel mortisers; 2. Chai saw mortisers; 3. Oscillating bit mortisers.

**Hollow Chisel Mortisers** usually provide a vertical movement of the tool into the work Horizontal types also are available. The tool (see p. 22–39) is attached directly to th motor, the rotating bit being motor-driven; the surrounding non-turning chisel is mounte in the chisel carriage on the motor frame, and both are forced together into the work. Th tables are equipped with clamps, adjustments and multiple end-gages, as several mortise may be made in the same piece. Tool strokes may be arranged automatically to repea cuts for oblong holes, up to about 50 per minute. Horizontal movement of the table als may be automatic for a series of mortises. Motor sizes range up to 4 Hp. at 3600 r.p.m Multiple hollow chisel mortisers always are horizontal stroke, limited to six heads, with minimum of 4 in. center spacing, and a maximum spacing of 72 in. between outer heads

**Chain Saw Mortisers** operate with cutting teeth mounted on the outside peripher of a link chain. The minimum size of hole is $^3/_{16} \times {}^3/_4$ in.; the maximum may b $1\,^1/_2 \times 4$ in. Longer mortises may be cut at one stroke by using a double roll bar, or b successive cuts. The table moves the work up into the chain cutters. Sprocket motor usually are 3600 r.p.m., giving the following approximate chain cutting speeds:

| | | | |
|---|---|---|---|
| No. of sprocket teeth . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . | 5 | 7 | 9 |
| Chain speed, ft. per min . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . | 1330 | 1930 | 240 |

Multiple chain saw mortisers are available up to 8 heads, which may be stationary, o adjustable in and out as well as sideways. Chain saw mortisers are not made in hor zontal types.

**Oscillating Bit Mortisers** make round-end holes, whose depth is limited by the siz of the router-type bit. The bit oscillates between stops set for the desired length Oscillations seldom should exceed 200 per min.; bit speeds are up to 7200 r.p.m. Oscilla ing motor usually is 1 Hp.; bit motors may be 4 Hp. Air clamps hold stock on the tables.

## Double-end Tenoners

**DOUBLE-END TENONERS** are highly developed machines that accomplish a wid variety of cuts at one passage of the work. Their functions have been broadened and in creased by individual motors on each cutting head, which are more flexible than the bel driven heads of earlier machines. This machine originally was intended to cut tenon simultaneously on both ends of a piece of stock. Cutting heads of various types graduall wereadded for a diversity of end and intermediate cuts.

In essential principles, the double-end tenoner consists of a sole (or bed) plate carryin two pedestals. One pedestal is stationary, and carries all operating switches. The othe may be moved along the sole plate to accommodate the work, limited in standard machine to about 90 in. Between the two pedestals, and below the work, a pair of endless feec chains, with adjustable lugs, carry the work into the saws and other cutting units. The feed-chains are kept relatively close to the pedestals, but are adjustable along a spline feed shaft between them. They are fixed in the horizontal plane, all vertical adjustments being made with the cutter heads. Two idling pressure chains hold the work firmly, operating directly above the feed chains.

For double cut-off and double rip cuts, as in plywood, only one saw is used in each pedestal. This cut is made from above. On modern double-end tenoners, the main cutter head can be tilted to make angle cuts. One or more additional horizontal (sometimes tilting) cutter heads are located on each pedestal to make end tenons, grooved, shoulder or shaper cuts on either or both sides of the stock, and at one or both ends. Each pedestal also carries a vertical cutter head. An arbor can be placed between pedestals for intermediate saws or cutters. Cope units, dado heads, relishing cutters, etc., if required, may be attached in various ways. Jump dado cams can be used to enter and withdraw dado cutters near the edge of the stock. Dovetailing attachments may be used for either male or female cuts. Methods of adjustment, attachment of various cutter heads, and other operating features vary considerably in the product of the several machine builders. The variety of work is limited only by the available cutter heads and the extent to which production runs justify elaborate set-up.

Single-end tenoners, with corresponding attachments, are available for short runs. Stock is fed on a roller feed carriage, with the cutting units at the left of the operator.

## Sanders

**POWER-FEED DRUM-SANDERS** are of two general types: 1. Work is forwarded by a rubber-cushioned endless-belt bed; 2. Work is moved ahead by pairs of pressure rollers. Widths range from 31 to 103 in. Generally, the roll-feed type sands one side only, giving thinner and more accurate surface removal as in figured faced veneers on plywood; two-side types are available, if there is plenty of surface to be removed, and accurate centering between surfaces is unimportant. Endless-bed sanders sand one side only.

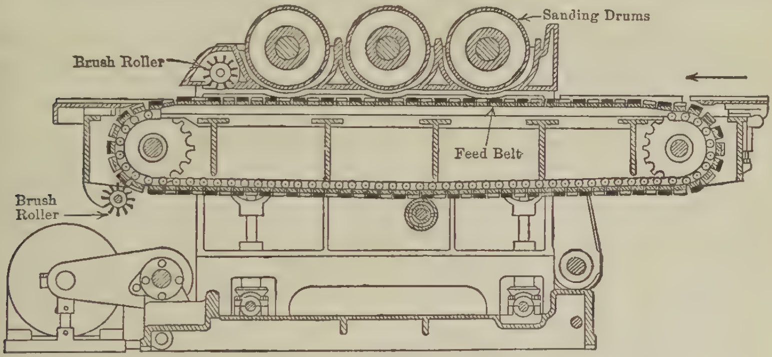


Fig. 70. Endless Bed Sander

**Endless-bed Sanders.**—In the machine shown in Fig. 70, the sandpaper drums, usually three, are on top. They are approximately 12 in. diameter, covered with felt cushion and sandpaper, sometimes wound spirally, but more often attached by clamps in slots. The feed-belt, with rubber cushions, is below. Surface speed of the first two drums usually is 3700 ft. per min. (1200 r.p.m.); that of the third drum is 5600 ft. per min. (1800 r.p.m.). Drums revolve against, rather than with the work. The work is fed at speeds ranging from 12 to 24 ft. per min. The coarsest sandpaper is on the first roll and the finest on the third roll. Each roller and the feed mechanism is driven by an individual motor. Motor sizes, depending on machine width, are: First and second rollers, 5 to 10 Hp.; third roller, 7 1/2 to 15 Hp.; feed mechanism, usually, 3 Hp. A brush roller frequently is installed at the delivery end to remove sandpaper dust. Standard drums oscillate slightly endwise (75–100 per min.), to simulate the side movement of hand sanding.

**Single-surfacing Roll-feed Sanders.**—The drum arrangement is similar to that of endless-bed sanders, except that drums are below the work, and sand through slots in the feed bed. An idler-pressure roller is above (see Fig. 71). Pairs of smooth feed rollers are located at the entering end, on both sides of the center drum, and at the discharge end. The roll type is not well suited to small dimensioned stock. The sandpaper drums are not so conveniently located for paper changing as in the endless-bed type.

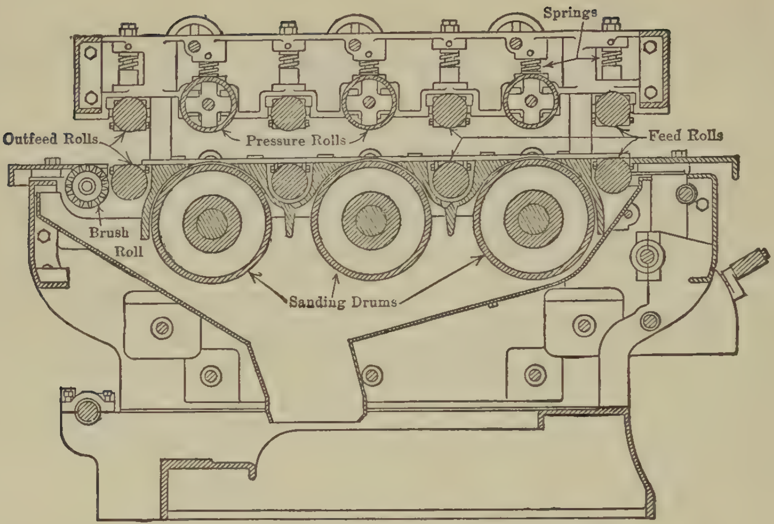


Fig. 71. Single-surfacing Roll-feed Sander

**Two-side Roll-feed Sanders.**—The upper and lower drums are directly over each other in pairs, with the same arrangement of pairs of feed rollers as in the single surface type.

**Belt Sanders** have a 6- to 8-in. endless sandpaper belt running clockwise on two 18-in. pulleys, 8 to 12 ft. apart. In the overhead type the work table, moving on ways toward and away from the operator, is below the lower belt, and the smooth side of the paper is against the pulley. In the underslung type, the table is between the belts, with the sand side of the paper against the pulley. In both types, the work is held on the table by stops at the left and an edge clamp at the right. The operator bears down on the cushion pad, hand operated or mounted on two-way sliding levers. The pad is above and applies controlled pressure through the moving sand belt to sand the stock. Belt sanders are preferred to drum sanders for work that requires a thin cut, as on plywood faces.

**Stroke Sanders** usually are used for polish and finish sanding. They are somewhat similar to belt sanders, except that the pad is mechanically reciprocated under weights. The same type of stroke machine is used for rubbing finish, the sandpaper then being omitted, and a rubbing compound used.

**Automatic Belt Sanders** for sanding mouldings consist of driven rubber feed rolls which carry the stock under a sandpaper belt, and a flexible shoe shaped to the reverse of the moulding contour. Feed is 32 to 50 ft. per min.

**Specialized Types of Sanding Machines** are: Belts on vertical pulleys for edge sanding; sand discs on rotating horizontal or vertical tables for drawers and boxes; sand drums of large diameter for curved work; sand arbors for chair legs and irregular pieces, etc. A small portable motor-driven belt sander is available for bench use.

**MANUFACTURERS** of woodworking machinery include the following (for manufacturers of saw mill machinery, see p. 22–14): E. C. Atkins & Co., Indianapolis, Ind.; Bell Machine Co., Oshkosh, Wis.; Buss Machine Works, Holland, Mich.; G. M. Diehl Machine Works, Wabash, Ind.; Henry Disston & Sons, Inc., Philadelphia, Pa.; Ekstrom, Carlson & Co., Rockford, Ill.; Charles E. Francis Co., Rushville, Ind.; Greenlee Brothers & Co., Rockford, Ill.; Hermance Machine Co., Williamsport, Pa.; Jenkins Machinery Co., Sheboygan Falls, Wis.; Leitelt Iron Works, Grand Rapids, Mich.; Mattison Machine Works, Rockford, Ill.; Mereen-Johnson Machine Co., Minneapolis, Minn.; Merritt Engineering & Sales Co., Lockport, N. Y.; J. M. Nash Co., Milwaukee, Wis.; Oliver Machinery Co., Grand Rapids, Mich.; Onsrud Machine Works, Inc., Chicago; C. O. Porter Machinery Co., Grand Rapids, Mich.; B. M. Root Co., York, Pa.; Tannewitz Works, Grand Rapids, Mich.; Vonnegut Moulder Corp., Indianapolis, Ind.; Baxter D. Whitney & Son, Winchendon, Mass.; Wilkin-Challoner Co., Oshkosh, Wis.; S. A. Woods Machine Co., Boston; Yates-American Machine Co., Beloit, Wis.

**BIBLIOGRAPHY.**—Modern Woodworking Machinery, G. F. Cosgrove, Owosso, Mich., 1936; Proceedings of Wood Industries Division, *Trans.* A.S.M.E. 1921–1938; Timber Products and Industries, N. C. Brown (John Wiley & Sons, 1937); Wood Handbook, Forest Products Laboratory, Madison, Wis. (1935); Woodsman's Handbook, H. S. Graves and E. A. Ziegler (Government Printing Office, 1912); Veneers and Plywood, Knight and Wulpi (Ronald Press, 1926); Air Seasoning and Kiln Drying of Wood, H. L. Henderson (J. B. Lyon, Albany, N. Y., 1936).

# Section 23

# MATERIALS HANDLING

By Frederic V. Hetzel

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# MATERIALS HANDLING

By Frederic V. Hetzel

## HOISTING

### 1. MANILA ROPE HOISTING

See pp. 6-26 to 6-30 for allowable stresses, etc., of manila rope. Table 1 gives results of tests of manila rope hoists made in 1903 on a testing machine at Pencoyd Iron Works. The blocks were large, with Metaline bushed sheaves. The lead line was run through snatch blocks to simulate the course of a hoisting rope on a derrick from end of boom to engine drum. Column 5 shows overall efficiencies under best conditions. Column 6 shows probable efficiencies of the hoist alone after allowances for snatch blocks and other known losses.

**MANILA ROPE HOISTING IN PRACTICE.**—New manila rope is stiffer than used rope, friction losses due to bending are greater, and hoisting efficiency is less. Table 2 gives efficiency of manila rope hoists for new rope and used rope, based on tests and experience. Values represent efficiencies of commercial tackle blocks used in pairs under everyday operating conditions, wherein blocks are not always mounted in the best way, sheaves are relatively small in diameter, ropes are not reeved right and lubrication is neglected (Western Block Co., Lockport, N. Y.; Plymouth Cordage Co., Plymouth, Mass.). In using Table 1 to determine load capacity of a hoist, note that manila rope limber enough to show higher efficiency than new rope has lost some of its strength, and will not be as safe. *Safety in a hoist is more important than efficiency.* For instance, a 1000-lb. pull on a lead line might be rated by Table 1 to lift 3400 lb. in an 8-part hoist with new rope, and 4400 lb. with used rope, but the latter would not be so safe.

Manila rope hoists are seldom used with more than 6 or 8 parts of line. Fig. 1 shows a 6-part hoist with rope properly reeved. The lower block is at right angles to the upper, lead line 1 comes off a middle sheave in one block, and becket line 7 comes off a middle sheave in the other. A 6-part hoist has 6 parts of rope at the *movable* block.

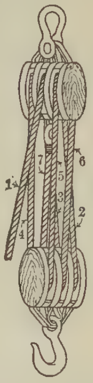


Fig. 1. Triple Tackle Block

**LOADS ON TACKLE BLOCKS.**—If lead line comes off upper block, Fig. 1, the block must carry weight lifted plus pull in the line. If Fig. 1 is reversed so that the hook block is uppermost, this block carries the weight minus pull in lead line. The reversed illustra-

**Table 1.—Efficiencies of Manila Rope Hoists in Testing Machine**

| Rope Diam., in. | Parts of Rope | Sheave | | Efficiency, Percent | | Rope Diam., in. | Parts of Rope | Sheave | | Efficiency, Percent | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Outside Diam., in. | Groove Diam., in. | As Observed | Calculated | | | Outside Diam., in. | Groove Diam., in. | As Observed | Calculated |
| 1 1/4 | 4 | 12 | 6 1/2 | 57 | 84 | 1 3/4 | 5 | 16 | 9 | 45 | 77 |
| | 6 | 12 | 6 1/2 | 52 | 75 | | 7 | 16 | 9 | 41 | 68 |
| | 7 | 12 | 6 1/2 | 49 | 70 | | 9 | 16 | 9 | 39 | 61 |
| 1 1/2 | 4 | 14 | 8 | 50 | 84 | 2 | 7 | 20 | 11 | 41 | 68 |
| | 7 | 14 | 8 | 46 | 70 | | 9 | 20 | 11 | 37 | 61 |
| | 9 | 14 | 8 | 47 | 62 | | | | | | |

**Table 2.—Efficiencies of Commercial Tackle Blocks and Manila Rope**

| Parts of Rope | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|
| New rope.................. | 81 | 73 | 65 | 59 | 53 | 47 | 43 |
| Used rope.................. | 83 | 77 | 71 | 66 | 62 | 60 | 55 |

Rope sizes and corresponding block and sheave sizes are given in Table 3.

tion shows a 7-part hoist, and lead line pull is less than in a 6-part hoist where line comes off upper block.

**MANILA ROPE BLOCKS** are made in a variety of patterns for different services, as occasional or steady, slow or rapid hoisting. Table 3 gives working loads for regular-pattern manila rope blocks used in pairs. These loads are what the blocks can continue to lift safely in daily service until sheave, pin and shell are worn out. Bronze self-lubricating bushings prolong the life of sheave and block. If sheaves for hoisting rope can be hung or mounted independently of the block it is often wise to make them larger than block sheaves. The rope will last longer.

**LOADS SUDDENLY APPLIED.**—Calculations of rope hoists assume the load to be uniformly divided among all parts of rope. With many parts of rope on multiple-sheave blocks, the distribution may be unequal if load is applied suddenly, or hoist started at full speed. In such cases, a rope may break at an apparently light load.

**Table 3.—Working Loads (Pounds) for a Pair of Manila Rope Blocks**
(Western Block Co.)

| Diam. of Rope, in. | Size of Block | Diam. of Sheave | Singles | Doubles | Triples | Diam. of Rope, in. | Size of Block | Diam. of Sheave | Singles | Doubles | Triples |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3/8 | 3 | 1 3/4 | 150 | 250 | 350 | 1 | 9 | 5 1/2 | 2000 | 2700 | 3400 |
| 1/2 | 4 | 2 1/4 | 300 | 450 | 600 | 1 1/8 | 10 | 6 1/4 | 2400 | 3100 | 3800 |
| 5/8 | 5 | 3 | 500 | 750 | 1000 | 1 1/4 | 12 | 8 | 2800 | 3700 | 4600 |
| 3/4 | 6 | 3 1/2 | 800 | 1200 | 1600 | 1 3/8 | 14 | 9 1/2 | 3800 | 4900 | 6000 |
| 7/8 | 7 | 4 1/4 | 1200 | 1800 | 2400 | 1 5/8 | 16 | 11 | 4400 | 5700 | 7000 |
| 1 | 8 | 4 3/4 | 1600 | 2300 | 3000 | | | | | | |

Extra heavy wood blocks and metal frame blocks are good for loads listed for blocks one size larger.

## 2. WIRE ROPE HOISTING

For the general theory of hoisting blocks, see p. 8–04. In practice allowance must be made for friction losses in sheaves and blocks, bending stresses in the ropes, etc.

**WIRE ROPE HOISTING.**—For wire rope, allowable stresses, fastenings and fittings, see pp. 6-34 to 6-47. Table 4 (Am. Bridge Co. Handbook for Erecting Engineers) gives lead line pull for 1000-lb. load using 6 × 19 rope on sheaves 16 in. outside diam., with Metaline bushings.

Table 4 is based on formula

$$P = Wk^n(k - 1)/(k^n - 1) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [1]$$

where $P$ = load line pull, lb.; $W$ = weight lifted, lb.; $k$ = factor from formula [2]; $n$ = number of parts of rope at movable block.

$$k = 1 + 0.225(d/R) + 0.16(r/R), \quad . \; . \; . \; . \; . \; . \; . \; . \; [2]$$

where $d$ = diam. of wire rope, in.; $R$ = radius of bending of rope on sheave, in.; $r$ = half-diam. of sheave pin, in. Both formula and table have been checked by tests and accepted by Am. Bridge Co. as practically correct. The following notes are applicable to Table 4: 1. Table is calculated for the case of the lead line coming off upper block; if it comes off lower block, divide by $k$, or increase $n$ by 1. 2. If lead line is snatched or deflected between block and hoist drum, add 8% for each 180° bend, 4% for each 90° bend. 3. Lead line should always come off a middle sheave of a multiple-sheave block, and the becket line, *i.e.*, the end of the rope that is dead-ended on a block also should lead off a middle sheave (see Fig. 1). Otherwise blocks will tip sideways, ropes will chafe, and stress in the lead

**Table 4.—Lead Line Pull Per 1000 Lb. of Load**
6 × 19 Wire Rope in Am. Bridge Co. 16-in. Blocks with 2-in. Pin

| No. of Parts $n$ | 5/8-in. Rope | 3/4-in. Rope | 7/8-in. Rope | 1-in. Rope | No. of Parts $n$ | 5/8-in. Rope | 3/4-in. Rope | 7/8-in. Rope | 1-in. Rope |
|---|---|---|---|---|---|---|---|---|---|
| 2 | 531 | 534 | 536 | 539 | 12 | 107 | 110 | 112 | 114 |
| 3 | 361 | 364 | 366 | 368 | 13 | 101 | 103 | 105 | 108 |
| 4 | 276 | 279 | 281 | 283 | 14 | 95 | 97 | 100 | 102 |
| 5 | 225 | 228 | 230 | 232 | 15 | 90 | 93 | 95 | 97 |
| 6 | 191 | 194 | 196 | 198 | 16 | 86 | 89 | 91 | 93 |
| 7 | 167 | 170 | 172 | 174 | 17 | 82 | 85 | 87 | 90 |
| 8 | 149 | 152 | 153 | 156 | 18 | 79 | 82 | 84 | 86 |
| 9 | 135 | 138 | 139 | 142 | 19 | 76 | 79 | 81 | 84 |
| 10 | 124 | 126 | 128 | 131 | 20 | 74 | 76 | 79 | 81 |
| 11 | 115 | 117 | 119 | 122 | $k$ = | 1.041 | 1.045 | 1.048 | 1.052 |

line will be greater. Dynamometer tests with a 13-part hoist lifting 60 tons showed a 20% increase in pull with lead and becket lines reeved from outside sheaves. 4. Wire rope sheaves are listed by outside diameter. Diameter at bottom of groove is about 2 in. less. 5. In lowering a load by slackening on lead line, friction and bending effects are reversed as compared with hoisting. The relation between $P$ and $W$ is found by substituting $1/k$ for $k$ in formula [1]. Hoists having many parts of line may require weight to be added to lower block to make it overhaul the lead line if latter is long or much deflected. 6. Table is calculated for Am. Bridge Co. standard blocks with 16-in. sheaves on 2-in. pins. Values of $k$ range from 1.041 for 5/8-in. rope to 1.052 for 1-in. rope. For other combinations of rope, sheave and pin, $k$ may be calculated and interpolated in Table 4. Within limits of Table 5, giving commercial sizes of wire rope sheaves for 6 × 19 rope, it is unnecessary to calculate $k$. For these sizes of sheaves in standard blocks, lead line pulls from Table 4 may be taken as correct within 5%, as qualified below.

**EFFICIENCY OF WIRE ROPE HOISTING.**—Table 6 shows percentage ratio of theoretical to actual pull (called efficiency), based on Table 4. Values in Tables 4 and 6 probably represent the best that can be obtained in wire rope hoisting. Loads in the Am. Bridge Co. tests were from 20 to 60 tons, riggers were skilled and operating conditions good. Makers' tests of tackle blocks with lighter loads show lower efficiency. Table 7 gives test results that represent average practice in wire rope hoisting.

Table 7 refers to 6 × 19 wire rope used with manufacturers' standard blocks carrying sheaves of sizes shown in Table 5 as related to rope diameters, fitted with self-lubricated bronze bushings.

Considering only life of rope, sheaves of commercial tackle blocks are of too small diameter; practically, it is inconvenient to make them larger. Some single blocks for whip or rapid cargo hoisting have construction refinements, making the blocks safer, more durable and less troublesome, and add life to the rope. Sheaves are relatively large, of manganese steel, with ball or roller bearings. Shells are cast-steel with heat-treated alloy steel pins.

**WIRE ROPE BLOCKS** are made as regular, heavy, and extra heavy, with hook (lower block, Fig. 1) or shackle (upper block, Fig. 1) connections. Shackles are used for permanently-attached blocks, and hooks for detachable blocks. Hooks fail by straightening, thus giving notice of overloading before failure. Shackles fail by shearing the pin at loads about 5 times those at which round iron hooks straighten, size for size. Hooks flattened to give more depth of cross section at the back are 10 to 15% stronger than hooks

**Table 5.—Commercial Sizes of Wire Rope Sheaves**

| | | | | | | |
|---|---|---|---|---|---|---|
| Diameter of rope, in. | 3/8 | 1/2 | 5/8 | 3/4 | 7/8 | 1 |
| Diameter of sheave, in. | 6 | 8 | 10, 12 | 12, 14 | 14, 16 | 16, 18, 20 |

**Table 6.—Percentage Efficiencies of Wire Rope, Based on Table 4 (Abridged)**

| No. of Parts | 2 | 4 | 6 | 8 | 10 | 15 | 20 |
|---|---|---|---|---|---|---|---|
| Rope, 5/8 in. diameter | 95 | 90 | 87 | 83 | 80 | 73 | 67 |
| " 3/4 " " | 93 | 89 | 86 | 82 | 79 | 71 | 65 |
| " 7/8 " " | 93 | 88 | 85 | 82 | 78 | 70 | 63 |
| " 1 " " | 92 | 88 | 84 | 80 | 76 | 68 | 61 |

**Table 7.—Percentage Efficiencies with Standard Wire Rope Block**
(Western Block Co., Lockport, N. Y.)

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Sum of No. of sheaves, both blocks | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
| Efficiency, percent | 98 | 96 | 91 | 88 | 84 | 82 | 80 | 76 | 74 | 73 | 70 | 68 |

Combination of rope diameter and sheave diameter are those listed in Table 5.

**Table 8.—Working Loads (Tons) for a Pair of Wire Rope Blocks**
(Western Block Co.)

| Regular Pattern | | | | | | | Extra Heavy Pattern | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diam. of Sheave, in. | With Hooks | | | With Shackles | | | Diam. of Sheave, in | With Hooks | | | With Shackles | | |
| | Singles | Doubles | Triples | Singles | Doubles | Triples | | Singles | Doubles | Triples | Singles | Doubles | Triples |
| 6 | 1 1/2 | 2 | 2 1/2 | 2 | 3 | 4 | 8 | 2 1/2 | 3 1/2 | 4 1/2 | 4 | 6 | 7 |
| 8 | 2 | 2 1/2 | 3 | 3 | 5 | 6 | 10 | 3 1/2 | 4 1/2 | 5 | 5 | 7 | 9 |
| 10 | 2 1/2 | 3 | 3 1/2 | 4 | 6 | 7 | 12 | 4 1/2 | 5 | 6 | 7 | 10 | 12 |
| 12 | 3 | 4 1/2 | 5 1/2 | 5 | 7 | 8 | 14 | 5 | 6 | 8 | 8 | 12 | 14 |
| 14 | 4 1/2 | 5 1/2 | 6 1/2 | 7 | 9 | 10 | 16 | 6 1/2 | 8 | 10 | 12 | 15 | 20 |
| 16 | 5 1/2 | 6 1/2 | 8 | 8 | 10 | 11 | 18 | 7 1/2 | 9 | 11 | 15 | 20 | 25 |

bent from round bars. Drop forged hooks are more generally used and are even stronger. Table 8 gives recommended working loads for a pair of wire rope blocks. Recommended working loads refer both to ultimate strength of block and its parts, and to durability and safety of block in use.

**STRESSES IN HOIST ROPES** depend on load, acceleration or retardation of load, and stresses due to bending over sheaves and drums. The factor of safety is intended to cover other items, accidental or incidental. For bending stresses, see Table 11. If a load $W$ (including weight of rope), starting from rest, is brought to velocity $V$ in distance $D$ by uniformly accelerated motion, rate of acceleration is $V^2/2D$; since $f = Wa/g$, the added pull required to bring the load up to speed is $f = WV^2/2gD$.

Example.—To accelerate a 10-ton load from rest to 20 ft. per sec. in 10 ft. requires $(10 \times 2000 \times 400)/(64.3 \times 10) = 12{,}440$ lb. plus 20,000-lb. load. The rope is under the same stress if the hoist brake is applied to reduce the speed from 20 to 0 ft. per sec. within 10 ft. of travel.

If rope slack is wound in at full speed, the sudden increase in stress when load comes on may injure or break the rope, unless it is long enough to absorb the shock by its own elasticity. Table 9 gives results of dynamometer tests on taking up slack at full speed. Speed and length of rope are not stated.

**STRESSES ON INCLINES.**—A rope pulling a load up an incline is stretched to a tension

$$T = (W + R)\sin\alpha + WF\cos\alpha + Rf\cos\alpha, \qquad [3]$$

where $W$ = weight of load, usually a car or cars; $R$ = weight of rope; $\alpha$ = angle to horizontal of incline; $F$ and $f$ = respectively, coefficient of friction of cars on track and of rope friction over rollers. Table 10 gives tension required to pull 2000 lb. up inclines of various slopes; $F$ (assumed) = 2 1/2%; $f$ = 5%.

For descending loads, $T = (W + R)\sin\alpha - WF\cos\alpha - Rf\cos\alpha$. For the added pull for acceleration or retardation on an incline substitute $T$ for $W$ in the acceleration formula. For speeds under 500 ft. per min., allow 5% to cover acceleration stress. For speeds of 500 to 1000 ft. per min., allow 10%; for higher speeds, calculate it.

**BENDING STRESSES IN WIRE ROPE.**—A standing rope bent over a thimble or eye loses some strength, as the bent wires do not all share in taking load. Such a rope shows, in a testing machine, strength of 88 to 90% that of straight rope. A standing rope bent over a sheave of a size in common use may lose 5% strength.

Running ropes have a similar loss. A greater loss comes from wire breakage due to fatigue of metal. Fatigue depends on the number of bends, that is, on velocity of the rope, and also on bending stress on outside wires. Magnitude of stress due to bending is impossible of direct calculation or estimation; various formulas give discordant results. James F. Howe (*Trans.* A. S. M. E. vol. xl, p. 1043, 1918) tested many sizes and kinds of wire rope. The data were used in a formula giving values shown in Table 11, which represent a stress to be added to direct tension to show equivalent total load on the rope.

**Table 9.—Rope Stresses in Mine Hoists—Taking Up Slack at Full Speed**

(Trenton Iron Co.)

| | Average of 2 Dynamometer Tests for Each Case | |
|---|---|---|
| Weight of cage and load | 6,375 lb. | 11,300 lb. |
| Taking up 3 inches slack | 11,200 | 19,025 |
| " " 6 " " | 12,250 | 24,000 |
| " " 12 " " | 15,675 | 26,000 |

**Table 10.—Rope Stresses on Inclined Planes in Pounds per Ton (2000 Lb.)**

| Rise, ft., per 100 ft. Horizontal | Angle of Incline | Stress Due to $(W + R) \times \sin\alpha$ | Car Friction, $WF\cos\alpha$ | Rope Friction, $Rf\cos\alpha$ | Rise, ft., per 100 ft. Horizontal | Angle of Incline | Stress Due to $(W + R) \times \sin\alpha$ | Car Friction, $WF\cos\alpha$ | Rope Friction, $Rf\cos\alpha$ |
|---|---|---|---|---|---|---|---|---|---|
| 5 | 2° 52′ | 100 | 50 | 100 | 90 | 41° 59′ | 1338 | 37 | 74 |
| 10 | 5° 43′ | 199 | 50 | 99 | 100 | 45° 0′ | 1414 | 35 | 70 |
| 15 | 8° 32′ | 297 | 49 | 99 | 110 | 47° 44′ | 1480 | 34 | 67 |
| 20 | 11° 19′ | 392 | 49 | 98 | 120 | 50° 12′ | 1537 | 32 | 64 |
| 25 | 14° 02′ | 485 | 49 | 97 | 130 | 52° 26′ | 1585 | 30 | 61 |
| 30 | 16° 42′ | 575 | 48 | 96 | 140 | 54° 28′ | 1628 | 29 | 58 |
| 35 | 19° 17′ | 660 | 47 | 94 | 150 | 56° 19′ | 1664 | 28 | 55 |
| 40 | 21° 48′ | 743 | 46 | 93 | 160 | 58° 0′ | 1696 | 26 | 53 |
| 45 | 24° 14′ | 821 | 46 | 91 | 170 | 59° 32′ | 1724 | 25 | 51 |
| 50 | 26° 34′ | 894 | 45 | 89 | 180 | 60° 57′ | 1748 | 24 | 49 |
| 55 | 28° 49′ | 964 | 44 | 88 | 190 | 62° 15′ | 1770 | 23 | 47 |
| 60 | 30° 58′ | 1029 | 43 | 86 | 200 | 63° 26′ | 1789 | 22 | 45 |
| 70 | 34° 59′ | 1147 | 41 | 82 | 250 | 68° 12′ | 1857 | 19 | 37 |
| 80 | 38° 40′ | 1250 | 39 | 78 | 300 | 71° 34′ | 1897 | 16 | 32 |

**Table 11.—Bending Stresses over Sheaves for Cast-steel Rope**

| Size of Rope, in. | Kind of Rope | Diameter of Sheaves, in. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 12 | 18 | 24 | 30 | 36 | 48 | 60 | 72 |
| | | Bending Stress, lb. | | | | | | | |
| 1/2 | 6 × 7 | …… | 3,680 | 2,800 | 2240 | 1,840 | 1,400 | 1120 | 920 |
| | 6 × 19 | 2,720 | 1,860 | 1,360 | 1120 | 920 | 680 | 560 | 460 |
| | 6 × 37 | 1,840 | 1,240 | 920 | 760 | 620 | 460 | …… | …… |
| | 8 × 19 | 1,720 | 1,120 | 860 | 680 | 560 | 420 | 340 | 280 |
| 3/4 | 6 × 7 | …… | …… | 9,440 | 7520 | 6,240 | 4,720 | 3760 | 3120 |
| | 6 × 19 | 9,440 | 6,400 | 4,720 | 3760 | 3,200 | 2,360 | 1880 | 1600 |
| | 6 × 37 | 6,240 | 4,160 | 3,120 | 2520 | 2,080 | 1,560 | 1260 | 1040 |
| | 8 × 19 | 5,760 | 3,840 | 2,880 | 2320 | 1,920 | 1,440 | 1160 | 960 |
| 1 | 6 × 7 | …… | …… | …… | …… | 14,960 | 11,200 | 8960 | 7480 |
| | 6 × 19 | …… | 14,880 | 11,200 | 8960 | 7,440 | 5,600 | 4480 | 3720 |
| | 6 × 37 | 15,040 | 10,080 | 7,520 | 6000 | 5,040 | 3,760 | 3000 | 2520 |
| | 8 × 19 | …… | 9,120 | 6,840 | 5480 | 4,560 | 3,420 | 2740 | 2280 |

Values must be regarded as comparative only and not absolute. The principal value of the table is to suggest fatigue effect due to bending. High bending stress with respect to direct load means relatively short life of rope. Low bending stress as compared with direct load prolongs the life of rope. One maker says that if in calculating rope stresses the total static load is increased by force necessary to accelerate it to maximum rope speed, and if sheave and drum diameters accord with good practice, bending stresses will have little effect on safety of the rope or economy of operation. C. D. Meals (*Trans. A.S.M.E.*, MH–51–5, 1929) gives diagrams and tables to show comparative effects of bending in wire rope of various constructions.

Fatigue of metal is more rapid if rope is subjected to reverse bending. Reverse bends should be avoided if possible; if not, all sheaves should be made larger than ordinary.

If a hoisting rope runs direct from drum to load, Fig. 2, each part of the rope makes one bend in wrapping on the drum and one bend in leaving it, a total of 2 bends for the cycle of hoisting and lowering. In Fig. 3 the rope makes 8 bends and 2 reverse bends per cycle, and the wear on rope, due to bending, will be at least 5 times that of Fig. 2; the reverse bends of Fig. 3 will make the ratio still more unfavorable.

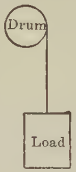


Fig. 2

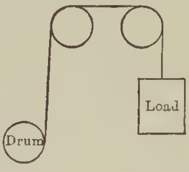


Fig. 3

Small equalizer sheaves used with double ropes on hook blocks on cranes often cause wire breakage. The motion of the rope on the sheave is slight, bending is concentrated on a short length of wire and breakage results. Some makers use equalizer bars instead of sheaves. One rope is fastened to each end of the bar.

Internal wear in the rope due to wires and strands sliding on each other can be reduced by proper lubrication. Tests of 9/16-in. plow steel ropes showed that a lubricated rope run over a 10-in. sheave made 2 1/2 times as many turns as a dry rope before the rope failed; over a 24-in. sheave, 5 times as many.

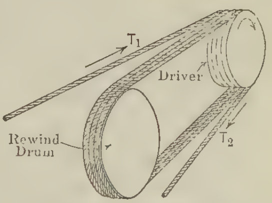


Fig. 4. Winding Machine Drive

**DRUMS AND SHEAVES FOR WIRE ROPE.**—A wire rope may be used for hoisting haulage or power transmission in various ways:

1. By winding on a drum, grooved or smooth, in one or more layers. The drum may be cylindrical, a frustum of a cone or a combination of the two. Winding on a drum generally is used in hoists and in haulage back and forth.

2. By winding on a winch head. Rope leads on a large diameter, makes several wraps and pays off on a smaller diameter. Tension is applied to the slack rope. Under load, a side slip tends to wear rope and winch head. This method is used on capstans, car pullers and for intermittent hoisting and hauling.

3. By winding several wraps of an endless rope on a concave-faced drum or sheave shaped like a winch head. Tight side of rope leads on a high diameter; slack rope leads off a low diameter to a tail sheave to which tension is applied. The side slip of the rope on the drum is objectionable and this method is not used widely.

4. By winding and rewinding an endless rope around a pair of grooved drums, and from drums to a tail sheave to which tension is applied. This method is used in endless rope hauling, in traction drives for high-speed elevators, etc. See Fig 4. If $T_1$ = ten-

sion in rope leading to main drum, and $T_2$ = slack tension in rope coming off drum, then

$$T_1 = T_2 \times 10^{0.00758f\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where $f$ = coefficient of friction between rope and drum, $\theta$ = total angle of wrap on the driving drum, deg. of arc. In Fig. 4, $\theta = 720°$. For discussion of this formula, see p. 24–14. For values of $f$, see p. 8–24. If rope runs in V-shaped grooves, bears on the sides only and does not touch the bottom, the increased coefficient of friction, $f_1 = f/(\sin V/2)$. Angle $V$ between sides may be from 25° to 45°. The smaller this angle, the larger the friction and the greater the value of $T_1$ for a given value of $T_2$, but the wear on rope and groove from slip, creep and motion at engagement and disengagement may be destructive if the load is great or the hours of service long.

5. By tractive effort of a sheave with grooves lined with wood, leather or rubber friction blocks. This method is used for power transmission with high-speed wire ropes. Such large diameter sheaves used with counterbalanced mine cages is the Koepe system of hoisting. See Peele, Mining Engineers' Handbook (John Wiley & Sons), and catalogs of wire rope makers.

6. By the use of a grip sheave. Within the rim of the sheave pairs of jaws hinged together are spaced. As they yield to the pressure of the rope they grip it. Springs open the jaws at point of disengagement. This method is used for endless rope carhauls for rope tramways. Older forms of grip sheaves have been known to cut and break ropes; for an improved form (Am. Steel and Wire Co., Worcester, Mass.) see *Trans.* A.S.M.E. 1931, MH-53–1.

German hoisting machinery uses several forms of grip sheaves; the sheave is parted along the axis of the rope, and the two parts squeezed together to exert pressure on the rope. Another German device obtains tractive force from a single drum. The rope wraps several times around a plain cylinder but rests in grooves in sections of a link chain, which hangs loose and is pushed across the face of the drum by auxiliary mechanism. Another method is to wind the rope around two drums (Fig. 4) and drive both drums through gearing which equalizes the work on each drum. See *Fördertechnik*, 1929–1932.

**SHEAVE AND DRUM SIZES.**—The larger the diameter over which rope runs, the longer will it last, but the cost and trouble of renewing rope usually is less than the difficulty of accommodating larger drums and sheaves and the greater weight and cost of heavier machinery. Such cases are tackle blocks, small hoists, shop cranes, derricks, dredges, excavators, etc.

Ropes on small sheaves and drums are subjected both to a larger stress due to direct bending (see Bending Stresses), and pinching of wires and strands on each other, preventing free movement; also, they may be nicked and broken by the pressure. When a rope runs over a sheave, the tightening of the outer wires tends to twist the rope. The smaller the sheave, the greater is the twist, and also the greater is the unit pressure of the rope on the sheave tread. This twist under pressure causes the rope to cut the sheave, producing sharp edges which wear away outer wires and shorten rope life. To avoid this trouble cast-steel sheaves should be used. Manganese-steel sheaves, ground true in the groove, are still better. C. D. Meals (*Civil Engg.*, 1930) gives a table of allowable pressures for various kinds of rope on cast-iron, cast-steel or manganese-steel sheaves. One way to destroy a new rope is to run it on sheaves worn and corrugated by an old one.

Table 12 gives in the form of ratios of sheave or drum diameter to rope diameter, the following: 1. Critical diameters; those which pinch the rope and prevent free movement of strands. Rope has short life under such conditions. 2. Minimum economical diameters; these should be chosen where sizes of sheaves and drums are not limited by considerations of available space or weight of parts, and where economy in the use of rope is important. 3. Desirable diameters; those recommended for heavy work in important installations, as mine hoists, skip hoists, heavy cranes, hot-metal cranes, and in places where safety is important, and where shut-down from rope breakage cannot be tolerated. The table refers to cast-steel ropes which tests as wire about 180,000 lb. per sq. in. Ropes of stronger steels are stiffer, harder and cost more. They should be used with larger sheaves and drums. If of improved or special cast-steel, good for 200,000 lb., $^1/_9$ larger;

**Table 12.—Diameter Ratios of Sheaves and Drums for Cast-steel Running Ropes**

| | 6 × 7 | 6 × 19 Seale | 6 × 21 Seale<br>6 × 19 Warrington | 6 × 19 Filler Wire | 6 × 31 Seale | 6 × 37 | 8 × 19 Warrington | 8 × 19 Seale | Tiller |
|---|---|---|---|---|---|---|---|---|---|
| Critical Ratios............. | 28 | 20 | 16 | 16 | 15 | 14 | 14 | 16 | 10 |
| Minimum Economical Ratios | 36 | 28 | 24 | 22 | 18 | 16 | 20 | 22 | 14 |
| Desirable Ratios............ | 72 | 50 | 45 | 42 | 38 | 27 | 31 | 42 | 20 |

if of plow steel, good for 210,000 lb., $^1/_6$ larger; if of improved plow steel, good for 240,000 lb., $^1/_3$ larger. See Bending Stresses.

**PROPER GROOVE CONTOUR.**—Groove diameter as measured by template or gage should be slightly larger than rope diameter. New ropes are slightly over size, and for a proper fit groove diameter as measured by template should be larger than nominal rope diameter as follows: $^1/_2$-in. rope and smaller, $^1/_{16}$ in.; $^9/_{16}$- to 1-in. rope, $^3/_{32}$ in.; 1 $^1/_{16}$- to 2-in. rope, $^5/_{32}$ in. See Fig. 5.

As metal wears away in the tread or bottom of groove, and as rope wears smaller in diameter there is a new condition of fit, with groove diameter equal to that of the worn rope. See Fig. 6. If a new rope runs over worn sheaves, it will not fit the worn groove with proper clearance but will be jammed into it, preventing free movement of the rope, and concentrating abrasion on two lines. Also, there will be a tendency to distort the shape of the rope and cause nicking and breaking of wires. Rope diameter is the largest diameter outside of strands.

**DETERIORATION OF WIRE ROPE.**—Ropes deteriorate by: 1. Frictional wear. 2. Breakage of wires due to defects of materials. 3. Breakage due to bending stresses. 4. Corrosion, inside and outside. 5. Fatigue in steel in the wires. All defects finally show as broken wires, and their appearance guides decisions of when to discard a rope. After a new rope has been put in service, the crowns of outside wires wear to an extent necessary to form a wearing surface or contact area to support the rope on sheaves and drums. The reduction in strength of rope is slight and need not cause concern if wear is slow. The appearance of broken wires does not necessarily indicate rapid deterioration. If the rope has started with the proper factor of safety, early abrasion and breakage of wires may be considered normal deterioration within safe limits, but when broken wires continue to appear regularly, removal of the rope should be considered.

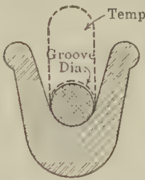


FIG. 5. Correct Design of Rope Groove

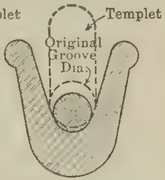


FIG. 6. Correct Design with Rope Worn

Safety of operation depends on strength of the rope as it nears the time of removal. The factor of safety at this time is hard to determine; it depends both on wear and breakage of visible outside wires, and on corrosion and breakage of invisible inside wires. These combine to cause a reduction in diameter of the rope, which, when greater than wear on outside wires would account for, may indicate internal corrosion. *Coal Age*, Nov. 1924, reports a 1 $^3/_8$ in., 6 × 19 mine hoist rope regularly inspected and reported in good condition, which suddenly broke at the part in contact with the head sheave, with cage at bottom of the shaft. Examination of the broken ends showed internal corrosion due to entrance of water; the hemp core was crushed to a pulp 10 ft. on each side of break and rope diameter was reduced from 1 $^3/_8$ in. to 1 $^9/_{32}$ in.

If a rope has not been corroded, its remaining strength may be estimated by counting broken wires in one rope lay in the worst part of the rope, and measuring amount of metal removed by abrasion of outside wires. In one rope lay all outside wires in that length of rope can be seen. Most of the strength of the rope is in these wires, ranging from 83% for 6 × 7 rope to 57% for 6 × 19 Warrington rope. John A. Roebling's Sons Co. publishes charts which show, from observed facts of breakage and abrasion, the remaining strength of a used rope, provided there is no internal corrosion. Amer. Cable Co. publishes tables telling when ropes in various kinds of services should be discarded. These tables may be summarized as follows: Rope should be discarded when in one rope lay from 8% to 20% of visible wires are broken; or when 15% to 25% of wires in one strand are broken, especially if they are adjacent wires; or when outside wires are worn to $^1/_2$ or $^2/_3$ of original diameter. For high-speed work, heavy loads, and when risk to life or property must be kept low, the lower percentages are recommended. If breakage of a rope merely means a machine stoppage, ropes can be run until outside wires are worn off or until a strand breaks.

Rules and tables for discarding ropes are suggestive only. For instance, it is not possible to group all shop cranes in one class. It is better to treat each installation separately, examine rope systematically and regularly, note and record breakage and wear of wires. A minimum life based on observed facts can be set up as a standard; if tensile tests of parts of discarded rope show it to have more than requisite strength, considering a proper factor of safety, a new basis can be set up. After a few ropes have been discarded and tested, a standard for that installation can be established that will allow rope to be used with safety and economy.

**EARLY FAILURE OF WIRE ROPE.**—Faults which appear early in the use of a rope should not be laid always to poor manufacture. Careless handling and reeving of the rope often is responsible. The life of a rope often can be increased by putting an obligation of responsibility and competition on the men whose duty it is to use it. (*Proc.* A.S.C.E. May, 1932).

Tech. Paper 237, Bureau of Mines, 1919, gives valuable information on use of rope in mine hoists. It suggests: 1. If one broken wire is found in a rope at a socket fastening, socket should be cut off and renewed. 2. Even if no broken wire is found at a socket or thimble it should be renewed at intervals of not over 4 months, depending on life of rope in service. 3. Bolts in clip fastenings must be kept tight. 4. After 3 yr. in service, even if idle, rope should not be used unless tested for ultimate strength. 5. Safety factor must never be less than 4 $^1/_2$. 6. Ropes of 6 × 19 construction must be discarded when 6 broken wires occur in one rope lay, or when outside wires are worn to $^2/_3$ original diameter. 7. It is better to err on the side of safety than to prolong life of a rope. Accidents generally cost more than new rope.

A Research Committee of the A. S. M. E. is (1936) co-operating with the U. S. Bureau of Standards to establish by tests of used rope a relation between observed facts of wire breakage and abrasion and the strength remaining in the rope.

**ATTACHING WIRE ROPE.** (Tech. Paper, 237, Bureau of Mines.)—The rope may be attached to its load by a socket, Fig. 7, or it may be bent back on itself to form an eye and the loose end clamped to the rope with rope clips, Fig. 8. Properly done, the first method develops the full strength of the rope, but has as disadvantages: poor workmanship is concealed; it cannot be readily inspected; the rope is rigidly held and the effect of bending is concentrated just above the socket; it is more difficult to make. Fastening by clips develops not over 85% of the strength of the rope. It bruises and crushes the rope where the clips are applied, and is liable to slip, but it can be readily inspected, and requires little skill in making. Rope makers recommend the socket method of fastening, which, if made according to instructions, never fails. After either a socket or clip attachment is made it should be tested by making several trips under a heavy load before men are allowed to ride.

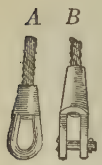


Fig. 7. Socket

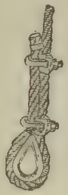


Fig. 8. U-bolt and Clip

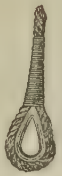


Fig. 9. Thimble

Methods of Attaching Wire Rope

The clip attachment is made as follows: The rope is bent back over a thimble of the proper size and the loose end clamped against the long end of the rope with the proper number of clips. The Crosby type of U-bolt-and-drop-forging clip has proved most satisfactory. The clips should be spaced a distance apart of 6 times the rope diameter, the drop forging being against the long end and the U-bolt against the loose end of the rope. The number of clips required to develop approximately 80% efficiency of the rope and the proper length of wrench to tighten the bolts are given in Table 13. Never put the drop forging of the clip against the loose end of the rope.

The clips must be carefully inspected occasionally and tightened if they show any signs of loosening through the reduction in diameter which comes from stretching of the rope and compression of the hemp core.

Thimbles (Fig. 9) may be held in the bend of the rope by clips or they may be spliced in. A spliced end, properly made, will not pull out, but it will not develop the full strength for $^1/_2$-in. rope, and not over 60% for 2 $^1/_2$-in. rope. Thimbles and sockets form the means of attachment of hooks and shackles and eyes of various kinds. See Figs. 12, 13 and 14. For high tensile ropes (plow-steel, etc.), it is best to use the special heavy thimbles and sockets listed by wire rope makers.

**FASTENING WIRE ROPES TO DRUMS.**—Fig. 10 (Link-Belt Co.) shows a method of readily attaching or detaching the rope from the circumference of a drum on an electric hoist. The last groove sweeps with a large radius into a depression which allows the socket on the end of the rope to be slipped into the inside of the drum; a pipe plug closes the cavity and locks the socket in place. By unscrewing the plug, the socket can be pulled out and the rope removed.

Where the end of the drum or its hub is readily accessible the rope can be secured by a few wraps around the hub with the end clipped to itself or to one of the arms or spokes of the drum.

**Table 13.—Number of Crosby-type Rope Clips Required to Develop 80% of Strength of 6 × 19 Plow-steel Rope**

| Diameter of rope, in. | $^3/_4$ | $^7/_8$ | 1 | 1 $^1/_8$ | 1 $^1/_4$ |
|---|---|---|---|---|---|
| Number of clips | 5 | 5 | 5 | 5 | 6 |
| Length of wrench, in. | 18 | 18 | 24 | 24 | 24 |
| Efficiency, percent | 77 | 79 | 77 | 80 | 82 |

**Table 14.—Drop-forged Hoist Hooks**
(J. H. Williams & Co., Brooklyn, N. Y.)

| Shank Hook. | | | | Hook with Eye | | | | Capacity, Net Tons | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Diam. of Shank, in. | Length of Shank, in. | Extreme Length, in. | Extreme Width, in. | Inside Diam. of Eye, in. | Outside Diam. of Eye, in. | Extreme Length, in. | Extreme Width, in. | Safe Working Load | Aver. Load at Elastic Limit | Approximate Load Required to Straighten Out |
| 1/2 | 2 | 5 | 2 7/8 | 3/4 | 1 1/2 | 4 3/8 | 2 7/8 | 0.5 | 0.9 | 1.9 |
| 9/16 | 2 1/4 | 5 1/2 | 3 1/8 | 7/8 | 1 3/4 | 4 7/8 | 3 1/8 | .6 | 1.2 | 2.3 |
| 5/8 | 2 1/2 | 6 | 3 1/2 | 1 | 2 | 5 3/8 | 3 1/2 | .7 | 1.5 | 3. |
| 3/4 | 2 3/4 | 6 3/4 | 3 7/8 | 1 1/8 | 2 1/4 | 6 3/16 | 3 7/8 | 1.2 | 2.5 | 5.7 |
| 7/8 | 3 | 7 1/2 | 4 3/8 | 1 1/4 | 2 1/2 | 6 7/8 | 4 3/8 | 1.7 | 3.5 | 7. |
| 1 | 3 1/4 | 8 1/4 | 4 7/8 | 1 3/8 | 2 3/4 | 7 5/8 | 4 7/8 | 2.1 | 4.2 | 8.5 |
| 1 1/8 | 3 1/2 | 9 1/4 | 5 5/8 | 1 1/2 | 3 | 8 9/16 | 5 5/8 | 2.5 | 5.4 | 10. |
| 1 1/4 | 3 3/4 | 10 1/4 | 6 3/8 | 1 5/8 | 3 1/4 | 9 9/16 | 6 3/8 | 3. | 6.2 | 13. |
| 1 3/8 | 4 | 11 | 6 7/8 | 1 3/4 | 3 1/2 | 10 3/8 | 6 7/8 | 4. | 8. | 17. |
| 1 1/2 | 4 1/4 | 12 | 7 1/2 | 2 | 4 | 11 1/2 | 7 1/2 | 4.7 | 9. | 19. |
| 1 5/8 | 4 1/2 | 13 | 8 1/4 | 2 3/8 | 4 5/8 | 13 | 8 1/4 | 5.5 | 11. | 26. |
| 1 3/4 | 5 | 14 1/2 | 9 1/4 | 2 3/4 | 5 1/4 | 14 3/4 | 9 1/4 | 6.8 | 13. | 32. |
| 2 | 5 1/2 | 16 | 11 | 3 1/8 | 6 1/8 | 16 3/4 | 11 | 8. | 17. | 35. |
| 2 3/8 | 6 | 17 7/8 | 13 1/2 | 3 1/2 | 7 | 19 1/8 | 13 1/2 | 11. | 21. | 48. |
| 2 1/2 | 6 1/2 | 19 3/4 | 14 1/2 | ........ | ........ | ........ | ....... | 15. | 25. | 55. |
| 2 3/4 | 7 | 21 1/2 | 15 | 4 | 8 1/2 | 22 3/4 | 15 | 20. | 40. | 80. |
| 2 3/4 | 7 | 21 1/2 | 15 | 4 | 8 1/2 | 22 3/4 | 15 | 25. | 50. | ......... |

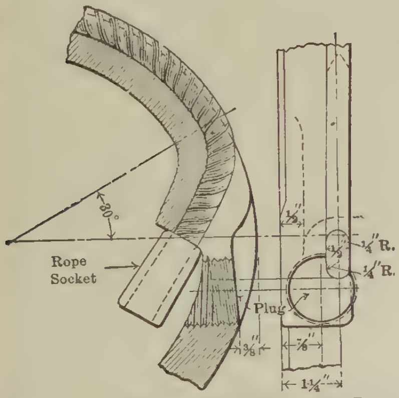


FIG. 10. Method of Fastening Wire Rope to Drum

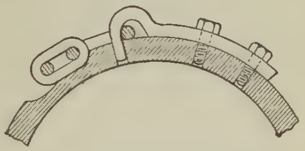
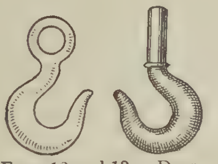


FIG. 11. Method of Fastening Chain to Drum



FIGS. 12 and 13. Drop Forged Hoist Hooks

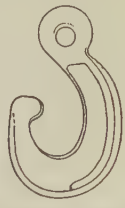


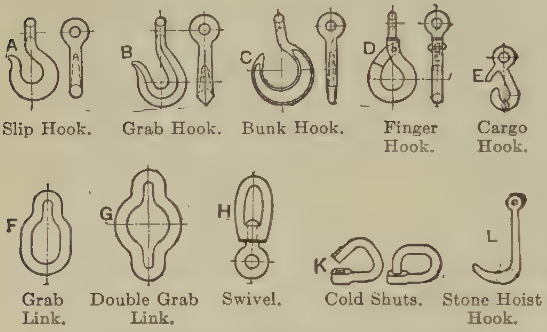
FIG. 14. Bemis Safety Hook

The hole in the rim through which the rope passes should be well rounded to avoid a sharp kink in the rope. Wherever possible the drum should be cast with a ledge or lug to lead the rope tangent to the hub over a radius of 6 or 8 diameters of the rope (similar to Fig. 10). To fasten the end link of a chain to a drum, it is customary to use a forged clamp, tap-bolted to the rim of the drum, and provided with a finger equal in diameter to the size of the chain. (Fig. 11.)

**DROP-FORGED HOIST HOOKS** of steel without welds are made in sizes up to 25 tons capacity. They are heat treated after forging and tested to 50% beyond rated working load. See Figs. 12 and 13 and Table 14. Drop forge companies have stock dies for most sizes and furnish hooks in moderate quantities without making charge for dies. Drop-forged hooks of dimensions in Table 14 are good for loads of 3 1/2 to 5 times rating, if sling or fixture rests in the bottom of the jaw. If the load improperly comes on the point of the hook, the carrying capacity is much less. In no event is the hook apt to break suddenly; the jaw will open out first, and signify overloading or improper engagement of the hook.

The Bemis patent safety hook, Fig. 14, (Am. Loco. Co., New York) has rated capacities up to 14,000 lb. It is made in four sizes. Hooks fitted with movable guards are not favored by stevedores. They are apt to get out of order and time is lost in putting slings on or off.

Fig. 15 shows various forms of forged connections used with hoists and slings. *A* is a slip hook used on end of rope or chain sling passed around a load to tighten on the load when hoisted. *B* is a grab hook used on chain slings which are not to tighten on the load; the throat of the hook allows the chain to enter but not to slip. *C* is a bunk hook used on tackle to take several rope slings at one time. *D* is a finger hook to avoid the danger of the hook catching under the edges of hatches in ships. *E* is a cargo hook for the same purpose. *F* is a grab link used instead of a grab hook; one end of the chain sling is fastened to the large end of ring; the chain is passed through and when the loop is of proper size the small end of the link is pulled between two links of chain. *G* is a double grab link used to shorten a chain or join two quickly and temporarily. *H* is a swivel used to prevent twisting of load by twisting of the hoisting rope or chain. *K* is a cold-shut for repairing a chain or connecting two chains. *L* is a stone hoist hook, used in quarry work; the points of the hooks engage the sides of the stone blocks.



Slip Hook. Grab Hook. Bunk Hook. Finger Hook. Cargo Hook.

Grab Link. Double Grab Link. Swivel. Cold Shuts. Stone Hoist Hook.

FIG. 15. Forms of Forged Connections Used with Hoists and Slings

Steel for hooks, rings and hoisting fixtures should be of known analysis. Test specimens should be subjected to bending to insure absence of pipes, seams or other imperfections in the bar. Hot-rolled steel of forging quality, S.A.E. specification 1010 is used by some makers.

In one important works all chains, slings, hooks and lifting fixtures are inspected once a month; parts that can be benefited by annealing are so treated once a year (*Iron Age*, May 22, 1930).

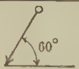
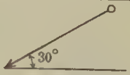
**WIRE ROPE SLINGS**, as compared with chain slings, are lighter and, in general, will not fail without warning. Breakage of outer wires is the visible warning, but the inner wires alone will carry loads at which the slings are rated by the makers.

Plain slings are made of standard 6 × 19 or 6 × 37 rope spliced endless, or fitted with sockets, thimbles or eye ends. In the Atlas patent slings (MacWhyte Co., Kenosha, Wis.) each member consists of one right and one left lay rope braided around a hemp core back and forth, so that the sling member has a strength of 8 or more strands of the rope. See Fig. 16. The ends are loops which can be used as eyes, or fitted to sockets or thimbles. In the Drew sling, made by the same company, an endless wire rope is braided into a band

**Table 15.—Plow Steel Wire Rope Slings**

(John A. Roebling's Sons Co., Trenton, N. J.)

The safe load for any number of parts may be obtained from values given below by direct multiplication. Slings 3/8 in. to 1 1/8 in. diameter inclusive, 6 × 19 rope; slings 1 1/4 in. diameter and larger, 6 × 37 rope.

| Diam. of Rope, in. | Safe Load Values when Used in Single Part | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Vertical | |  | | | |  | |
| | Connection | | Connection | | Connection | | Connection | |
| | Socket | Splice | Socket | Splice | Socket | Splice | Socket | Splice |
| 3/8 | 950 | 950 | 850 | 850 | 700 | 700 | 500 | 500 |
| 1/2 | 1,600 | 1600 | 1,400 | 1400 | 1100 | 1100 | 800 | 800 |
| 5/8 | 3,200 | 3200 | 2,800 | 2800 | 2300 | 2300 | 1600 | 1600 |
| 3/4 | 4,800 | 4550 | 4,200 | 3900 | 3400 | 3200 | 2400 | 2250 |
| 7/8 | 6,400 | 5750 | 5,600 | 5000 | 4500 | 4100 | 3200 | 2900 |
| 1 | 8,400 | 7100 | 7,300 | 6200 | 6000 | 5000 | 4200 | 3550 |
| 1 1/8 | 10,300 | 8400 | 8,900 | 7300 | 7300 | 6000 | 5150 | 4200 |
| 1 1/4 | 11,900 | 9800 | 10,300 | 8500 | 8400 | 7000 | 5950 | 4900 |

about 2 rope diameters thick, 6 or more diameters wide. John A. Roebling's Sons Co., Trenton, N. J., makes slings whose members are of "grommet" construction, *i.e.*, one strand braided back on itself to form a 6- or 7-strand rope, endless except for a splice in the one strand. The ends preferably are held by patented thimbles, which are large, permit some slip to equalize the load, and in the form shown in Fig. 17, one thimble can be passed through the other to form a loop or anchor sling. The grommet link, Fig. 18, avoids using a forged steel ring to connect the two legs of a sling. All tension comes on a wire rope ring spliced endless as described above.

Fig. 19 shows a sling with clamp hooks for lifting plate. It has an endless grommet with 3 equalizing thimbles, and carries plate clamps with a positive grip. Structural steel slings may be fitted with a toggle and releasing pin for detaching the sling, when slack, by pulling a hand line from a distance.

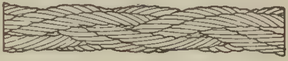


Fig. 16. Atlas Patent Sling

The rated carrying capacity of a sling depends on the kind of fittings, the way the sling engages the load and other factors obtained by experience. Table 15 gives carrying capacities of plow-steel slings which bind against the load. If the sling does not come in contact with the load or if it passes around a large diameter, so that stresses are not concentrated at corners, safe loads may be increased somewhat above the tabular figures.

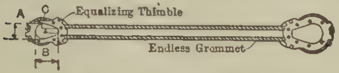


Fig. 17. Grommet Sling

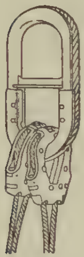


Fig. 18. Grommet Link

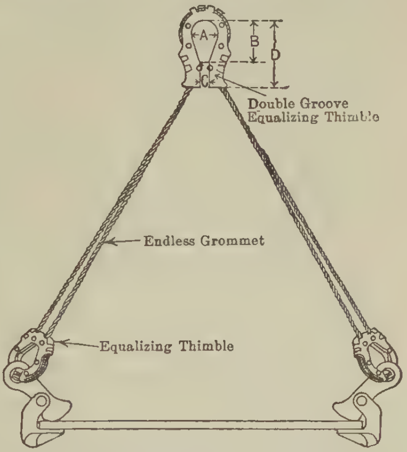


Fig. 19. Grommet Sling with Clamp Hooks

# HANDLING MATERIAL IN BULK

## 1. WIRE ROPE HAULAGE

Wire rope haulage is less important now than in the days before the general use of electric mine locomotives, chain carhauls, coal conveyors, etc. The following is abridged from publications of John A. Roebling's Sons Co., and The American Steel & Wire Co.

**SYSTEMS OF WIRE ROPE HAULAGE** are: 1. Self-acting inclined plane, used on grades where the descending loaded cars raise the empty cars. 2. Engine plane, where a reversible hoisting machine and a single wire rope are used to raise and lower cars on a slope. 3. Tail rope system, consisting of a reversible hoisting machine with two drums, clutches and brakes, a main rope hitched to the cars to pull them in one direction and a tail rope hitched to the other end of the train to pull them in the opposite direction, the brakes on the drums being used to check the speed of the cars on down-grades and prevent them from overrunning the forward rope. 4. Endless rope system, used in mines, on piers, and at coal and ore storage plants for moving railroad cars. The rope is driven by a grip sheave, or it is wrapped a number of times between the driving drum and a rewind (or back) drum mounted on the engine base, the number of half-wraps on the driving drum being sufficient to prevent slipping under load. The cars are attached to the rope at any place by hand-operated grips and can be detached by the same means without stopping the rope. The driving contact between drum and rope must be maintained by tension applied to the rope by a weighted sheave somewhere in the circuit. The weight $W = T + S$, where $T$ and $S$ are the tensions in the tight and slack sides, respectively. The resistance of cars, rope, idlers, etc., $= R = T - S$. Then to prevent slip, $W = KR$, where $K$ is a factor depending on the coefficient of friction between rope and drum, and on the number of half-wraps. Values of $K$ are given in Table 1. In general,

$$T = R(K + 1) \div 2$$

The increase in tension in the endless rope, compared with the main rope of the tail-rope system, where the stress in the rope is equal to the resistance, is about as follows, assuming $f = 0.400$:

| | | | | | | |
|---|---|---|---|---|---|---|
| $n$ = | 1 | 2 | 3 | 4 | 5 | 6 |
| Increase in tension in endless rope, compared with direct stress, % | 40 | 9 | 2 1/3 | 2/3 | 1/5 | 1/10 |

These figures are useful in determining the size of rope. For instance, if the rope makes two half-laps on the driving drum, the strength of the rope should be 9% greater than a main rope in the tail-rope system. The values of $f$ below are coefficients of friction between rope and drum.

| | Dry | Wet | Greasy |
|---|---|---|---|
| Wire rope on a grooved iron drum, $f$ | 0.120 | 0.085 | 0.070 |
| Wire rope on wood-filled sheaves, $f$ | 0.235 | 0.170 | 0.140 |
| Wire rope on rubber and leather filling, $f$ | 0.495 | 0.400 | 0.205 |

The importance of keeping the rope dry is evident from these figures, but calculations should allow for wet ropes in most cases.

**WIRE ROPE TRAMWAYS,** or simply **Ropeways** are transporting systems in which moving carriers or containers hang from a wire rope carried on a series of spaced supports, and are moved along by a rope, generally from one terminal to another. Ropeways which both hoist the load and carry it are cableways.

For distances over a few hundred feet a ropeway often is the best, cheapest, and sometimes only means of transport. Under favorable conditions it is low in first cost, cheap to operate, and maintenance and depreciation are low. Ropeways are durable and serviceable. One European builder says (1932) that over 90% of those built by him in the past 30 years are in use; of those built within 50 years, 75%. A ropeway 18 miles long carrying 36,000 tons a year at one-third the cost of transportation by truck is described in *Trans.* A.S.M.E. 1929, MH–51–2. Tramways are single rope and double rope.

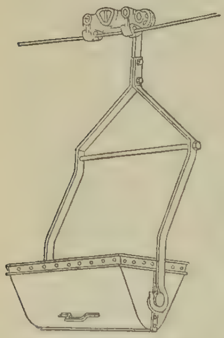


FIG. 1. Single Rope Tramway Carrier

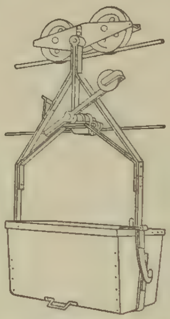


FIG. 2. Double Rope Tramway Carrier

**Single Rope Tramways** comprise one endless moving rope which both supports and moves the carriers (Fig. 1). At the driving terminal is a large grip sheave (see p. 23-08) or a multiple-groove sheave working with a rewind sheave (see p. 23-07). At the other end of the line, an idler sheave with a pull-back weight puts tension in the rope. Towers on the run carry sheaves to support rope and carriers. The carrier has a pair of clamps to grip the rope. In one construction (Pohlig) the grip permits some axial rotation of the rope, to lessen wear on grip and rope. Grips are automatically engaged or disengaged at terminals, and the carrier runs on its own rollers over fixed track to be loaded or unloaded. Ropeways are best built in a straight line, but angle stations can be provided. Single-

**Table 1.—Values of Coefficient $K$**

| $f$ | $n$ = Number of Half-laps on Driving-wheel | | | | | |
|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 |
| 0.070 | 9.130 | 4.623 | 3.141 | 2.418 | 1.999 | 1.729 |
| .085 | 7.536 | 3.833 | 2.629 | 2.047 | 1.714 | 1.505 |
| .120 | 5.345 | 2.777 | 1.953 | 1.570 | 1.358 | 1.232 |
| .140 | 4.623 | 2.418 | 1.729 | 1.416 | 1.249 | 1.154 |
| .170 | 3.833 | 2.047 | 1.505 | 1.268 | 1.149 | 1.085 |
| .205 | 3.212 | 1.762 | 1.338 | 1.165 | 1.083 | 1.043 |
| .235 | 2.831 | 1.592 | 1.245 | 1.110 | 1.051 | 1.024 |
| .400 | 1.795 | 1.176 | 1.047 | 1.013 | 1.004 | 1.001 |
| .495 | 1.538 | 1.093 | 1.019 | 1.004 | 1.001 | ..... |

rope tramways are used for relatively light loads, which can be spaced with some regularity, and where grades on the line are not over 30%.

**Double Rope Tramways** are those most used in the U. S.; they are suited to a wider variety of loads, grades and spans than single rope tramways. They have two track cables, one for loads, one for empties, usually of locked coil construction (see p. 6-42) anchored at one end, with tension weights at the other. The carriers, Fig. 2, have rollers running on the track cables and are pulled along by a hauling rope which they grip. The hauling rope is driven, tensioned and supported as is the rope of a single rope tramway. The grip is worked by two jaws on a threaded pin to which a lever is attached. At the terminal a stop throws the lever over and tightens or loosens the clamp.

Wire rope tramways are built by manufacturers [4], [13], [71], [78] listed on p. 23-81, and others. For details of American ropeways, see Peele, Mining Engineers' Handbook (John Wiley & Sons) and makers' catalogs. For European practice, see P. Stephan, Die Drahtseilbahnen (Julius Springer, Berlin). Costs of handling are compared with costs by other methods in *Trans.* A.S.M.E., 1930, MH–52–4. See also *Trans.* A.S.M.E., 1931, MH–53–4, E. J. Lloyd, on recent improvements in automatic dumps, etc.

**SUSPENSION CABLEWAYS OR CABLE HOIST-CONVEYORS** are systems of hoisting and transporting materials consisting of one or two track cables stretched in one span between terminals, a bucket, container or grab suspended from a carriage running on the track cable, ropes for hoisting the suspended load and moving the carriage, and winding drums for the ropes. In quarrying, rock-cutting, stripping, piling, dam-building, and many other operations where it is necessary to hoist and convey large individual loads economically it frequently happens that the application of a system of derricks is impracticable, by reason of the limited area of their efficiency and the room which they occupy. To meet such conditions cable hoist-conveyors are used, as they can be operated in clear spans up to 1500 ft., and in lifting individual loads up to 15 tons. Two types are made—one in which the hoisting and conveying are done by separate running ropes, and the other applicable only to inclines in which the carriage descends by gravity, and but one running rope is required. The moving of the carriage in the former is effected by means of an endless rope, and these are commonly known as "endless-rope" hoist-conveyors to distinguish them from the latter, known as "inclined" hoist-conveyors.

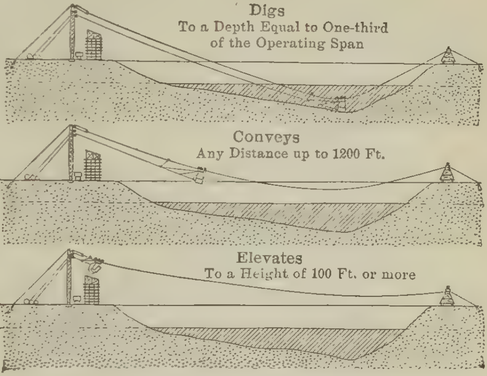

Digs
To a Depth Equal to One-third of the Operating Span
Conveys
Any Distance up to 1200 Ft.
Elevates
To a Height of 100 Ft. or more

Fig. 3. Slackline Cableway

The general arrangement of the endless-rope hoist-conveyors consists of a main cable passing over towers, A-frames or masts, as may be most convenient, and anchored to the ground at each end. Cable tension is maintained by a turnbuckle at one anchorage.

The carriage travels on this cable. It is moved back and forth by means of the endless rope. The hoisting is done by a separate rope, both ropes being operated by an engine specially designed for the purpose, which may be located at either end of the line, and is constructed in such a way that the hoisting-rope is coiled up or paid out automatically as the carriage is moved in and out. Loads may be picked up or discharged at any point along the line. Where sufficient inclination can be obtained in the main cable for the carriage to descend by gravity, and the loading and unloading are done at fixed points, the endless rope can be eliminated. The carriage, which is similar in construction to the carriage used in the endless-rope cableways, is arrested in its descent by a stop-block, which may be clamped to the main cable at any desired point, the speed of the descending carriage being controlled by a brake on the engine-drum.

For supporting the weight of the "fall" rope or hoisting rope, The Lidgerwood Mfg. Co., New York, uses "fall-rope" carriers spaced along a separate rope. The S. Flory Mfg. Co., Bangor, Pa., spaces them along the main track cable by sections of chains in loops. For details see makers' catalogs; also Peele, Mining Engineers' Handbook.

**SLACKLINE CABLEWAY.**—Fig. 3 shows three steps in the digging, conveying and elevating of material by a drag-line bucket (see p. 23-22) used with a slackline cableway. The bucket (Fig. 16) travels on an inclined track cable, one end of which is anchored. The other end is so rigged to the top of the mast that it can be tightened or slackened by the operator to vary its sag. One end of the hauling rope (load cable) leads to the bucket; the other end winds on a drum. The track cable spans the area served. When it is slack the bucket rests on the material to be dug, the hauling rope is wound in, the bucket digs its load and fills. The track cable is tightened, the bucket raised, and, according as the dump is at the high or low end, the carriage is pulled to or allowed to run to the dump. One man controls all operations. This type of plant is cheaply built and economically operated. It digs sand and gravel from pits and river beds, strips overburden and makes cuts and fills. By changing the position of the track cable anchorage and turning the mast it will dig a semi-circular pit. Fittings for slackline cableways and double drum hoists for operating them are made (see p. 23-81) by manufacturers [11], [50], [79].

**CABLE CRANES** are similar to suspension cableways, but by having one or both end towers movable they serve a greatly increased area. They are used in building canals, dams, locks and to serve large storage areas. In American practice (1936), work of this kind usually is done by power shovels (locomotive or crawler cranes) or slackline cableways, or combinations of them, with drag scrapers.

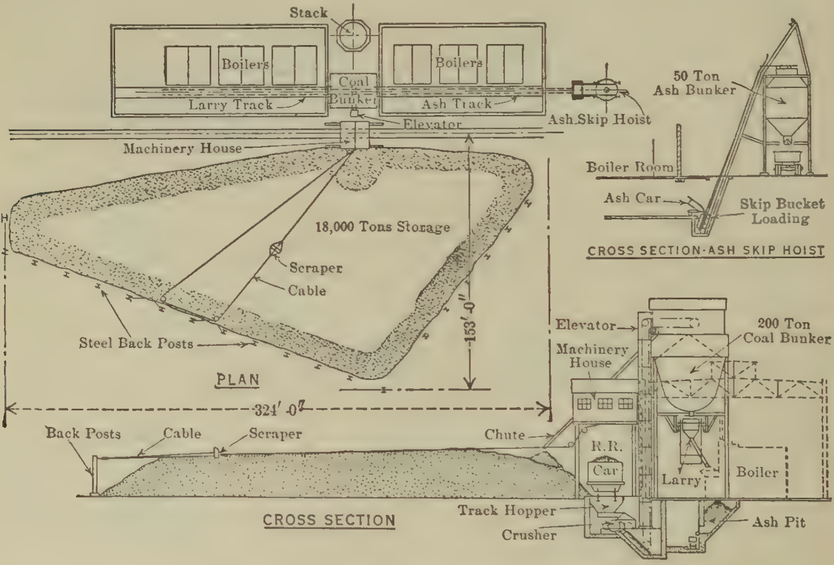


Fig. 4. Power Drag Scraper System

**POWER DRAG SCRAPER SYSTEM OF STORAGE.**—Fig. 4 shows elements of the system in a plant receiving coal by rail. The coal handling system crushes, elevates and delivers coal to the boiler-house, or forms a small pile near the railroad track. Two ropes run across the storage from a double-drum winding machine through tail blocks fastened to back posts. They are hitched to the front and back of an open end steel box, the scraper. Faced one way the scraper drags coal from the starting pile into storage; faced the other way it drags coal back to the track hopper. If the coal is frozen it goes to the crusher, otherwise the crusher is by-passed.

In small plants, tail blocks are hooked to tail posts, which are short steel columns in concrete piers. They may be hitched to a bridle cable stretched between adjacent posts. In large plants, back posts may be mounted on a car on a standard-gage track, and counterweighted to resist pull of the scraper rope. For high piles the tail block may be carried by a steel tower, 20 or 30 ft. high, on a broad flat car on wide-gage tracks. In one plant (Beaumont) the tail sheave is mounted on a caterpillar tractor which can travel anywhere on the storage lot and form a pile of any size or shape.

Advantages claimed for this system are: 1. Storage area can be any shape or size up to 900 ft. scraper travel. 2. Simple machinery, low first cost, cheap repairs, one-man operation. 3. Low fire risk in stored coal; ability to reach fire easily should it occur. 4. Wide range of economical handling capacities up to 1000 tons per hr.

Dragline scraper systems are built (see p. 23-81) by manufacturers [11], [50], [79]. Fig. 5

shows a Sauerman scraper; smallest size, $1/3$ cu. yd.; largest size, 12 cu. yd. Small drag scrapers for mines, tunnels, quarries are made by manufacturers [42], [86].

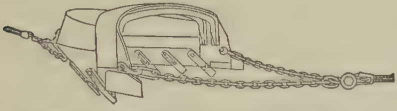


Fig. 5. Dragline Excavating Scraper

Fig. 6 shows a combination slackline cableway and dragline scraper system used in digging a canal in Russia and disposing of excavated material. All 3 towers are mounted on wheeled trucks running parallel to the canal. This combination has a higher capacity than could be obtained with a cable crane (see p. 23-16) spanning the whole width of the plant.

**Drag Scrapers in Cargo Boats.**—A number of self unloading boats on the Great Lakes and in coastwise service carrying bulk cargoes have a belt conveyor on a boom which can be slung to either side to discharge cargo ashore. See Fig. 7. Conveyor on the boom is fed from an inclined conveyor from the hold of the boat. There are several ways to take material from the hold and deliver it to the inclined conveyor. Some boats have two belt conveyors, others two apron conveyors, in tunnels extending the length of the cargo hold. Another method, patented by L. D. Smith, Sturgeon Bay, Wis., uses a pair of drag scrapers, each operated by its own double-drum winding machine, in two tunnels with smooth steel floors. Material flows into the tunnels through side openings. Along the top of each tunnel is an I-beam trolley track. The haulage cable is hitched to the open end of the scraper, Fig. 5; the pull-back cable for each scraper runs through a small tunnel, around end sheaves, over a trolley sheave hung from the I-beam, thence to the back of the scraper. The scrapers drop material into a hopper over the inclined conveyor. On the back pull the scraper is lifted up to the trolley, passes over material which has flowed into the tunnel, back to the starting point. Unloading capacities are 750 to 2000 tons per hr. Small scrapers are 4 cu. yd. capacity; large ones, 12 cu. yd. This system has been applied to tow barges with electric motors for operating the winding drums, but without generating plant. Power cables are run from the dock for unloading.

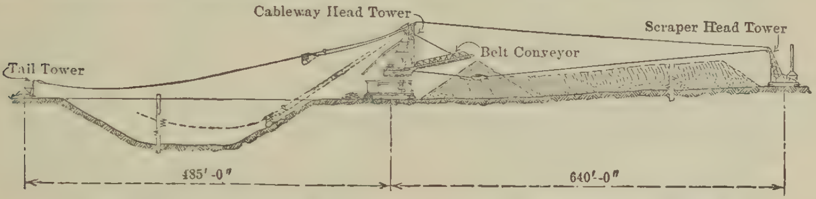


Fig. 6. Combination Slackline Cableway and Drag Line Scraper System

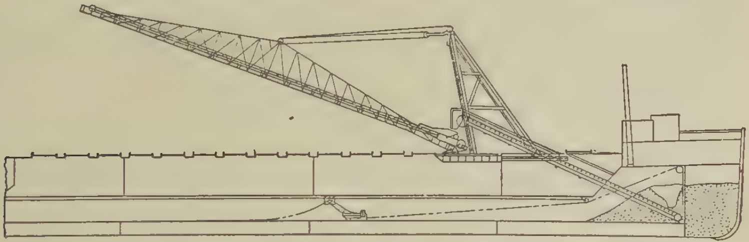


Fig. 7. Drag Line Scraper and Belt Conveyor System for Cargo Unloading

Boat unloading machinery with belt conveyors or apron conveyors in the tunnel is made by manufacturers [50], [77], [85] listed on p. 23-81.

## 2. LOCOMOTIVE CRANES

**LOCOMOTIVE CRANES,** Fig. 8, consist of a lower frame fitted with 4 or 8 wheels for use on track, and a rotating upper frame carrying the power plant and operating machinery. A boom, hinged to the upper frame, can be raised or lowered to handle loads at various radii. The crane will act as a portable hoisting engine, a swinging derrick, a grab bucket unloader or a switching engine. It will handle a lifting magnet, scoop shovel, dragline scraper or pile driver. Locomotive cranes with (usually) 8 wheels find widest application around industrial plants served by tracks. Besides lifting loads by hook block, they handle pig iron and scrap with a magnet, and shift cars.

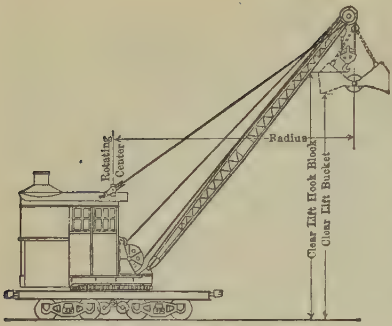


FIG. 8. Locomotive Crane

Steam-operated cranes require a boiler, fuel tank and water tank. Usually a licensed engineer is required to run the boiler. Many later cranes have gasoline or Diesel engines, which require less space, can be started or shut-down instantly, and have lower standby expense.

**Crawler Cranes** have, instead of wheels, a pair of treads made of steel plates linked together and driven by sprocket wheels. The weight is carried by a row of rollers on the lower frame. The width of tread plates and length of assembly make the unit bearing pressure so low that the crane can travel on soft, rough or uneven ground, and it requires no track or mats. Fig. 9 shows a machine built by Harnischfeger Corp., Milwaukee, Wis.

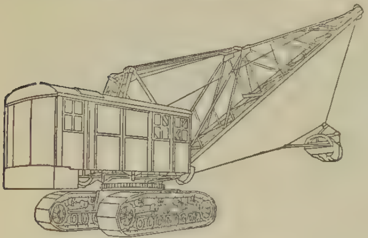


FIG. 9. Crawler Crane

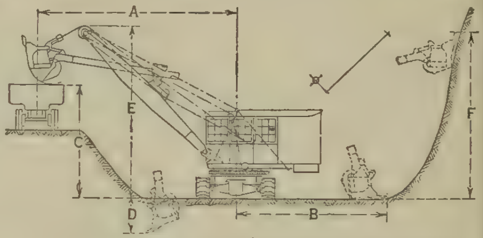


FIG. 10. Link-Belt Crawler Shovel (See Table 2)

Crawler cranes as made by several companies are fitted with digging shovels, dragline buckets, trench hoes, skimmer buckets, etc. and can be converted easily from one to the other. The following information is from Link-Belt Co., Chicago:

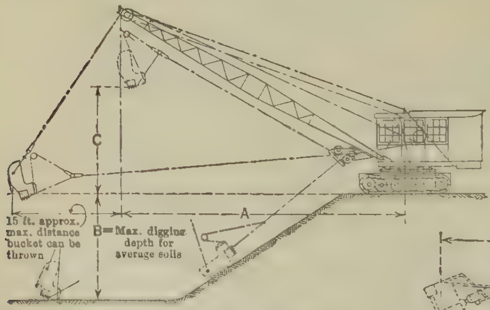

C
A
15 ft. approx. max. distance bucket can be thrown
B=Max. digging depth for average soils

FIG. 11. Dragline Crawler (See Table 3)

**The Shovel,** Fig. 10 is for general excavating, digging from the bank, handling blasted rock and for grading, including shallow cuts below grade. Its particular field is making cuts into and through banks, loading blasted rock on cars or trucks, where the machine stands in the hole or at base of the excavation. It can be used on loose piled materials, but if reach is long or lift high a grab bucket often works better. Table 2 gives range of work for two sizes of machines.

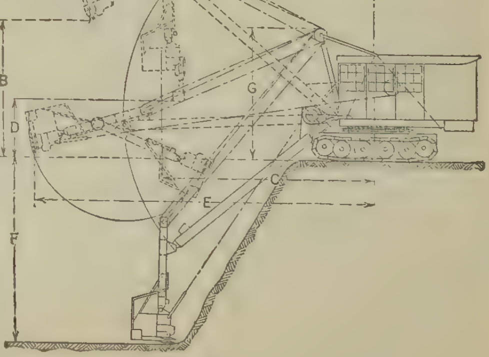


FIG. 12. Trench Hoe (See Table 4)

**The Dragline,** Fig. 11, is for removing overburden, digging under water, cutting wide or deep trenches, and where it is necessary to work above material handled. Dragline excavators work outside the hole, and when fitted with long booms can cover a large area, pile material high, and load cars some distance away. The dragline bucket is not so well suited to side cutting as the trench hoe. Table 3 shows the range of work for two sizes of dragline excavators. For dragline buckets, see Table 5.

**The Trench Hoe,** Fig. 12, is used to dig trenches up to 12 ft. wide and 20 ft. deep in earth that can be cut and which will not cave in. It is faster than the dragline, can be more accurately controlled and will cut a greater length of trench in a given time. Table 4 shows the range of work for two sizes.

**The Skimmer Bucket** is a form of shovel, comprising an open-end box armed with teeth. It travels back and forth on the underside of the crane boom and is used for grading, leveling, tearing up old pavements, etc.

Locomotive and crawler cranes with hoisting capacities up to 15,000 lb. at 40 ft.

### Table 2.—Dimensions and Capacities of Crawler Shovels

(Link-Belt Co., Chicago)

| Clearance Dimensions (See Fig. 10) | | | | |
|---|---|---|---|---|
| Dimension | Std. Lift Shovel, 20 ft. Boom, 15 ft. Stick | | High Lift Shovel, 35 ft. Boom, 21 ft. Stick | |
| | 40° Boom Angle | 60° Boom Angle | 40° Boom Angle | 60° Boom Angle |
| A | 25′–6″ | 20′–6″ | 35′–6″ | 32′–0″ |
| B | 17′–0″ | 15′–6″ | 24′–0″ | 21′–0″ |
| C | 13′–6″ | 19′–6″ | 16′–6″ | 24′–6″ |
| D | 6′–0″ | 4′–0″ | 10′–6″ | 8′–0″ |
| E | 18′–6″ | 23′–0″ | 29′–0″ | 37′–0″ |
| F | 20′–0″ | 26′–6″ | 25′–6″ | 35′–0″ |

| Approximate Operating Speeds, Dipper Pull, etc., for Shovel Service | | |
|---|---|---|
| Size of shovel * | K 25 | K 55 |
| Dipper hoist line speed, ft. per min | 175 | 200 |
| Max. dipper pull, 2-part line, lb | 24,000 | 44,000 |
| Rotating speed, r.p.m | 4.2 | 3.5 |
| Travel speed, miles per hr | 1.1 | 0.9 |
| Machine will climb grade of | 30% | 30% |
| Operating weight, complete, lb | 60,000 | 145,000 |
| Size of dipper, struck measure, cu. yd | 3/4 | 2 |

* 4 intermediate sizes are built.

### Table 3.—Dimensions and Capacities of Crawler Dragline Cranes

(Link-Belt Co., Chicago)

| Clearance Dimensions (See Fig. 11) | | | | | | |
|---|---|---|---|---|---|---|
| | 40′–0″ Boom | | | 70′–0″ Boom | | |
| Angle of Boom | *A*, ft. | *B* (approx.), ft. | *C*, ft. | *A*, ft. | *B* (approx.), ft. | *C*, ft. |
| 35° | 37 | 18 | 18 | 63 | 31 | 34 |
| 30° | 39 | 19 | 16 | 67 | 33 | 30 |
| 22° | 41 | 20 | 11 | 71 | 35 | 21 |

| Operating Speeds, Line Pull, etc., Dragline Bucket Service | | |
|---|---|---|
| Size | K 25 | K 55 |
| Dragline speed, ft. per min | 120 | 130 |
| Maximum dragline pull, lb | 19,000 | 37,000 |
| Hoist line speed, ft. per min | 140 | 155 |
| Rotating speed, r.p.m | 3.2 | 2.7 |
| Travel speed, miles per hr | 0.9 | 0.75 |
| Will climb grade of | 35% | 33% |
| Operating weight complete, lb., no bucket or load | 57,000 | 128,000 |
| Bucket size recommended | 3/4 *; 1/2 † | 2 1/4 †; 1 1/4 ‡ |

* 40 ft. boom. † 50 ft. boom. ‡ 70 ft. boom.

### Table 4.—Dimensions and Capacities of Trench Hoes

(Link-Belt Co., Chicago)

| Clearance Dimensions (See Fig. 12) | | |
|---|---|---|
| Size | K 30 | K 55 |
| A—(Door closed) | 27′–0″ | 33′–0″ |
| B—( " " ) | 25′–0″ | 23′–0″ |
| C—(Door open) | 21′–0″ | 25′–0″ |
| D—( " " ) | 12′–0″ | 11′–0″ |
| E | 35′–0″ | 43′–6″ |
| F | 20′–0″ | 22′–0″ |
| G | 17′–0″ | 23′–0″ |

Three intermediate sizes are built.

| Speeds, Capacities, etc. | | |
|---|---|---|
| | K 30 | K 55 |
| Drag-in speed, ft. per min | 90 | 105 |
| Max. bucket drag-in pull, lb | 24,000 | 40,000 |
| Hoist line speed, ft. per min | 150 | 150 |
| Rotating speed, r.p.m | 3.5 | 2.7 |
| Travel speed, miles per hr | 0.9 | 0.75 |
| Will climb grade of | 30% | 30% |
| Operating weight, lb | 67,000 | 144,000 |
| Size of bucket, cu. yd | 1–1 1/4 | 2–2 1/2 |

radius or with shovel capacity up to 3 cu. yd. are made (see p. 23–81) by [14], [39], [41], [50], [63], [64], [66]. Smaller cranes mounted on wide faced wheels or crawler treads with shovels of less than 1 cu. yd. are made for road builders and contractors by [16], [60], [87], and others. Some of these cranes cannot swing a full circle. Large excavators with shovel capacity of 4, 6, 9 cu. yd. are made by [15], [53], [67].

For a comparison of these large excavators with the chain and bucket ladder excavators used in Germany, see *Fördertechnik*, Oct. 26, 1928, May 8, 1931, Aug. 26, 1932.

**STORAGE SYSTEMS USING LOCOMOTIVE CRANES,** Fig. 13, shows a system of storing coal in semi-circular piles. Drop-bottom cars discharge to a central pit around which as a center are broad gage tracks for a long-boom crane. In large plants, working radius is 100 ft. The grab bucket has a capacity of 1 to 4 tons of coal which it digs from the pit and drops on the pile; in reclaiming, it digs from the pile and drops into cars. With coal piled 25 ft. deep a crane of 100 ft. radius will store 70,000 tons in a full circle with crane tracks covered. By continuing the tracks into more loops successive storage piles can be formed by one crane. The handling capacity, depending on size of bucket, is from 100 to 250 tons per hr.

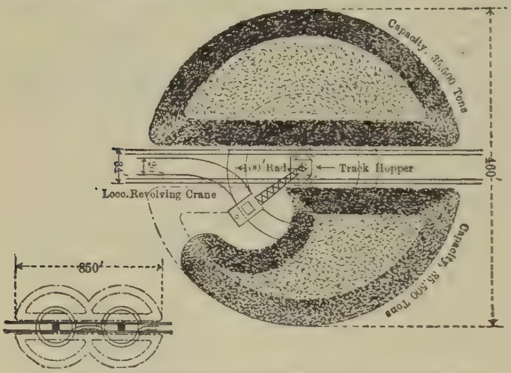


FIG. 13. Semi-circular Pile System of Coal Storage

Locomotive shovel cranes and crawler shovels are used with belt conveyors to remove excavated material, or to build levees. Dragline cranes, feeding a system of belt conveyors in tandem, are used in building levees along the Mississippi River. In 1929, over 6,000,000 cu. yd. were dug from a hill in the middle of Seattle, Wash., by crawler shovels, dumped into feed hoppers, thence to belt conveyors, some of them portable, and carried 3000 ft. for disposal. For the system used in building the Grand Coulee dam see *Engg. News-Record*, Aug. 1, 1935.

## 3. GRAB BUCKETS

**GRAB BUCKETS.**—Most rope-operated grab buckets, clam-shell buckets or orange-peel buckets work on the same principle, *viz.*, an upper head, and movable, relative to it, a lower head. The holding line, attached to the upper head, sustains the weight of the bucket when it opens. The lower head is connected to the bowl of the bucket and the closing line, so that when this line slacks the bucket opens. When it is pulled the bucket closes. Various styles of bucket differ in the manner of reeving closing line through sheaves or around a drum (power wheel) or of attaching it to arms or a crosshead so that pulling the closing line will cause the digging edges of the bowl to dig material and fill the bucket. See Fig. 14. Very heavy buckets may be operated with two holding and two closing lines to reduce size of rope, sheaves and drum. Fig. 15 is a diagram of two holding lines run from drum $H$ and clipped to an equalizer lever at the top of the bucket. The closing line runs from drum $C$, around sheaves $U$ and $L$ in the bucket, thence to an anchorage or back to the drum. This is the "fleeting through" method of reeving. If the crane handles a hook block or buckets of different sizes, the bucket is readily detachable if the two closing lines in the bucket are dead-ended, instead of being in one length.

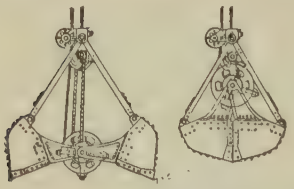


FIG. 14. Grab Bucket

The following is the cycle of operation of any rope-operated bucket: Assume bucket to be in upper position, Fig. 15, empty and closed; the weight is on the closing line. Drum $C$ turns to lower bucket; drum $H$ pays out holding line at the same rate of travel. As the bucket approaches material to be dug, $H$ is braked, $C$ pays out closing line, the weight of the parts $L$ pushes hinge of the bucket down, the halves open and the bucket is ready to bite. Both lines then are let go, and the bucket drops on the material. With holding line slack, the weight of the bucket resting on its digging lips, the closing line is pulled in; upper head *B* sinks, bucket closes, digs and fills. To raise bucket, *C* turns to pull in closing line, which now takes the weight; *H* pulls in the holding line. If the loaded bucket hangs from a trolley or carriage both drums are braked while the carriage moves. To lower the loaded bucket, both drums, under brake control, pay out line. To open the bucket, *H* is braked or stopped, *C* pays out closing line, lower head *A* sinks, and bucket opens. In its traverse back to the loading point, both drums are braked and the bucket stays open.

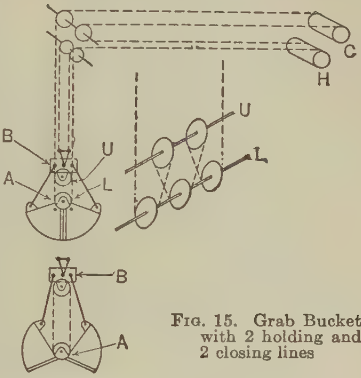


Fig. 15. Grab Bucket with 2 holding and 2 closing lines

The choice of a grab bucket depends chiefly on the material to be dug. Buckets with half-round bowls, Fig. 14, will dig loose free-flowing material and coal that is not too lumpy. To dig material that is lumpy or packed hard, the open bucket should have its digging edges in position to penetrate the material and not skid over it.

Digging buckets are made (see p. 23–81) by manufacturers [12], [40], [41], [92]. Table 5 shows heavy duty buckets made by Blaw-Knox Co., Pittsburgh, Pa. Similar buckets up to 7 cu. yd. size are made to handle quarry rock, blast furnace slag and other difficult materials. Buckets of special shapes are made to rehandle stone from gondola cars, ores from the hold of steamers, and to excavate under water.

Orange-peel buckets made by Hayward Co. and others have a hemispherical bowl divided into sections fitted with digging points. Holding, opening and closing is the same

**Table 5.—Heavy Duty Buckets for Excavating or Hard Digging**
(Blaw-Knox Co., Pittsburgh, Pa.)

| Nominal rating, cu. yd. | 3/8 | 1/2 | 1/2 | 3/4 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 |
|---|---|---|---|---|---|---|---|---|---|
| Capacity, heaped, cu. ft. | 18.67 | 22.07 | 25.66 | 31.50 | 40.30 | 50.37 | 62.30 | 70.50 | 83.35 |
| Deck covering area, sq. ft. | 12.50 | 14.97 | 16.15 | 22.31 | 26.40 | 30.68 | 34.14 | 38.33 | 41.36 |
| Width—overall | 2′–0″ | 2′–5″ | 2′–8″ | 3′–2″ | 3′–6″ | 3′–9″ | 4′–0″ | 4′–2″ | 4′–4″ |
| Length—open | 6′–3″ | 6′–3″ | 6′–5″ | 7′–0″ | 7′–7″ | 8′–2″ | 8′–8″ | 9′–2″ | 9′–6″ |
| Length—closed | 5′–7″ | 5′–7″ | 5′–9″ | 6′–0″ | 6′–5″ | 6′–11″ | 7′–4″ | 7′–9″ | 8′–0″ |
| Height—open | 7′–11″ | 7′–11″ | 8′–1″ | 8′–6″ | 9′–9″ | 10′–6″ | 10′–11″ | 11′–9″ | 12′–3″ |
| Height—closed | 6′–4″ | 6′–4″ | 6′–7″ | 7′–2″ | 8′–1″ | 8′–8″ | 9′–0″ | 9′–9″ | 10′–2″ |
| Size of cable | 1/2″ | 1/2″ | 5/8″ | 3/4″ | 3/4″ | 3/4″ | 3/4″ | 7/8″ | 7/8″ |
| Line to reeve open bucket (4 parts) | 32′–9″ | 33′–9″ | 34′–0″ | 35′–4″ | 40′–6″ | 43′–8″ | 45′–4″ | 48′–10″ | 51′–0″ |
| Line pulled out to close bucket | 22′–8″ | 23′–2″ | 23′–2″ | 24′–2″ | 26′–5″ | 28′–6″ | 30′–10″ | 31′–10″ | 33′–3″ |
| Weight—pounds * | | | | | | | | | |
| Without counterweights—Weight A | 1900 | 2200 | 2600 | 3300 | 4300 | 4900 | 6000 | 6500 | 7200 |
| With Counterweight—Weight B | 2200 | 2615 | ..... | 3580 | 4500 | ..... | ..... | ..... | ..... |
| With Counterweight—Weight C | ..... | 2735 | 3000 | 3875 | 4750 | ..... | ..... | ..... | ..... |
| PICK-UP CAPACITY, CU. FT. FOR MATERIALS LISTED IN COLUMN 1 *† | | | | | | | | | |
| Sand, slack coal, granulated blast furnace slag, gravel, 1/2- and 1 1/2-in. stone, dolomite, cement clinker, ground asphalt, acid phosphate | 13–B | 17–A | 20–A | 26–A | 35–A | 45–A | 56–A | 65–A | 75–A |
| Ordinary excavating, earth, clay and gravel | 18–BT | 22–BT | 25–CT | 31–BT | 38–AT | 48–AT | 61–AT | 70–AT | 82–AT |
| Hard clay, cementitious gravel, hard-digging earth with boulders, broken shale | 11–BT | 18–CT | 19–CT | 23–CT | 28–AT | 34–AT | 42–AT | 48–AT | 56–AT |
| Iron ore, 4-in. and 5-in. stone | ..... | ..... | ..... | ..... | 28–A | 34–A | 42–A | 47–A | 54–A |

**A* = No weights added; *B*, *C* = Digging capacity attained by adding weight to bucket; *T* = digging teeth required.

† Similar buckets to handle quarry rock, blast furnace slag, etc., are made in sizes up to 7 cu. yd. capacity.

**Table 6.—Weights and Dimensions of Single-line Buckets**

(G. H. Williams Co., Erie, Pa.)

| Capacity, cu. yd. | Approx. Weight, lb. | Overall Dimensions | | | | |
|---|---|---|---|---|---|---|
| | | Height, Closed | Height, Open | Length, Closed | Length, Open | Width |
| 1/2 | 2600 | 6′–3″ | 6′– 9″ | 4′– 6″ | 6′–9″ | 2′–9″ |
| 3/4 | 2800 | 6′–6″ | 6′–10″ | 5′– 0″ | 7′–3″ | 2′–9″ |
| 1 | 3400 | 7′–0″ | 8′– 0″ | 5′– 8″ | 7′–8″ | 3′–3″ |
| 1 1/4 | 3700 | 7′–3″ | 8′– 0″ | 5′– 8″ | 8′–2″ | 3′–6″ |
| 1 1/2 | 4800 | 8′–6″ | 9′– 3″ | 6′–10″ | 9′–3″ | 3′–9″ |

as in clam-shell buckets operated by two lines. They are made in sizes up to 1 1/2 cu. yd.; also in very small sizes that can be used in digging test-holes and holes for concrete piles.

Single-line buckets are made in the smaller sizes to rehandle loose material from cars or stock piles or for light excavating. They are hung on a single rope which runs from the hoist drum of a crane or derrick, through sheaves on the bucket to a closing-arm which pulls the halves of the bucket together. A latched arm holds bucket closed; it is tripped open by a hand line. It is not possible to lower a single-line bucket to a pile and then dump or discharge a part of its contents. Table 6 shows the usual sizes.

**Electric Motor Buckets.**—The Hayward electric motor bucket differs from all others. It uses an electric motor with a compact geared hoist in the upper head. Winding up a chain raises the lower head and closes the bucket; reversing the motor opens the bucket. A slip clutch prevents overload on the motor if the current is not cut off when the bucket is closed. Current is supplied through a flexible cable, which, when the travel is considerable, is automatically wound and unwound from a drum operated by an internal coiled spring, and with collector rings on the end of its axle. An electric controller, hand-operated, starts, stops, or reverses the motor to close, hold or open the bucket. This bucket can be hung on a single hoisting rope, or from the hook of a rope tackle used with a single drum, or from the hook of an ordinary hoist or crane.

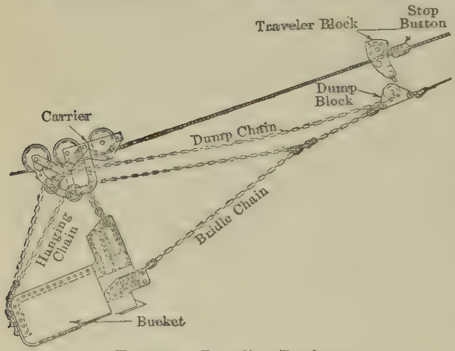


Fig. 16. Dragline Bucket

**DRAGLINE BUCKETS** are used with locomotive cranes, Fig. 11, or slackline cableways. The bucket is an open end box with a digging lip, with or without teeth. The dragline is hitched by a pair of chain bridles with clevis connections to front of bucket. When used with a locomotive crane, the bucket is pulled into dumping position by a line from one of the drums over the end of the boom, and hitched to the back end of bucket. In the slack cable system, the bucket, hung from the carriage, runs up the track cable with a traveler block, hits a stop button; further motion of the hauling rope pulls on the dump chain hitched to back of bucket. See Fig. 16. Dragline buckets are made (see p. 23–81) by manufacturers [12], [40], [79], [90], and others.

## 4. LIFTING MAGNETS

**LIFTING MAGNETS** are used with bridge cranes, monorail carriers, locomotive cranes and derricks to handle iron and steel of all kinds, sizes and shapes, hot or cold. In general the only labor needed is that of the crane man, who operates both crane motor and the magnet controllers. A magnet, in combination with a bridge crane, can pile steel higher and more compactly than any other means; cars can be unloaded more quickly and demurrage avoided; magnets will handle steel hot from the roll and will work under water.

A lifting magnet consists of a circular or rectangular cast-steel case enclosing copper coils, inner and outer pole shoes to form contact with the material lifted, and a non-magnetic shield to protect the coils from the material. Rectangular magnets have less lifting power than circular magnets. They lift only what they actually touch, but round magnets will operate through an air gap of 5 to 15 in. according to their size.

Fig. 17 is a cross-section of a round magnet made by Cutler-Hammer, Milwaukee,

**Table 7.—Average Lift, Pounds, for All-day Service of Lifting Magnets**
(Cutler-Hammer, Milwaukee, Wis.)

| Size of magnet, in. diam.* | 65 | 55 | 45 | 39 | 24 |
|---|---|---|---|---|---|
| Skull cracker balls | 40,000 | 30,000 | 24,000 | 18,000 | 6000 |
| Slabs | 60,000 | 50,000 | 35,000 | 20,000 | 6500 |
| Machine-cast pig from cars | 2,800 | 2,280 | 1,275 | 750 | 250 |
| Sand-cast pig from cars | 2,650 | 2,170 | 1,150 | 680 | 225 |
| Broken scrap | 1,800 | 1,300 | 1,000 | 600 | 200 |
| Cast-iron borings | 2,300 | 1,500 | 1,100 | 600 | 200 |
| Steel turnings | 1,100 | 750 | 450 | 300 | 100 |
| Principal Uses | Large capacity; heavy duty | Open-hearth plants; steel mills | Gen. work; larger cap'y with loco. cranes | Pig. iron; scrap; med. cap'y with loco. cranes | Light finished parts; pipe; castings |

* Round magnets 18 in., 12 in., 7 in., 5 in. diam., also are made.

and Table 7 gives capacity per lift for different kinds of service for average 10 hr. steady work. If service is not continuous, the coils run cooler and the lifting capacity is greater.

Round magnets and magnets of other shapes are also made by Ohio Elec. Mfg. Co., Cleveland. Fig. 18 shows 3 rectangular magnets hung from a spreader beam for lifting long plates.

To provide direct current needed for a magnet, locomotive cranes are fitted with a generating set in the cab. On alternating-current bridge cranes, a motor-generator set on a platform over the operator's cage supplies direct current. The trolley has a cable winding drum for taking up or paying out electric cable to the magnet.

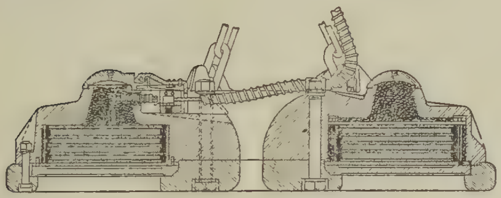


Fig. 17. Lifting Magnet

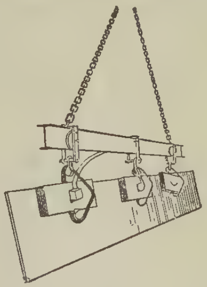


Fig. 18. Lifting Magnet, Handling Long Plates

Round magnets often are hung over a belt conveyor to remove stray iron from coal, foundry sand, etc., carried on the belt. The diameter of the magnet should be $1/3$ larger than the belt width, and it should not be higher above the belt than $1/4$ the belt width. Belt speed should not exceed 450 ft. per min.

## CRANES AND DERRICKS

A hoist is a machine for raising and lowering loads. A crane is a hoist or number of hoists so mounted as to move loads both vertically and in other directions.

**SAFETY STANDARDS FOR CRANES, DERRICKS AND HOISTS.**—The safety code sponsored by the A.S.M.E. and the A.S.A. (1933) defines terms, states requirements of design and construction and gives rules to promote safety in operating.

Cranes generally are tested by their makers with a load 25% greater than the load for which they are rated.

### 1. TYPES OF CRANES

Cranes which lift and swing the load are pillar cranes, jib cranes in various forms, derricks, rotary bridge cranes, locomotive cranes. Cranes which lift the load and travel with it comprise many forms of bridge and gantry cranes.

**A PILLAR CRANE** consists of a stiff self-supported column, turning on a vertical axis and fitted with an inclined boom, the end of which carries the load. The boom swings in a full circle on ball or roller bearings. Hoisting is by hand power winch on the column. Fig. 1 shows a type made in sizes of 2 to 20 tons; working radius 12, 15 or 20 ft.

**Pillar Jib Cranes** have a trolley traveling on a horizontal boom to carry the load. The hoist is hand or power; the trolley usually is moved by hand. The jib may be two channels or a single I-beam for chain, electric, or air hoist. See Fig. 2.

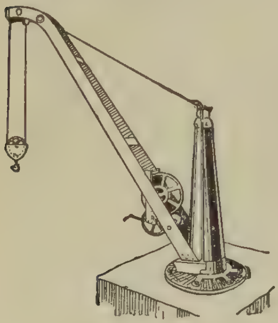


Fig. 1. Pillar Crane

## Jib Cranes

**TOP-SUPPORTED JIB CRANES** require the top of the column to be braced to a wall or ceiling, or by guy-ropes or struts. They include yard derricks, foundry cranes and sidewall cranes for machine shops. Fig. 3 shows one with separate hand winches for hoist and traverse. Jib cranes sometimes are made with single I-beam jib for hoists hung from 4-wheel trolleys as auxiliaries to traveling bridge cranes.

**PORTABLE JIB CRANES** can be made to serve different areas of shop floor as needed, by fitting building columns with socket connections to fit loose-pin connections on the crane mast.

**BRACKET JIB CRANES** have a horizontal jib hinged to a wall bracket by a vertical pin, and stayed by a rod hinged to an upper bracket. The form shown in Fig. 4 has an I-beam jib for a trolley, from which hangs a hand or power hoist. Bracket jib cranes also are made with built-up girder rail jibs for Cleveland tramrail or American monorail. See p. 23–44. The ends of the special rail on the jib can be fitted with interlocks to engage with fixed tramrail. Trolleys on the jib thus also may run on tramrails radiating from the bracket crane, and serve a large area. To serve trolley tracks running through a doorway, Richards-Wilcox Co., Aurora, Ill., and others, make bracket jib cranes hinged to lie flat against the wall and allow the door to close.

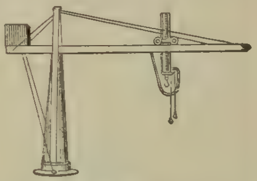


Fig. 2. Pillar Jib Crane

**WALKING CRANES** and **Traveling Wall Cranes** are jib cranes mounted for longitudinal travel; the first runs on a track on the shop floor with an upper rail on the wall or building columns to take the thrust at the top of the mast. The wall crane, Fig. 5, (Niles-Bement-Pond Co., New York) has an upper and a lower thrust track on which run flat-faced rollers, while the vertical load is taken on a separate rail mounted on wall brackets. The thrust load is much greater than the dead load when the trolley is at the extreme end of the crane. Such cranes act as auxiliaries to shop bridge cranes in serving long lines of machine tools or in handling riveters on long structural steel work.

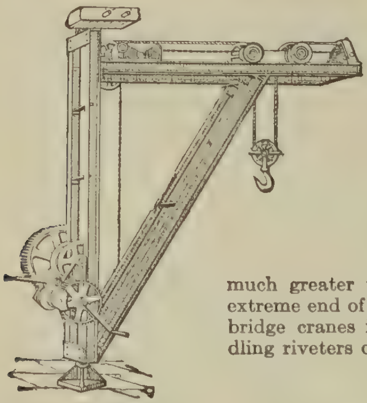


Fig. 3. Top Supported Jib Crane

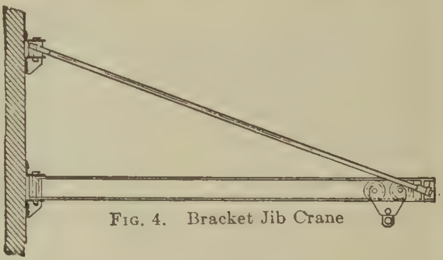


Fig. 4. Bracket Jib Crane

## Derricks

**GUY DERRICKS** have a mast, with a step and vertical pivot pin, at the base, a boom hinged to the foot of the mast, whose inclination can be varied by rope tackle joining tops of boom and mast. A pivot pin on top of the mast fits a cap to which guy ropes are attached. For hand operation, a double-drum winch is mounted at the base of the mast for hoisting and luffing respectively. If angle of boom is not often changed, a single-drum winch will suffice, luffing being by a hand line hitched to the mast. Fig. 6 shows a guy derrick used in quarries and by contractors in bridge and building construction. It is operated by steam or power hoist with two hoisting drums for load and luffing, and a pair of swinging drums. Ropes from the latter lead to opposite sides of a bullwheel, on the foot of the mast, to swing mast and boom. The luffing rope runs to foot of mast, to masthead, thence to tackle between masthead and top of boom. The hoisting rope follows a similar path, and runs through sheaves on the boom to the suspended load. The stress in the hoisting line (fall rope) can be taken from Table 4, p. 23–04. Stress in boom-hoist tackle increases as boom is lowered. Hoisting rope is generally 6 × 19 steel wire rope; boom hoist line is the same, but with more parts in the tackle. Stresses in guy lines can be calculated from their arrangement around the mast with respect to the position of the load and also with respect to the angle they make with the horizontal. For other points on derrick design, see Amer. Cable Co. Wire Rope Users' Handbook. Contractors' derricks often are made of steel for safety, and in sections for ease in handling.

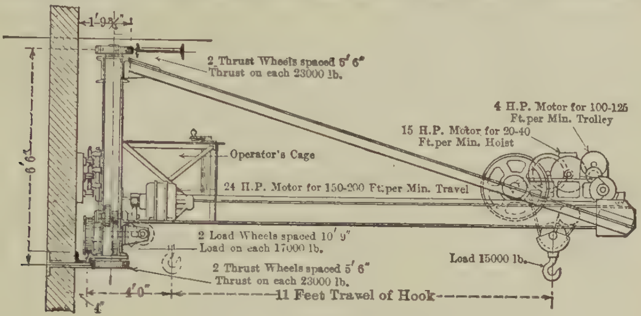

1′ 9½″
2 Thrust Wheels spaced 5′ 6″
Thrust on each 23000 lb.
4 H.P. Motor for 100-125 Ft. per Min. Trolley
15 H.P. Motor for 20-40 Ft. per Min. Hoist
Operator's Cage
24 H.P. Motor for 150-200 Ft. per Min. Travel
6′ 6″
2 Load Wheels spaced 10′ 9″
Load on each 17000 lb.
Load 15000 lb.
2 Thrust Wheels spaced 5′ 6″
Thrust on each 23000 lb.
4′ 0″
11 Feet Travel of Hook
4″

Fig. 5. Traveling Wall Crane

**STIFF-LEG DERRICKS** are used where it is not feasible to run guy ropes, or where derrick must be shifted often. See Fig. 7. Struts from masthead, inclined at 45°, are framed, or hinged, to two sills set on the ground at 90°, meeting under the step of the mast. The boom can swing 270°. Sills can be bolted to foundations or held down by weights.

**JINNIWINK DERRICKS** are similar to stiff-leg derricks. An A-frame replaces the mast, and with its back strut is mounted on a T-shaped frame acting as a base. The boom, hinged to the base frame, is adjustable for angle. These derricks, light and easily set up, are used by building contractors.

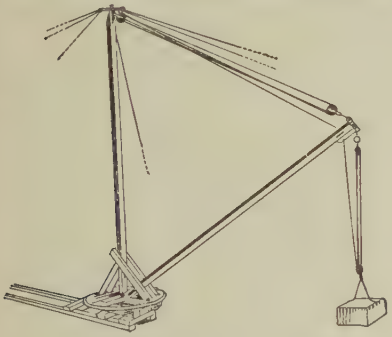


Fig. 6. Guy Derrick

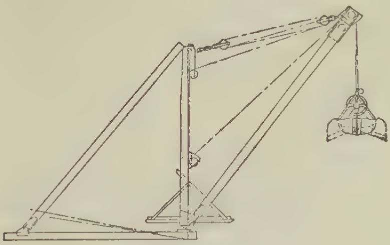


Fig. 7. Stiff-leg Derrick

**TOWER DERRICKS** consist of a square tower of wood or steel with footings for booms on one or more corner posts. They are used in building construction, booms being shifted higher on the posts as the building goes up. A steel tower derrick with a revolving pivoted counterweighted 50-ft. boom is used in building construction in Germany. It will lift 1 ton with boom horizontal; maximum height of lift with boom luffed, 100 ft. The tower rests on a wheeled platform, and has 3 small electric winches, controlled by a man on the platform.

The concrete distributing apparatus made by Insley Mfg. Co., Indianapolis, Ind., is a modified mast and tower derrick. The mast hoist has a framed steel mast, about 20 in. square, made in 20-ft. sections with bolted end connections and guy attachments. It has guides for a dump bucket which discharges into a receiving hopper adjustable in height on the mast, from which chutes distribute concrete. The upper end of the chute is a boom, guyed from the mast head. Extensions of the chute hang from trolleys on a chute-extension cable which spans the area over which concrete is distributed. Fig. 8 shows a 60-ft. mast with a 40-ft. boom chute, served by a 1-yd. hoist bucket. Larger plants have steel or wood guyed towers, with one or two hoist buckets. Masts and towers for this system of distributing concrete have been designed and standardized for rapid and easy erection.

Complete steel derricks or fittings for wooden derricks are made by manufacturers [3], [21], [49], [52] (see p. 23-81) and by makers of steam, electric or gasoline hoists and of equipment for contractors.

## Traveling Cranes

**TRAVELING BRIDGE CRANES.**—Makers of tramrail equipment (see p. 23–81) list and illustrate in their bulletins a wide variety of light cranes for 2 tons or less load. Table 1 classifies Cleveland tramrail bridge cranes; similar cranes are made for American monorail. These light cranes are notable for several reasons: 1. End trucks have equalizer levers and swivel wheels; wheel flanges do not bind on the rail. 2. Structure is light; wheels have ball bearings; crane is easy to move. 3. The light construction of bridge, flexible mounting of the wheels, and absence of a squaring-shaft permit cranes to be hung from 3 runways if bridge is longer than 30 or 35 ft. By using 3 runways weight of the bridge can be kept so low that a 50-ton crane can be moved by pushing or pulling the load, and there is no danger of pinching or binding the runway. 4. Most types can be fitted with interlocks to register bridge rails with fixed tramrail, so that hoists, carriers and trailers can travel on or off crane bridge. 5. Electrical conductors and mountings, control and accessories are well standardized. See Fig. 9.

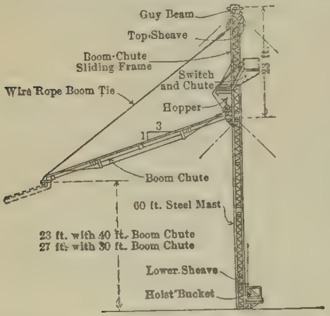


Fig. 8. Tower Derrick and Boom Chute

Other bridge cranes, for loads up to 5 tons and for spans under 30 ft., are made of single I-beam spans with trolley hoists hung from lower flanges. The bridge travel of these cranes is generally by hand-chain hung from a chain-wheel on a squaring shaft that drives truck wheels at each end of the span. For loads over 3 tons, it is better to have a gear reduction between chain gear and squaring shafts.

**Double I-beam Cranes** are used for loads up to 20 tons and for spans of 20 to 50 ft. The trolley runs on top of the girders. The hoist rope is wound on a drum which may be geared to hoist light loads at faster speed than heavy loads. With double I-beams, the load hook can go between the beams, and for a given lift, the building need not be so high as it would be if trolleys were hung below the beam. Some cranes have two trolleys for handling long or heavy pieces.

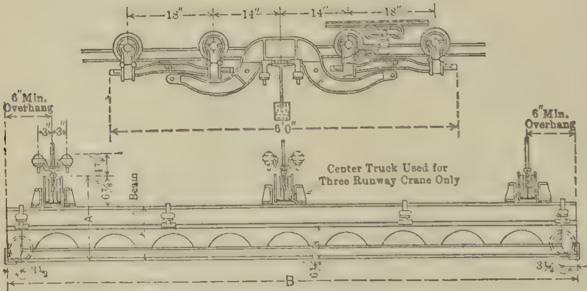


Fig. 9. Cleveland Tramrail Crane

The hand-power cranes mentioned are for what crane builders call class D, *i.e.*, very intermittent, service. If hand operation is too slow, I-beam cranes can be fitted with one or two motors (class C service), or with three motors for class B service, which means the ordinary work of a crane in shops, foundries, yards, etc., where the crane is busy 8 or 10 hr. and adjustments, inspections and repairs can be made in idle time.

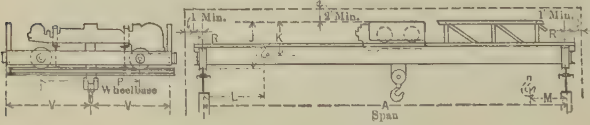


Fig. 10. Light-duty Bridge Crane

**A 3-motor Bridge Crane** for class B service is shown in Fig. 10 and Table 2. The bridge is made of two I-beams to carry the load, with two light lattice-girders to stiffen the frame and to help carry the travel machinery. The bridge often can be shipped and erected as a unit. The trolley frame is a one-piece steel casting; the double-flanged

**Table 1.—Cleveland Tramrail Bridge Cranes**
(Cleveland Crane & Engg. Co., Wickliffe, O.)

| Type | Bridges Propelled by Hand (Pull or Push Load) |
|---|---|
| CH | For any Cleveland carrier or hoist, hand or electric up to 2 tons capacity; 2 runways for bridge lengths up to 30 ft.; 30 to 50 ft. 3 runways. |
| CHL | For any Cleveland carrier or hoist, hand or electric, up to 2 tons capacity; 2 runways for bridge lengths up to 30 ft.; low headroom. |
| DGH | Double girder bridge; trolley hoist on top. Capacities up to 1 ton; 2 runways for bridge lengths up to 30 ft.; low headroom. |
| | Bridges Propelled by 2 or 3 Motors |
| UCE | For any Cleveland carrier or hoist, hand or electric up to 2 tons capacity, 2 runways for bridge length up to 30 ft.; for 30 to 50 ft. 3 runways. |
| DGE | Double girder bridge; hand pushed trolley; capacities up to 2 tons; 2 runways for bridge lengths up to 30 ft.; low headroom. |
| DGEE | Same as DGE except that trolley is motor-driven. |
| DGC | Like DGEE, but operator controls all motions from cab mounted on bridge. |

**Table 2.—Clearance Dimensions for Light Duty Cranes**
(Shaw Electric Crane Co., Inc., Muskegon, Mich.)

| Capacity, tons | Dimensions (see Fig. 10) | | | | | | | | Runway Rail, lb. per yd. | Max. Wheel Load, lb. |
|---|---|---|---|---|---|---|---|---|---|---|
| | *A*, ft. | *J* | *K* | *L* | *M* | *P* | *R* | *V* | | |
| 1 | 20 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 4680 |
| | 30 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 5120 |
| | 40 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 6240 |
| | 50 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 7150 |
| 2 | 20 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 5600 |
| | 30 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 6070 |
| | 40 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 7200 |
| | 50 | 2′–2″ | 1′–6″ | 1′–8″ | 1′– 8″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 8120 |
| 3 | 20 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 6880 |
| | 30 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 20 | 7400 |
| | 40 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 8540 |
| | 50 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 9470 |
| 3 1/2 | 20 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 7380 |
| | 30 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 7900 |
| | 40 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 9040 |
| | 50 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 9970 |
| 5 | 20 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 8730 |
| | 30 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 9250 |
| | 40 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 10400 |
| | 47 | 2′–7″ | 2′–2″ | 2′–1″ | 1′–11″ | 5′–6″ | 5 1/2″ | 4′–9″ | 30 | 11300 |
| 7 1/2 | 20 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 30 | 12000 |
| | 30 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 30 | 12700 |
| | 40 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 30 | 14000 |
| | 45 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 30 | 14900 |
| 10 | 20 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 30 | 14200 |
| | 30 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 30 | 15000 |
| | 40 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 40 | 16200 |
| | 42 | 3′–6″ | 3′–0″ | 3′–1″ | 2′– 0″ | 5′–9″ | 6″ | 5′–6″ | 40 | 17000 |

Crane Ratings

| Capacity | Full Load Speeds, ft. per Min. | | | Hp. of Motors | | | Diameter of Rope, in. |
|---|---|---|---|---|---|---|---|
| | Hoist | Trolley | Bridge | Hoist | Trolley | Bridge | |
| 1 ton | 42 | 110 | 175 | 3 | 3/4 | 3 | 1/4 |
| 2 tons | 35 | 110 | 175 | 5 | 3/4 | 3 | 1/4 |
| 3 tons | 33 | 110 | 175 | 7 1/2 | 1 | 3 | 3/8 |
| 3 1/2 tons | 28 | 110 | 175 | 7 1/2 | 1 | 3 | 3/8 |
| 5 tons | 26 | 110 | 175 | 10 | 1 | 3 | 3/8 |
| 7 1/2 tons | 24 | 110 | 175 | 15 | 2 | 5 | 9/16 |
| 10 tons | 18 | 110 | 175 | 15 | 2 | 5 | 9/16 |

trolley wheels run on flat bars welded to top flanges of the girder beam. These and similar cranes are made (see p. 23–81) by manufacturers [28], [41], [54], [61], [76], [80], [91].

Cranes for class A service work 24 hr. per day in steel mills, cement mills and foundries, or on heavy hoist and grab bucket or magnet work, where interruption of service is serious. Speeds of hoists, trolley and travel often are higher than for cranes in lighter service; motors are larger; parts are designed with a higher factor of safety and with more regard to access to working parts and certainty of lubrication. Such cranes are made by some of the companies noted above, also by (see p. 23–81) manufacturers [1], [20], [39], [59], [81]. Fig. 12 and Table 3 show cranes for heavy service. Cranes made by various makers do not differ much in essential dimensions.

**AUXILIARY HOISTS FOR CRANES.**—The dimensions *P* and *G* in Table 3 for cranes over 10 tons refer to position of hook of the auxiliary hoist, which is used for rapid hoisting of loads lighter than those for which the crane is designed. For this duty the trolley carries two motors, each with its own set of drums and gearing. One set is for the main hoist of the rated capacity of the crane; the other is for 3, 5 or 10 tons auxiliary capacity.

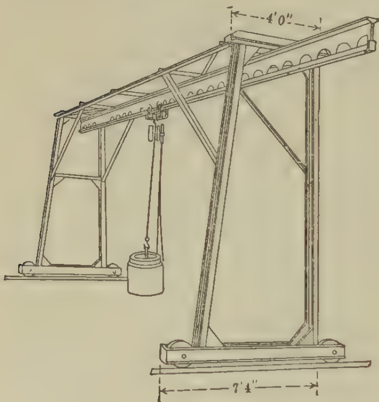


Fig. 11. Tramrail Gantry Crane

**CRANES FOR OPERATING GRAB BUCKETS** for handling coal at power plants, raw material and clinker at cement plants, unloading bulk material from cars, etc., have a trolley with two drums, one for closing line and one for holding line. The drums are mounted parallel to the bridge girders; each has its own motor, solenoid brake and reduction gearing. If the crane does miscellaneous hoisting and a bucket is used only occasionally, the trolley can be of the ordinary hoisting type, and the bucket of the single-line or one of the motor-driven types, to be put on the hook block when needed. See p. 23–20.

Grab bucket cranes are made by most builders of heavy type bridge or gantry cranes. Dimensions of trolley vary with weight and type of grab bucket and height of lift. Parts generally are of heavy construction, and precautions are taken to keep dust out of gears. All motions of crane and bucket are electrically controlled from the operator's cab, usually mounted on the trolley.

**TRENDS IN CRANE DESIGN.**—(*Trans.* A.S.M.E., 1928, MH–50–1) Modern cranes are characterized by less dead weight, by rolled beams instead of box girders, lighter motors, alloy steel parts, ball or roller bearings, heat-treated gears, gears enclosed for protection and running in oil baths, grease cups and intermittent oiling being avoided. Bridge cranes with structural parts of aluminum alloy are described in *Engg. News-Record*, Oct. 8, 1931. Advantages claimed are reduction of weight, less power for travel motions and no necessity of painting. Comment of *Fördertechnik* is to the effect that the percentage of weight saved would not be so great for German cranes as they are lighter than American cranes for the same service.

Cleveland Crane and Engg. Co. makes cranes in which end trucks, trolley frames and bridge girders are arc-welded and not riveted. Advantages claimed are greater rigidity and strength, less weight, faster acceleration, reduced maintenance.

**GANTRY CRANES.**—For loads up to 2 tons Cleveland tramrail can be fitted to two gantry legs. See Fig. 11. All wheels have ball bearings, and the structure, with load, can be moved by one man at either end pushing or pulling. For rapid or long travel the crane can be electrically propelled. These cranes can be used in shops as auxiliaries to traveling bridge cranes. Cleveland gantry cranes can be interlocked to register bridge rails with fixed tramrail so that hoists, carriers and trailers can move on and off the gantry bridge.

Gantry cranes of heavier type, Fig. 13, range in size from 5 to 250 tons, of any span. As regards construction, they are like bridge cranes built for hoist, bucket or magnet work, but have better protection from weather for operator and machinery. For travel motion, power is transmitted from a motor on the bridge through horizontal and vertical shafts to geared truck wheels at ground level. Truck should have brakes to prevent high winds moving the bridge. Such mechanical and electrical brakes are described in *Trans.* A.S.M.E., 1930, IS–52–4. If the crane serves a yard alongside a building it is sometimes convenient and economical to support one crane track on the building wall. The crane is then a *single-leg gantry.*

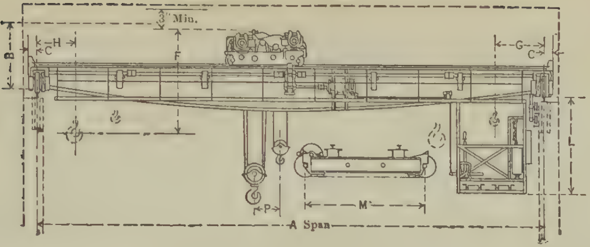


Fig. 12. Heavy-duty Bridge Crane

**Table 3.—Clearance Dimensions, Heavy-duty, Roller-bearing Cranes**
(Harnischfeger Corp., Milwaukee, Wis.)

| Capacity, tons | Span A, ft. | Dimensions (see Fig. 12) | | | | | | | | Max. Load per Wheel, lb. | Runway Rail, lb. per yd. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | B | C | F | G | H | L | M | P | | |
| 5 | 40 | 5′–2 1/2″ | 8 1/2″ | 6′– 1″ | 2′–10″ * | 2′– 9″ | 8′–8″ | 9′–10″ | ....... | 14500 | 50 |
| | 50 | 5′– 1″ | 8 1/2″ | 6′– 1″ | 2′–10″ * | 2′– 9″ | 8′–8″ | 10′– 0″ | ....... | 16000 | 50 |
| | 60 | 5′– 1″ | 8 1/2″ | 6′– 1″ | 2′–10″ * | 2′– 9″ | 8′–8″ | 10′– 0″ | ....... | 17300 | 50 |
| | 70 | 5′– 1″ | 8 1/2″ | 6′– 1″ | 2′–10″ * | 2′– 9″ | 8′–8″ | 11′– 7″ | ....... | 19300 | 50 |
| | 80 | 5′– 5″ | 8 1/2″ | 6′– 1″ | 2′–10″ * | 2′– 9″ | 8′–8″ | 12′– 3″ | ....... | 21000 | 50 |
| | 100 | 5′– 5″ | 8 3/4″ | 6′– 1″ | 2′–10″ * | 2′– 9″ | 8′–8″ | 14′– 3″ | ....... | 25500 | 50 |
| 10 | 40 | 5′–10″ | 8 1/2″ | 6′–11″ | 3′– 0″ * | 3′– 0″ | 8′–8″ | 10′– 5″ | ....... | 21300 | 50 |
| | 50 | 5′–10″ | 8 1/2″ | 6′–11″ | 3′– 0″ * | 3′– 0″ | 8′–8″ | 10′– 7″ | ....... | 22800 | 50 |
| | 60 | 5′–10″ | 8 3/4″ | 6′–11″ | 3′– 0″ * | 3′– 0″ | 8′–8″ | 10′– 9″ | ....... | 24000 | 50 |
| | 70 | 5′–10″ | 8 3/4″ | 6′–11″ | 3′– 0″ * | 3′– 0″ | 8′–8″ | 11′– 5″ | ....... | 26000 | 50 |
| | 80 | 6′– 0″ | 8 3/4″ | 6′–11″ | 3′– 0″ * | 3′– 0″ | 8′–8″ | 12′– 7″ | ....... | 27500 | 60 |
| | 100 | 6′– 0″ | 8 3/4″ | 6′–11″ | 3′– 0″ * | 3′– 0″ | 8′–8″ | 14′– 5″ | ....... | 32500 | 60 |
| 15 Aux., 3 | 40 | 6′– 0″ | 8 3/4″ | 7′–11″ | 2′– 6″ | 2′– 6″ | 8′–8″ | 11′– 5″ | 3′–0″ | 28500 | 60 |
| | 50 | 6′– 0″ | 8 3/4″ | 7′–11″ | 2′– 6″ | 2′– 6″ | 8′–8″ | 11′– 7″ | 3′–0″ | 30000 | 60 |
| | 60 | 6′– 0″ | 8 3/4″ | 7′–11″ | 2′– 6″ | 2′– 6″ | 8′–8″ | 11′– 9″ | 3′–0″ | 32000 | 60 |
| | 70 | 6′– 0″ | 8 3/4″ | 7′–11″ | 2′– 6″ | 2′– 6″ | 8′–8″ | 11′–11″ | 3′–0″ | 34000 | 60 |
| | 80 | 6′– 0″ | 8 3/4″ | 7′–11″ | 2′– 6″ | 2′– 6″ | 8′–8″ | 12′– 7″ | 3′–0″ | 36000 | 60 |
| | 100 | 6′– 6″ | 9 1/2″ | 7′–11″ | 2′– 6″ | 2′– 6″ | 8′–8″ | 14′– 5″ | 3′–0″ | 43500 | 70 |
| 20 Aux., 3 or 5 | 40 | 6′– 3″ | 9 1/2″ | 8′– 7″ | 3′– 0″ | 2′–10″ | 8′–8″ | 11′– 5″ | 2′–6″ | 34000 | 60 |
| | 50 | 6′– 3″ | 9 1/2″ | 8′– 7″ | 3′– 0″ | 2′–10″ | 8′–8″ | 11′– 7″ | 2′–6″ | 36500 | 60 |
| | 60 | 6′– 9″ | 9 1/2″ | 8′– 7″ | 3′– 0″ | 2′–10″ | 8′–8″ | 11′– 9″ | 2′–6″ | 39000 | 70 |
| | 70 | 6′– 9″ | 9 1/2″ | 8′– 7″ | 3′– 0″ | 2′–10″ | 8′–8″ | 12′– 0″ | 2′–6″ | 41500 | 70 |
| | 80 | 6′– 9″ | 9 1/2″ | 8′– 7″ | 3′– 0″ | 2′–10″ | 9′–8″ | 12′– 7″ | 2′–6″ | 44500 | 70 |
| | 100 | 6′–10″ | 10″ | 8′– 7″ | 3′– 0″ | 2′–10″ | 9′–8″ | 14′– 5″ | 2′–6″ | 52000 | 70 |
| 25 Aux., 5 | 40 | 7′– 1″ | 9 1/2″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 8′–8″ | 11′– 5″ | 3′–0″ | 43000 | 70 |
| | 50 | 7′– 1″ | 9 1/2″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 8′–8″ | 11′– 7″ | 3′–0″ | 45700 | 70 |
| | 60 | 7′– 1″ | 9 1/2″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 11′– 9″ | 3′–0″ | 48000 | 70 |
| | 70 | 7′– 3″ | 10″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′– 2″ | 3′–0″ | 52000 | 80 |
| | 80 | 7′– 3″ | 10″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′–10″ | 3′–0″ | 52500 | 80 |
| | 100 | 7′– 9″ | 11″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 14′– 8″ | 3′–0″ | 61000 | 80 |
| 30 Aux., 5 or 10 | 40 | 7′– 3″ | 10″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′– 4″ | 3′–0″ | 49000 | 80 |
| | 50 | 7′– 3″ | 10″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′– 6″ | 3′–0″ | 52000 | 80 |
| | 60 | 7′– 3″ | 10″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′– 6″ | 3′–0″ | 55000 | 80 |
| | 70 | 7′– 3″ | 10″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′– 8″ | 3′–0″ | 58000 | 80 |
| | 80 | 7′– 9″ | 11″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 12′–10″ | 3′–0″ | 62500 | 80 |
| | 100 | 7′– 9″ | 11″ | 9′– 9″ | 3′– 1″ | 3′– 4″ | 9′–8″ | 14′– 3″ | 3′–0″ | 70500 | 80 |
| 40 Aux., 5 or 10 | 40 | 7′– 6″ | 11″ | 10′–10″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 12′–10″ | 3′–5″ | 65000 | 80 |
| | 50 | 7′– 6″ | 11″ | 10′–10″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 13′– 0″ | 3′–5″ | 68300 | 80 |
| | 60 | 7′– 6″ | 11″ | 10′–10″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 13′– 2″ | 3′–5″ | 71500 | 80 |
| | 70 | 7′– 6″ | 11″ | 10′–10″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 13′– 4″ | 3′–5″ | 76000 | 80 |
| | 80 | 7′– 6″ | 13″ | 10′–10″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 14′– 0″ | 3′–5″ | 78500 | 100 |
| | 100 | 8′– 0″ | 13″ | 10′–10″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 14′– 4″ | 3′–5″ | 85000 | 100 |
| 50 Aux., 5 or 10 | 40 | 8′– 6″ | 13″ | 11′– 2″ | 3′– 1″ | 3′– 6″ | 9′–8″ | 13′– 7″ | 3′–5″ | 77500 | 100 |

* To main hook.

Note: Cranes of 50 tons capacity and above, and of 60 ft. span and over usually are equipped with 8 truck wheels. On 10-ton crane with 3-ton auxiliary hoist, dimensions *B* and *F* are 3 in. greater. Wheel loads specified do not include impact.

Advantages of gantry cranes are: No expense for overhead runways; tracks can be extended or changed in position; no obstruction of yards by overhead track supports.

**Portal Cranes** (double-leg) or **semi-portal** cranes, single-leg, are similar to gantry cranes. They generally have a rotating crane, or a crane with luffing motion mounted on top of the structure to transfer freight from ship to wharf, or *vice versa.*

SPECIAL CRANES FOR STEEL WORK.—**Ladle cranes** are made up to 300 ton capacity to control all the motions of filling, transporting and pouring hot-metal ladles in open-hearth plants. They have two independent trolleys with a pair of girders for each. The main trolley travels on the outer girders and hoists the ladle; the auxiliary trolley travels on the inner girder and has two hoists to tilt the ladle one way or the other. To insure safety, the two hoist motors and two sets of hoisting machinery are geared together to prevent the ladle from tipping sidewise and spilling metal if one motor or set of gears should fail.

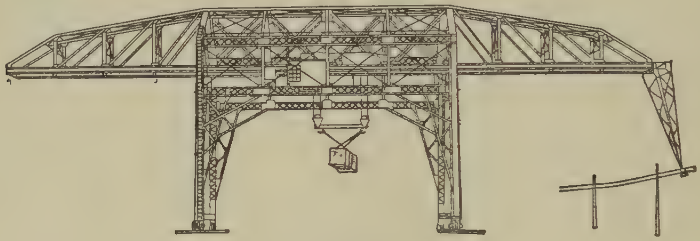


Fig. 13. 50-Ton Gantry Crane

**Ingot-stripping Cranes** are of several types. In one, a vertical arm suspended from a trolley carries jaws which seize the hooks of the ingot mold, while a screw-actuated plunger pushes the ingot from the mold. Some cranes carry auxiliary trolleys to handle stripped ingots. In rope-operated cranes, the hoist block carries tongs which lift the mold while a stationary ram pushes the ingot out.

**Soaking-pit Cranes** do all the work of handling ingots into and out of soaking pits. In one type, a trolley carries a guided vertical ram, gear-operated, with gripper jaws on the lower end. These have an automatic closing motion, suitable to ingots of different widths. The ram rotates on a vertical axis. This type requires high headroom. Rope-operated types require less headroom, have a shorter ram, and the guides are below the trolley (Alliance Machine Co., Alliance, O.).

**Crane-type Charging Machines** are made by Wellman Engg. Co., Cleveland. A vertical frame moves with the trolley and is braced against the under side of the bridge girders. It carries an arm that holds the charging box and turns it over when it has been pushed into the open-hearth furnace.

ELECTRICAL EQUIPMENT FOR OVERHEAD CRANES.—(From Crane Engineering by Whiting Corp., Harvey, Ill.) Alternating current generally is used for most cranes. More delicate control can be obtained than with direct current, especially for light loads. If both alternating and direct current are available, choose alternating; if only direct current, do not convert it to alternating, but use it.

**Motors.**—Slip-ring motors with wound rotors are preferable for crane service. Squirrel-cage induction motors may be used if delicate control is not required. Crane motors are rated for a given temperature rise in a stated operating time. For heavy service the Hp. rating is limited to a temperature rise of 40 to 50° C. in 60 to 90 min. operation. For intermediate or light service, a higher rating, based on 30 to 50 min., is permissible.

**Controllers.**—Drum-type controllers, with cast-grid resistance, are preferable for both direct and magnetic control. Arcing is relieved by blowout coils, and for protection, controller is completely enclosed. Controllers for operating from floor instead of from cab have a cable drum with spring return. Special controllers to give 90% reduction of full-load speed can be supplied for delicate control in foundry work.

**Switchboards.**—Main switch- and panel-boards are mounted in the rear of cab and totally enclosed. The switch may be locked in open position and it is impossible to open fuse box without opening main switch. Contactors for limit switches, magnets and magnetic control are accessibly located in the rear. Push buttons and master switches are in front near control levers. For heavy duty, cranes should have overload and low-voltage relays to protect all circuits. For regular service standard fuses generally are used.

**Crane Wiring** should conform to Underwriters' National Electric Code.

**Brakes.**—In alternating-current service, hoist motor has two brakes, one mechanical and one electrical, either of which will sustain full load. The principle of the mechanical brake is shown in Fig. 1, p. 23–37. The electric solenoid brake has a weighted plunger, which releases the brake shoes, and a wound coil to act against the weight and clamp the shoes to the drum on the motor shaft. Brake shoes are a composition of asbestos and graphite and are adjustable. Direct-current service uses dynamic braking. One or two solenoids are used to stop motor and hoist.

**Collectors and Conductors.**—Runway conductors are furnished by the crane purchaser. They should be spaced 8 to 10 in. vertically, and supported about every 20 ft. by insulated brackets. Wire conductors should be pulled tight by strain insulators at the ends.

**OPERATING SPEEDS OF CRANES.**—Table 2 shows motor sizes and operating speeds of shop cranes in intermittent service. For heavy-duty and 24-hr. service, heavier motors and higher speeds generally are used. High speeds of hoist and travel cannot be attained in short movements, because much of the time is used in accelerating and decelerating moving parts. See p. 23–32. In estimating capacity of a crane to hoist and transfer loads, allowance must be made for time thus lost. For short travel, as in transferring loads from car to adjacent storage, the allowance may be 25% or more.

**CRANE SUPPORTS.**—Running rails are furnished by the user; size is recommended by the maker for the particular service. They must be laid straight and at exact center to center distance. Track supports should be strong enough to carry maximum load close to one end of bridge, and stiff enough laterally to take the shock of sudden stops of the loaded crane trolley. Column connections and columns must be designed on a basis of maximum loads. It is well to consult crane builders before design of the building has gone too far.

**HOISTING ROPES** are cast-steel wire rope, usually 6 × 37; 6 × 19 and 8 × 19 also are used. Sheave and drum diameter should be (30 × rope diam.) See p. 6-34. For use of wire rope see p. 23-04.

## 2. POWER REQUIRED TO DRIVE CRANES

**POWER REQUIRED TO DRIVE CRANES.** (Adapted from Electric Crane Construction, by C. W. Hill, 1911.)—The current for hoisting is expended partly in lifting the load and partly in losses in the motor, the solenoid brake, the hoisting gearing and bearings and the friction and bending of the ropes on the drums and sheaves. As a rough average for shop cranes, these current expenditures may be 70% useful work and the losses 14%, 1%, 12%, and 3%, respectively, of the total current furnished. The size of the motor is determined by the torque required to lift the load plus the torque necessary to bring the load and all the rotating mechanism up to speed in the time allowed by the designer for the hoist to attain its full speed. The overload torque required for acceleration to a speed $v$ in $t$ seconds varies as $1/t$, and that factor, rather than the torque for direct lift, determines the size of the motor in all hoists except very slow ones. In a typical crane hoist, Fig. 14, the effect of the acceleration of all the moving parts on the torque of the motor can be obtained by calculating the weight and radius of gyration of the moving parts at the drum shaft and on all other rotating shafts, and then finding an equivalent weight which at the center of gyration of the motor armature, will produce the same energy effect. If $W$, $w$, $R$, $r$, $N$, $n$ are the weights of revolving parts, radii of gyration, and numbers of revolutions of two geared shafts respectively, then

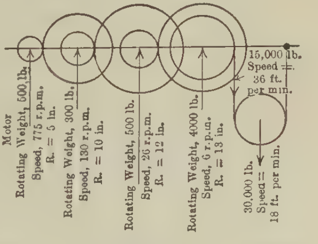


FIG. 14. Diagram of Crane Hoists Speeds

$$WR^2N^2 = wr^2 n^2, \text{ and } w = WR^2N^2/r^2 n^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

The force necessary to accelerate a weight $w$ to a velocity $v$ in time $t$ is $(w/g)(v/t)$. The duty required of the hoist motor in Fig. 14 can be calculated as follows:

1. Assumed load on hook, 30,000 lb. on four $^3/_4$-in. ropes; load on drum, 15,000 lb.
2. Drums, 23 in. diam., = 6 ft. circumference.
3. Speed of hoist rope, 36 ft. per min. = 0.6 ft. per sec.
4. Time to attain full speed, 5 sec.
5. Speed of motor, 775 r.p.m.; weight of revolving parts, 500 lb.; radius of gyration, 5 in.; velocity at center of gyration, 2029 ft. per min. = 33.8 ft. per sec.
6. Speed of first countershaft, 130 r.p.m.; weight of revolving parts, 300 lb.; radius of gyration, 10 in.; equivalent weight at center of gyration of armature is

$$300 \times 10^2 \times 130^2/5^2 \times 775^2 = 33.76, \text{ say } 33.8 \text{ lb.}$$

7. Speed of second shaft = 26 r.p.m.; weight of parts, 500 lb.; radius = 12 in.; equivalent weight at center of gyration of armature = $500 \times 12^2 \times 26^2/5^2 \times 775^2 = 3.2$ lb.

8. Speed of drum shaft 6 r.p.m.; weight, 4000 lb.; radius of gyration, 13 in. equivalent weight at center of gyration of armature $= 4000 \times 13^2 \times 6^2/5^2 \times 775^2 = 1.6$ lb.

9. Speed of weight and rope, 36 ft. per min.; equivalent weight at armature is

$$15{,}000 \times 36^2/2029^2 = 4.7 \text{ lb.}$$

10. Total equivalent weight at center of gyration of armature $= 500 + 43.3 = 543.3$ lb., say 543 lb.

11. Force at armature to lift weight $= 15{,}000 \times (36/2029) = 266$ lb. Add $^1/_6$ of this $= 44$, approximately, for losses in gearing and ropes; $266 + 44 = 310$ lb.

12. Force at armature to accelerate all moving parts in 5 sec. is

$$(w/g)(v/t) = (543/32.2) \times (33.8/5) = 114 \text{ lb.} + {}^1/_6 \text{ for losses} = 133 \text{ lb.}$$

13. Ratio of (12) to (11) $= 133 \div 310 = 0.429$, *i.e.*, overload during period of acceleration is 43% for 5 seconds.

14. Horsepower for lift only $= 310 \times (2029/33{,}000) = 19.06$ Hp.

15. Horsepower for lift and acceleration is $(310 + 133)(2029/33{,}000) = 27.23$ Hp.

In a similar way, the power required to travel the bridge may be investigated.

16. Dead weight of crane assumed 75,000 lb.; load 30,000 lb.; total 105,000 lb.

17. Assumed coefficient of rolling friction, 30 lb. per ton; tractive effort at truck wheels $= (105{,}000/2000) \times 30 = 1575$ lb.

18. Speed of travel, 300 ft. per min. = 5 ft. per sec.

19. Time to attain full speed of travel, 15 sec.

20. Speed of armature, 775 r.p.m.; weight of revolving parts, 500 lb.; radius of gyration, 5 in.; velocity at center of gyration = 2029 ft. per min. = 33.8 ft. per sec.

21. Equivalent weight at center of gyration of armature for revolving parts [like (6), (7), and (8)], assumed 100 lb.

22. Equivalent weight at center of gyration of armature for traveling load is

$$105{,}000 \times 5^2/33.8^2 = 2297 \text{ lb.}$$

23. Total equivalent weight at armature $= 500 + 100 + 2297 = 2897$ lb.

24. Force at armature to travel crane at 30 lb. per ton $= 1575 \times (5/33.8) = 233$ lb. $+ {}^1/_6$ for losses $= 272$ lb.

25. Force at armature to accelerate all moving parts in 15 sec. is

$$(w/g)(v/t) = 2897 \times 33.8/32.2 \times 15 = 202 \text{ lb.} + {}^1/_6 \text{ for losses} = 236 \text{ lb.}$$

26. Ratio of (25) to (24) $= 236 \div 272 = 0.87$, *i.e.*, overload during period of acceleration is 87% for 15 sec.

27. Horsepower for travel only $= 272 \times 2029/33{,}000 = 16.7$ Hp.

28. Horsepower for travel and acceleration $(272 + 236)2029/33{,}000 = 31.2$ Hp.

From these figures we see that while a 20-Hp. motor might do for the hoist, which approximately figures 19 Hp., and not be overtaxed by the overload of 50% for 5 sec., the travel, which figures 16.7 Hp., requires a larger excess of torque because the overload of 88% is exerted for 15 sec. The full power of the motor during that time amounts to 31.2 Hp., so that a 25-Hp. motor is advisable. It will at the same time take care of irregularities in the track and poor handling of the controller. From (25), it is apparent that if the time of acceleration were reduced to 7 $^1/_2$ sec. the horsepower for travel and acceleration would be over 45 Hp. Characteristic curves of motors published by the makers show the permissible relations of torque and time, and should be followed in the selection of sizes.

**PRACTICAL RULES.**—Builders of traveling electric cranes used in shops assume a hoisting efficiency of about 70%, which corresponds to 23,000 ft.-lb. per min. per Hp. of motor rating. For cranes used in steel mills, grab-bucket hoisting, etc., subjected to occasional overload and rough handling 20,000 ft.-lb. per min. per Hp. is used. The margin of 30% or 40% generally is considered sufficient to allow for acceleration and for electrical and mechanical losses, but for high hoisting speeds and quick acceleration, the forces due to acceleration should be calculated and allowance made for them.

For travel speeds not over 250 ft. per min. a rule used in practice is

$$\text{Motor Hp.} = 2\left[\left(W \times \frac{d}{D} \times 0.07 \times 1.07^n \times S\right) \div 33{,}000\right] \quad . \quad . \quad . \, [2]$$

where $W$ = weight of loaded crane, lb.; $d$ = diameter of truck axles, in.; $D$ = diameter of truck wheels, in.; 0.07 = coefficient of axle friction, and also is the loss at each cut gear reduction, including journal friction of countershaft; $n$ = number of gear reductions; $S$ = speed, ft. per min. The factor 2 covers overload due to acceleration, bad condition of crane track, "notching-up" controller too rapidly, etc. For greater safety against these adverse conditions, the factor sometimes is made 2 $^1/_2$. For speeds over 250 ft.

per min. and for quick acceleration the forces due to acceleration should be calculated and allowed for.

Crane Engineering, by Whiting Corp., Harvey, Ill., gives the following rules: Hp. of hoist motor = (Load in tons × Lifting speed, ft. per min.)/11. This is equivalent to assuming an overall efficiency of 66%. Hp. of trolley motor or bridge motor = $(WS/132)\ (d/D)$, if bearings are bronze bushed; or $SW\{30 + (S/20)\}/33{,}000$, if axles run in roller bearings. In these equations, $W$ = total weight moved, tons; $S$ = travel speed, ft. per min.; $d$ = diam. of axle, in.; $D$ = diam. of wheel, in.

## 3. LOADING AND UNLOADING EQUIPMENT

**ORE BRIDGES AND COAL STORAGE BRIDGES** are gantry cranes with a grab bucket to take ore or coal from boats, cars or a dump to storage, and to reclaim it from storage to cars or boats. As used at Great Lakes ports, these often are bridges spanning 300 ft. or more over the storage area, with cantilever extensions over the water at one end and over railroad tracks at the other. In modern bridges (1936) up to 10-ton buckets have all drums for control of bucket mounted, together with operator's cab, on the trolley. Man-trolley plants, fast in operation and of high capacity, are built (see p. 23–81) by manufacturers [41], [50], [77], [90]. For illustrations and descriptions of various types, see *Trans.* A.S.M.E., 1929, IS–51–4. For methods of operation and description of safety devices, see *Trans.* A.S.M.E., 1930, IS–52–4; also *Demag Nachrichten*, vol. 7B, No. 1, Feb., 1933.

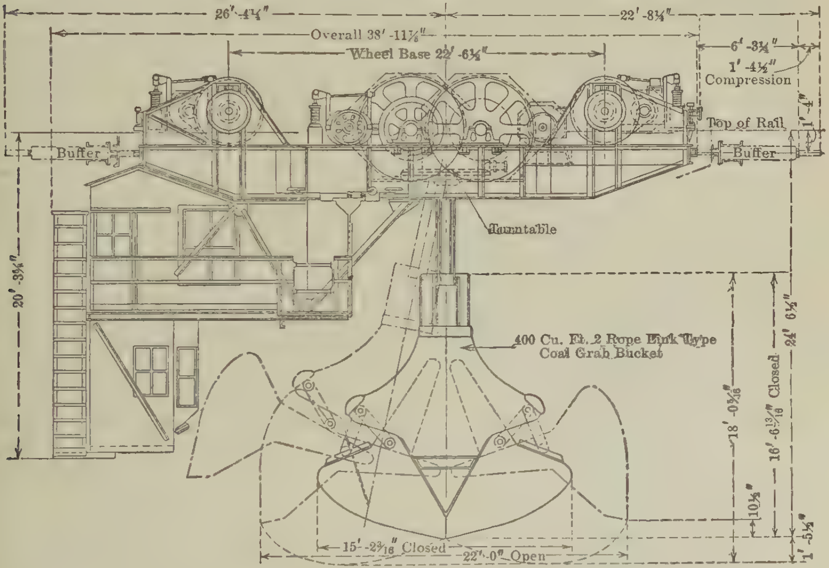


Fig. 15. 10-Ton Man Trolley

In some storage bridges a revolving crane, with a grab bucket on a horizontal boom, runs on the top chords of the bridge. Coal from boats is delivered to a belt conveyor on the bridge and tripped off into storage. In reclaiming from storage, the crane boom swings sideways from the bridge and the grab bucket drops the coal into a hopper delivering to the belt conveyor. Coal can be reclaimed at the rate of 300 tons per hr. (Robins Conveying Belt Co., New York.)

Fig. 15 shows a 10-ton man-trolley. The specifications as given by the makers, Industrial-Brownhoist Corp., are: Maximum moving load, 158,000 lb.; load on ropes, 51,000 lb.; weight of trolley, 107,000 lb.; hoisting speed, 200 ft. per min.; max. height of hoist, 80 ft.; trolley travel speed, 800–1000 ft. per min.; bridge travel speed, 75–100 ft. per min.; bridge moved from trolley cab at any point on the bridge.

**BOAT LOADING.**—At the upper end of the Great Lakes, over 50 million tons of iron ore are loaded into steamers each year. The ore is dug by steam shovel and loaded

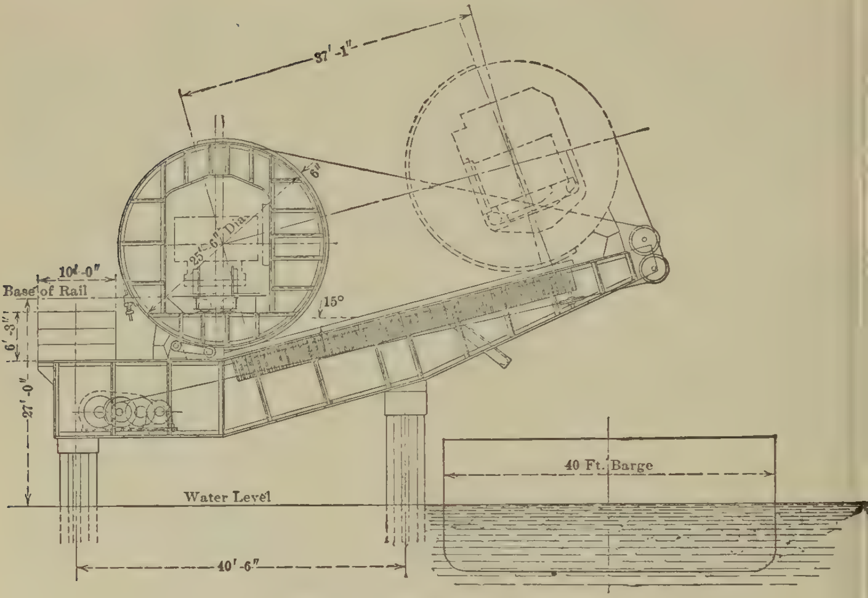


FIG. 16. Link-Belt Car Dumper

into large bottom-dump cars, which at the shipping pier run on high trestles. When the bottom doors open, the ore drops out and runs freely through long chutes into the boat.

On the Atlantic coast, coal is shipped from Norfolk, Baltimore, Philadelphia and other ports the year round. In winter, it sometimes is necessary to thaw the railroad cars in a closed thawing shed before dumping. At some piers the cars run into a rotary dumper, are turned over, and the coal runs into transfer cars. The transfer cars may be lifted to the top deck of the pier to dump their coal into hoppers and ship-loading chutes, or they may travel along the pier and discharge to a conveyor on a boom extending over the boat. In a plant at Baltimore the turn-over motion of the Wellman rotary dump is so controlled that the coal slides out, without much breakage, to an apron conveyor which in turn discharges to the conveyor on the boom. At the end of the boom a vertical chute with a control gate, to ease the fall, delivers coal to the boat. Boats loading coal at Atlantic ports are of all sorts; hatches generally are small, and chutes must be arranged to suit them and to clear ship's rigging. At a coal pier at Pensacola, Fla., a Link-Belt car dumper feeds two bucket elevators, of 300 tons per hr. capacity each. They can discharge at heights variable by 45 ft. to suit a great variety of boats and conditions of loading. Loading chutes have telescopic ends and a horizontal swing of 180°.

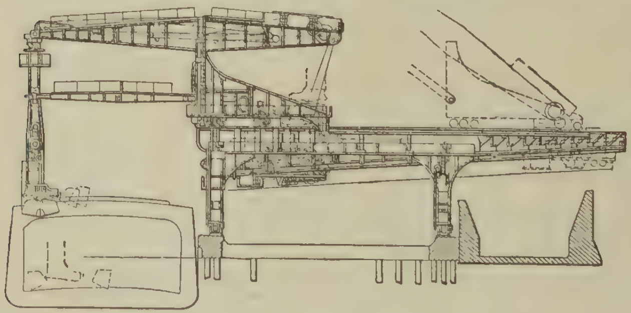


FIG. 17. Hulett Unloader

The large boats bringing ore from the head of the Great Lakes and taking coal back are designed for the purpose. At coal shipping ports, mostly in Ohio, boats are loaded by coal dumpers, generally of the lift type. Fig. 16 shows a Link-Belt rolling dumper which pours the contents of a 40 ft. car into a barge. When car is empty the cradle runs down an inclined track. For history and description of car dumpers, see A. F. Case, *Trans.* A.S.M.E., 1928, MH–50–6. Rotary dumpers and rolling dumpers are made (see p. 23–81) by manufacturers [50] and [90]. The latter also makes lift type car dumpers.

**BOAT UNLOADING.**—Clam-shell buckets are in general use for unloading boats. For intermittent service and small capacities, a mast and gaff with a clam-shell bucket and 2-drum hoist is good and cheap. An inclined boom (gaff) swings around the mast, the bucket being hung from the top end of the gaff. Guide sheaves for holding and closing lines (see p. 23–21) are spread apart at the heel of the gaff, so that the swing of the gaff toward and from a belt hopper can be controlled by putting more tension on one line than on another by means of clutches and brakes at the hoisting machine. Such hoists are used for boats and barges having no rigging to interfere with the swing of the gaff; for rigged boats, a tower with a boom hinged to swing vertically is necessary.

Hoisting towers, with separate hoists to lift the bucket and trolley it on the boom, permit rapid work, as the two motions are simultaneous part of the time. Such towers are made (see p. 23–81) by manufacturers [50], [52], [56] and others. They often work in combination with cable railways or conveying systems to handle coal at storage pockets and power houses on water fronts. To transfer coal or ore from boat to cars or to storage piles, the hinged boom may be a cantilever extension of a bridge tramway.

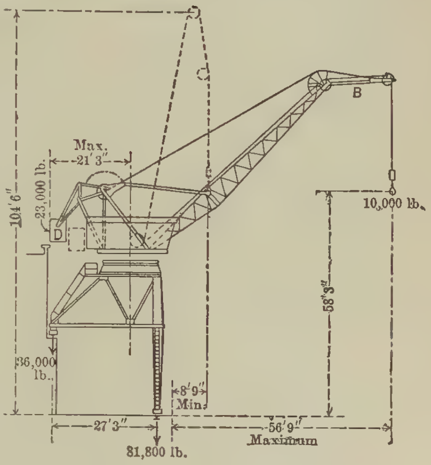


Fig. 18. Level-luffing Type of Crane

The Hulett unloader, Fig. 17 (Wellman Engg. Co.) comprises a gantry frame spanning a number of railroad tracks and carrying a trolley. To the trolley is pivoted a pair of arms carrying a vertical bucket leg. The grab-bucket is fastened to this leg. An operator in the leg controls all motions of bucket, trolley and gantry. A second operator on a traveling scale-larry, hung under the gantry, receives the discharge of the bucket through a hopper holding 3 bucket-loads. After weighing, the larry discharges to railroad cars or to the reserve storage space under the cantilever end of the gantry. Large Hulett buckets spread open to 24 ft., pick up 17 tons of iron ore, and make the cycle of load, traverse and dump in about 50 sec.

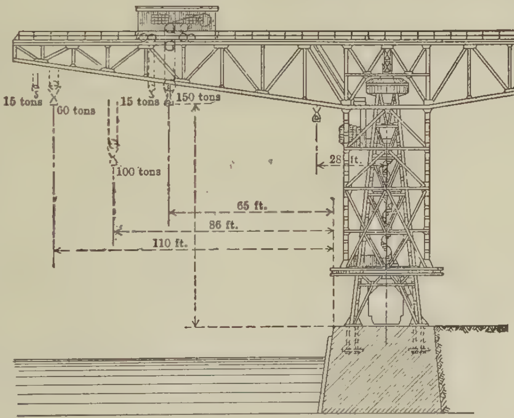


Fig. 19. 150-Ton Hammerhead Crane

**FREIGHT HANDLING TO AND FROM SHIP.**—In the U. S. ships generally are loaded and unloaded by the ship's own winches and rigging. In Europe most of the work is done by wharf cranes, as much of the freight comes and goes in barges without hoisting machinery. To handle freight to and from ships faster and cheaper than by use of the ship's winches, cargo cranes on the wharf must be kept to that one duty and not be burdened with distribution on the wharf. Monorail hoists and tramrail carriers have been used on a few wharves, but electric or gasoline tractors with trains of trailers are more flexible and useful.

Most wharf cranes are traveling gantries with a revolving crane on top of the bridge. Crane booms often are rigged with a " level-luffing " device. Fig. 18 shows one type. Outrigger *B* and counterweight *D* are so mounted that with hoist drum braked, boom can be moved in a vertical plane without raising or lowering the load. The advantages are: Several cranes can work in one hatch without interfering with each other or with the ship's rigging; time is saved; the boom hoist motor may not be so large. See Bernard Dunell, Cargo Cranes, *Trans.* A.S.M.E., 1928, MH–50–5. For methods of weighing the load on the crane according to the tension in the hoisting line, see *Fördertechnik*, Jan. 13, 1933.

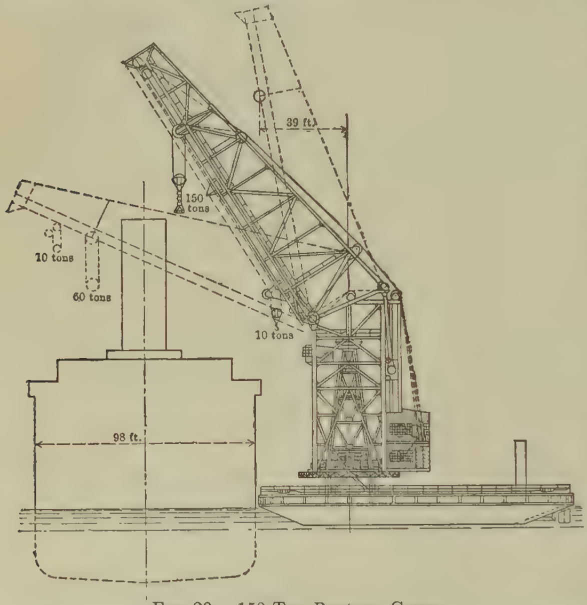


Fig. 20. 150-Ton Pontoon Crane

Cargo cranes generally have a lifting capacity of 10 tons or less, but important European harbors always have some wharves with large cranes to handle heavier loads. Fig. 19 shows a hammer-head crane that will lift 150 tons from a large ship at a distance of 65 ft. from the wharf front, and to 80 or 90 ft. above the wharf. Heavy lifting also is done by pontoon cranes; they often work from the harbor side of the ship while the wharf cranes work on the other side. Fig. 20 shows a 150-ton crane of this kind. It has a Diesel engine power plant for hoists, lighting, and driving the screws which propel the pontoon at a speed of 4 knots. See articles in *Demag Nachrichten*, Feb., 1932, and *Fördertechnik*, Feb. 12, 1932.

# HOISTS

## 1. HAND-OPERATED HOISTS

**HAND POWER HOISTS** (or Chain Blocks) are made in three types: 1. Differential hoists contain two chain wheels above, cast as one piece, and a single wheel below. An endless chain passes over one upper sheave, down around the bottom wheel that carries the load hook, then up over the other wheel. For theory of this device, see p. 8–05. 2. Screw-geared hoists have a worm gear that holds the suspended load by friction of the thread of the worm. 3. Spur-geared hoists have a train of spur-gears with a friction-disc brake of the Weston type to hold the load. See Fig. 1. Type 3, when fitted with ball-bearings, will deliver 85% of the work of the operator to the load-hook. Its mechanical efficiency is about twice that of a screw-geared hoist and nearly three times that of a differential hoist. See Table 1.

Table 2 shows hoisting speeds and pulls on hand-chain for the three types of hoist as stated by one maker. Other makers list the same sizes; their dimensions and ratings are nearly the same. The speed of a chain hoist depends on the pull required on the hand-chain and number of feet of hand-chain overhauled to lift load 1 ft. Speeds given in Table 2 are for short lifts with men accustomed to the work. For easy continuous lifting, use $^2/_3$ of the values given for hoisting speeds. The spur-geared hoist has less internal friction, lifts and lowers load more rapidly than the others. The screw-geared hoist is lighter, takes up less headroom, will work in any position (as for horizontal pulling), and is well suited to work of riggers and repair gangs. The differential hoist is still lighter and serves well for occasional use.

Similar hoists are made (see p. 23–81) by manufacturers [18], [30], [76], [93]. Fig. 1 shows the Cyclone hoist made by Chisholm-Moore. Gearing is compact and entirely enclosed in a grease-filled case. Brake-wheel *A* is keyed to the shaft that carries hand-chain wheel *B*, whose bore is threaded to fit a thread on hub of *A*. The pull of the load-chain on wheel *C* acts through the gears to move *B* to the left, squeezing the leather and steel friction rings against *A*. A pawl engages ratchet teeth on the rim of *A*, prevents the latter from turning and holds load suspended while the friction surfaces are squeezed together. To lift load, *B* is pulled in direction in which ratchet is free, the friction surfaces being engaged. To lower, load *B* is pulled in the opposite direction; it moves on threaded hub of *A* just enough to overcome friction imposed by the load; the pawl engages the ratchet and holds brake wheel when the motion of hand-chain stops. This description applies generally to all screw-type brakes (known also as Weston brakes) used on hand-driven or power-driven hoists.

The Cyclone hoist, in sizes up to 2 tons capacity, also is made with aluminum alloy parts. These hoists are 1/3 lighter than regular hoists, and can be put up and taken down more easily. A safety device throws the hand-chain out of gear if the hoists are subjected to an overload more than the 50% excess over normal rating to which they are tested by the maker. Constants for speed and pull are about the same as those given in Table 2.

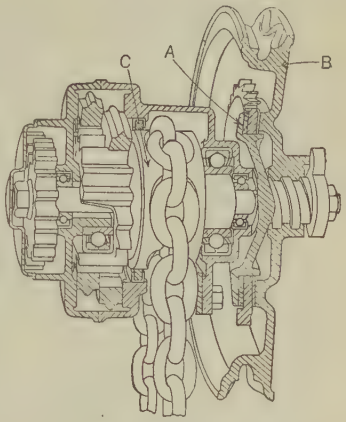


FIG. 1. Spur-geared Hoist with Friction Brake

If it is not convenient for the operator to stand under the hoist, a fixed winch operated

**Table 1.—Comparison of Chain Block Hoists**

| | Differential | Screw-geared | Spur-geared |
|---|---|---|---|
| Mechanical efficiency | 35% | 40% | 85% |
| Relative durability | 20% | 80% | 100% |
| Relative cost | 40% | 70% | 100% |

**Table 2.—Hand-power Chain Hoists**

(Yale & Towne Mfg. Co., Philadelphia)

| Capacity, tons (2000 lb.) | Pull on Hand Chain to Lift Full Loads, lb. | | | Feet of Chain Overhauled to Lift Load 1 ft. | | | Hoisting Speeds Attainable, ft. per min.; Number of Men Needed to Lift Stated Loads, without Pulling over 76 lb. on Hand Chain | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Spur-geared | | | | Screw-geared | | Differential | |
| | Spur-geared | Screw-geared | Differential | Spur-geared | Screw-geared | Differential | Full Load | Half Load | Quarter Load | No. of Men | Full Load | No. of Men | Full Load | No. of Men |
| 1/4 | 47 | ... | 72 | 12 1/2 | ... | 18 | 17.0 | 30.0 | 45.0 | 1 | ... | ... | 6.0 | 1 |
| 1/2 | 57 | 68 | 110 | 21 | 40 | 24 | 8.5 | 17.0 | 26.0 | 1 | 4.0 | 1 | 6.0 | 2 |
| 1 | 76 | 87 | 190 | 31 | 59 | 30 | 4.2 | 8.5 | 13.0 | 1 | 2.0 | 1 | 3.7 | 3 |
| 1 1/2 | 103 | 94 | 225 | 35 | 80 | 36 | 5.1 | 10.2 | 15.3 | 2 | 2.4 | 2 | 2.5 | 3 |
| 2 | 112 | 115 | 300 | 42 | 93 | 42 | 3.9 | 7.8 | 11.7 | 2 | 1.8 | 2 | 2.3 | 4 |
| 3 | 107 | 132 | ... | 70 | 126 | .. | 2.5 | 5.0 | 7.5 | 2 | 1.1 | 2 | ... | ... |
| 4 | 116 | 142 | ... | 84 | 155 | .. | 1.9 | 3.8 | 5.7 | 2 | 0.8 | 2 | ... | .. |
| 5 | 102 | 145 | ... | 126 | 195 | .. | 1.4 | 2.8 | 4.2 | 2 | 0.65 | 2 | ... | .. |
| 6 | 122 | ... | ... | 126 | ... | .. | 1.2 | 2.4 | 3.6 | 2 | ... | .. | ... | .. |
| 8 | 127 | ... | ... | 168 | ... | .. | 0.85 | 1.7 | 2.6 | 2 | ... | .. | ... | .. |
| 10 | 132 | ... | ... | 210 | ... | .. | 0.65 | 1.3 | 2.0 | 2 | ... | .. | ... | .. |
| 12 | 122* | ... | ... | 126* | ... | .. | 1.2 | 2.4 | 3.6 | 4 | ... | .. | ... | .. |
| 16 | 127* | ... | ... | 168* | ... | .. | 0.85 | 1.7 | 2.6 | 4 | ... | .. | ... | .. |
| 20 | 132* | ... | ... | 210* | ... | .. | .65 | 1.3 | 2.0 | 4 | ... | .. | ... | .. |
| 25 | 165* | ... | ... | 210* | ... | .. | .50 | 1.0 | 1.5 | 4 | ... | .. | ... | .. |

* On each of the two hand chains.

by a hand-wheel or by a hand-chain may be used. See Fig. 2. The mechanism is that of a chain hoist enclosed in a box that can be bolted to a wall, building-post or mast of a jib crane. These hoists, up to 2 tons capacity, are listed by several makers.

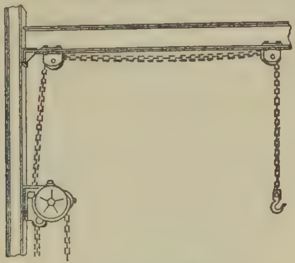


Fig. 2. Spur-geared Hand Winch

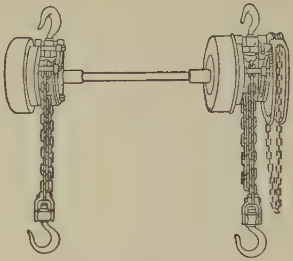


Fig. 3. Twin Hook Chain Block

Hoist makers furnish double hoists, Fig. 3, for lifting lumber, tanks, platforms, truck bodies, etc. They are made in sizes up to 10 tons, and with hoists spaced to suit dimensions of pieces handled. 4- or 8-wheel carriers with double hoists hooked on, or built in. are listed by some builders of tramrail systems.

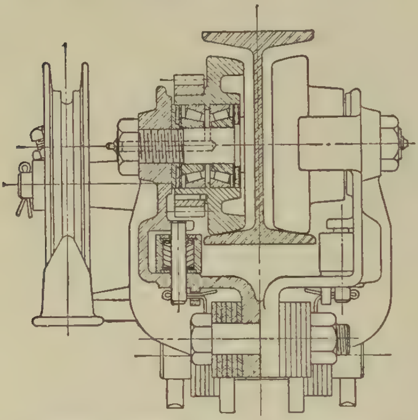


Fig. 4. Geared Trolley for Hand Hoist

The usefulness of a hand hoist can be increased if it is hung from a trolley on an overhead track. If the track forms the bridge of a traveling crane or is attached to a jib-crane, a greater area can be covered. Table 3 gives principal dimensions of trolleys for I-beam tracks. These trolleys are moved by pulling or pushing the suspended load, or are geared and propelled by a hand-chain. The Ace geared trolley (Robbins & Myers) is shown in Fig. 4. It has cast-steel side frames with adjustable spacing for track beams of different widths. The load wheels have no flanges; guiding is done by small rollers which bear against lower flanges of the beam. Similar geared trolleys are furnished by other makers of hoists. Trolleys with 4 or 8 ball-bearing wheels, not geared, are made to take standard makes of chain hoists up to 2 tons capacity, to run on American monorail or Cleveland tramrail tracks. See Fig. 5.

The so-called Army type of chain hoist is a combination of a trolley, either geared or plain, with the hoist mechanism built in. The advantage is that the load-hook can come

**Table 3.—Roller-bearing Steel-plate Trolleys**
(Ford Chain Block Co., Philadelphia)

| Capacity of Hoist, tons | Standard I-Beam * | | Least Radius Curvature of I-Beam, in. | Greatest Span of I-Beam, between Supports, ft. | Size of Wheel, in. | Overall Width, in. | Bottom of Beam to Eye in Clevis, in. | Weight of Trolley, lb. |
|---|---|---|---|---|---|---|---|---|
| | in. | lb. | | | | | | |
| 1/4 | 4 | 7.7 | 18 | 10 | 3 1/8 | 6 1/2 | 6 | 25 |
| 1/2 | 5 | 10.0 | 21 | 14 | 3 3/4 | 6 1/2 | 5 1/4 | 30 |
| 1 | 6 | 12.5 | 21 | 14 | 4 5/8 | 7 1/4 | 6 1/2 | 50 |
| 1 1/2 | 7 | 15.3 | 30 | 15 | 5 3/8 | 9 1/2 | 7 | 95 |
| 2 | 8 | 18.4 | 36 | 16 | 6 3/8 | 9 3/4 | 7 1/4 | 115 |
| 3 | 9 | 21.8 | 42 | 16 | 7 3/8 | 11 | 7 1/4 | 150 |
| 4 | 10 | 25.4 | 48 | 16 | 8 1/4 | 13 1/4 | 8 1/4 | 210 |
| 5 | 12 | 31.8 | 54 | 18 | 10 1/8 | 13 1/2 | 9 1/2 | 270 |
| 6 | 12 | 31.8 | 60 | 16 | 10 1/8 | 13 1/2 | 9 1/2 | 350 |
| 8 | 15 | 42.9 | 60 | 18 | 12 | 17 | 12 | 420 |
| 10 | 15 | 42.9 | 60 | 15 | 12 | 17 | 13 | 550 |

* By using spreaders, frames can be widened to suit beams other than standard.

closer to the bottom of the I-beam track. The disadvantage is that the hoist does not work so well if the pull of the load is much off to the side of the track. Fig. 6 shows a hoist of this type, the "Rail Hugger" (Yale & Towne). The hoist is spur-geared with ball bearings; the trolley frame is steel plate; the chilled wheels have roller bearings. Table 4 gives principal dimensions.

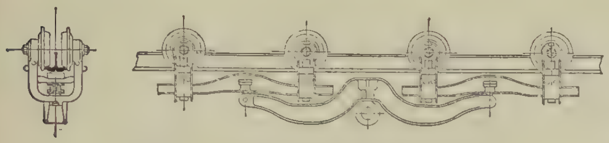


FIG. 5. Trolley to Fit Tramrail Track

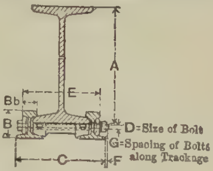


FIG. 7. Double Rail Hoist Track

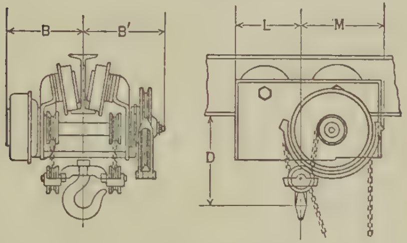


FIG. 6. Army-type Chain Hoist

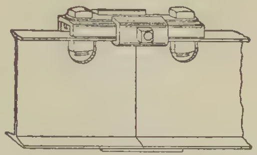


FIG. 8. I-Beam Track Joint

**TRACKS FOR HOISTS.**—Standard I-beams are in general use to carry trolleys of hoists, especially for heavy loads. Common sizes and greatest spans for stated loads are given in Table 3. The beams should be straight; where they join, the bottom flanges should be supported by a plate, otherwise the corners may be bent down by the trolley wheels. It is not usual to splice webs. Supports or hangers are fastened to top flange in various ways to leave room for trolleys. Under frequent passage of heavy loads, lower flanges tend to bend down from peening action of the wheels. This can be prevented by riveting strips to lower flanges to act as a track. The double-rail construction in Fig. 7 serves the same purpose. The I-beam track made by Richards-Wilcox Co., Aurora, Ill., consists of straightened beams; Fig. 8 shows a joint and Fig. 9 an I-beam switch. Pulling the hand chain releases a latch pin, then draws the movable beam over to the next position. A guard plate keeps the trolley from running off the open end of tracks not in register. Similar track work is made by Philadelphia Tramrail Co., Philadelphia, and some builders of hoists. See also Tramrail Systems.

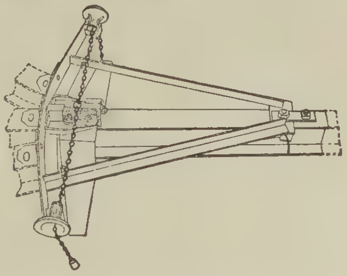


FIG. 9. I–Beam Switch

**Table 4.—Dimensions of Army Type Chain Hoist** (see Fig. 6)
(Yale & Towne Mfg. Co., Philadelphia, Pa.)

| Rated Capacity, tons (2240 lb.) | Standard Size of I-Beam | | Min. Radius of Curvature. | Dimensions, in. | | | | | Standard Lift, ft. | Chain Pull to Lift Full Load, lb. | Chain Overhauled to Lift Load 1 ft., ft. | Speed, ft. per min. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | in. | lb. | | *B* | *B′* | *D* | *L* | *M* | | | | |
| 1 1/2 | 7 | 15.3 | 2′–6″ | 10 1/2 | 11 | 6 3/4 | 8 1/8 | 11 1/4 | 8 | 69 | 56 | 5.1 |
| 2 | 8 | 18.4 | 3′–0″ | 10 1/2 | 11 | 7 | 8 7/8 | 11 1/8 | 9 | 92 | 56 | 3.9 |
| 3 | 9 | 21.8 | 3′–6″ | 12 | 12 1/2 | 7 3/4 | 10 5/8 | 12 1/4 | 10 | 115 | 68 | 2.5 |
| 4 | 10 | 25.4 | 4′–0″ | 14 1/2 | 14 1/4 | 9 1/2 | 11 1/2 | 13 3/8 | 10 | 121 | 86 | 1.9 |
| 5 | 12 | 31.8 | 4′–6″ | 16 1/4 | 15 | 10 3/8 | 13 5/8 | 14 5/8 | 12 | 92 | 161 | 1.4 |
| 6 | 12 | 31.8 | 4′–6″ | 16 1/4 | 15 | 10 3/8 | 13 5/8 | 14 5/8 | 12 | 110 | 161 | 1.2 |
| 8 | 15 | 42.9 | 5′–0″ | 18 7/8 | 18 3/4 | 12 | 14 5/8 | 16 5/8 | 12 | 135 | 168 | 0.85 |
| 10 | 15 | 42.9 | 5′–0″ | 20 3/4 | 19 1/4 | 13 | 15 3/4 | 17 1/4 | 12 | 137 | 242 | .65 |

**PORTABLE HOISTS** of the type shown in Fig. 10 for loads up to 3 tons are made (see p. 23–81) by manufacturers [25], [41], [48], and some hoist builders. They serve where it is not convenient to hang up a chain hoist. Lifting is by a winch built in the frame, or a chain hoist hung from top of the column.

**SPUR-GEAR HOISTS** in capacities up to 1 ton, operated by hand rope or hand wheel, and fitted with automatic stops and safety devices are made by Sedgwick Machine Co., New York, and others. They are fitted with platforms or containers for hoisting or lowering from floor to floor or from sidewalk to basement. Hoists especially designed to lift or lower from basement to sidewalk are also made by manufacturers [25], [32], [48] (see p. 23–81), and others. These machines are also made with electric motor drive.

Small hand-operated counterbalanced hoists for carrying parcels, tools and books rapidly from floor to floor are made by manufacturers [46], [83] (see p. 23–81), and builders of elevators. In some cases, installation of these machines is subject to State Elevator Codes and local regulations in regard to safety and fire.

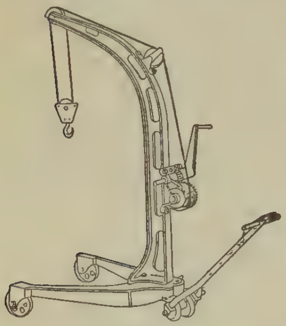


Fig. 10. Portable Hoist

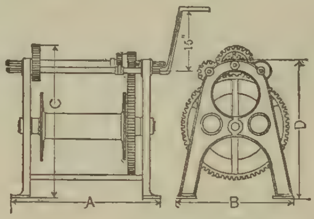


Fig. 11. Portable Hand Winch

**CRABS OR HAND-WINCHES** are made as self-contained units for use with derricks, with block and tackle for riggers, or for lifting through heights beyond the range of chain-blocks. In the portable form, Fig. 11, they consist of two frames carrying a drum with, usually, two sets of gears on shafts with squared ends for the cranks. Sometimes drum-shafts also are squared, to allow slack rope to be taken up quickly by using cranks directly on the drum. One pinion is made to shift in and out of gear, so that cranks can be used on either pinion shaft for different hoist speeds. The load is held and controlled for lowering either by a ratchet and pawl, by a strap-brake or by a friction disc brake of the Weston type. The friction brake will prevent the cranks from accidentally flying back and injuring workmen, as often happens with crabs equipped only with ratchet and pawl. The principle of the Weston brake is shown in Fig. 1. Table 5 gives sizes and dimensions of the style of winch shown in Fig. 11. Fig. 12 shows the same mechanism fitted to flat frames for use on masts of jib cranes and derricks. Simple worm-geared winches of a capacity of 1 ton or less are made in the form shown in Fig. 13 by nearly all makers of conveying machinery.

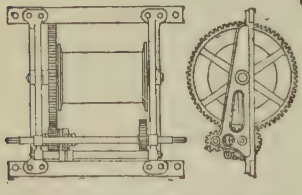


Fig. 12. Hand Winch for Jib-crane or Derrick

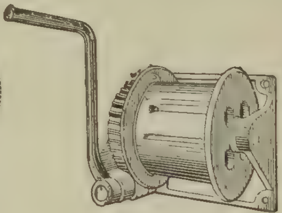


Fig. 13. Worm-geared Winch

**Table 5.—Dimensions of Hand-power Crabs or Winches**

(Industrial-Brownhoist Corp., Bay City, Mich.)

| Pull on Single Line, tons | Barrel Diameter × Length, in. | Dimensions, in. (See Fig. 11) | | | | Weight, lb. |
|---|---|---|---|---|---|---|
| | | *A* | *B* | *C* | *D* | |
| 1/2 | 4 1/2 × 12 | 23 | 22 | 31 | 30 | 230 |
| 1 | 5 × 16 | 30 | 24 | 35 | 32 | 370 |
| 1 1/2 | 6 × 20 | 35 | 27 | 36 | 33 | 520 |
| 2 | 7 × 24 | 41 | 30 | 41 | 35 | 810 |

Throw of cranks 15 in. Larger drums than those listed can be made.

## 2. ELECTRIC HOISTS

**ELECTRIC HOISTS** generally use ball and roller bearings, alloy steels, and heat-treated parts. Pressed steel has been substituted for castings in some designs. Welding has replaced bolting in many cases. Motors usually are enclosed, ball-bearing, and specially designed for hoist work; high-torque squirrel-cage induction motors are more general than direct-current motors. The latter are useful for close control of load movement. Both rope and push-button control are used. Push-buttons give quicker, safer and more convenient control. Electric hoists range in size from $^1/_8$ to 6 tons, with a wide range of hoist speeds. They are made in floor-operated and cab-operated types. For single-speed hoisting, push-button control is the most convenient; one hand is free to steady or guide the load, operation is faster, and the load can be placed more accurately. Cab-controlled hoists are used to transport loads over a considerable distance, where the operator must observe the load from above, or where rapid transfer is necessary.

Worm-driven electric hoists hold the load by friction on the worm thread. Spur-geared hoists use a screw brake (Weston) to hold the load. See Fig. 1. All have a limit switch, to prevent over-travel of hoist hook at top of the lift, and a motor brake to stop the rotor if current fails, and to prevent drifting when controller is in "off" position.

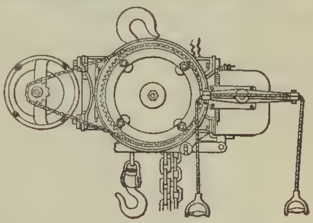


FIG. 14. Electric Chain Hoist

Electric chain hoists are made by manufacturers [18], [95] (see p. 23–81). In Fig. 14 (Yale & Towne) a chain drive from the motor operates gearing like that in a chain hoist. The load is lifted by a chain driven by a pocket-wheel; slack chain is paid out as load ascends. For high lift and for pulling at loads not directly underneath, a chain hoist often is preferable to a rope hoist. Chain hoists are made in sizes of $^1/_8$ to 2 tons.

**SPUR-GEARED HOISTS.**—Fig. 15 shows a hoist built in sizes up to 2 tons (Euclid-Armington). The size of main frame determines distance from overhead track to upper position of hook. It is of steel plate with welded joints, and holds drum, drum shaft and motor brake. Motor and controller are bolted to one side, and an oil tight case with spur gears and brake to the other. All shafts have roller bearings and all load-carrying parts are steel. Table 6 gives dimensions of trolley hoists with drum at right angles to run of track, enabling the hoist to work close to end of the track. With drum axis parallel to track, the hoist can pass through narrow doorways. Similar hoists are made by manufacturers [2], [28], [39], [54], [61], [80], [81], [94] (see p. 23–81). The "Bantam" hoist (Robbins & Myers, Inc.) lists 500 lb. at 30 ft. per min. It is 14 × 21 × 27 in., and weighs 110 lb. without trolley, 145 lb. with trolley. The frame is aluminum alloy with shelf-mounted motor and the usual safety devices.

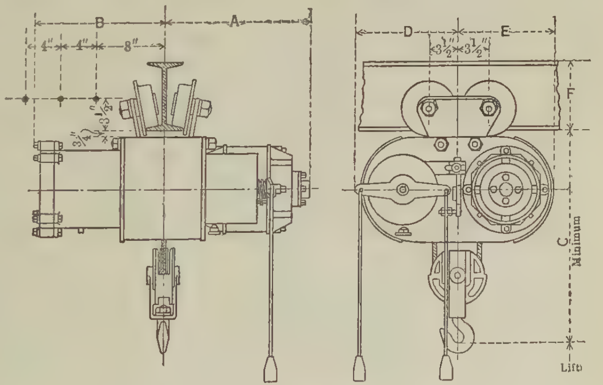


FIG. 15. 2-Ton Spur-geared Hoist

**WORM GEAR HOISTS** are made in sizes up to 6 tons by manufacturers [26], [70], [81]. (See p. 23–81). In this type a hardened steel worm works in a bronze worm wheel.

**Table 6.—Dimensions of Spur-geared Trolley Hoists**
(Euclid-Armington Corp., Euclid, O.)

| Capacity, tons | Standard Lift, ft. | Cable Size, in. | Hoist Speed, ft. per min. | Hp. of Motor | Dimensions, in. (See Fig. 15) | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | *A* | *B* | *C* | *D* | *E* | *F* |
| 1/2 | 20 | 1/4 | 30 | 1 1/2 | 18 | 14 | 16 1/2 | 11 1/2 | 10 | 5 |
| 1 | 16 | 5/16 | 30 | 2 1/2 | 18 | 14 | 16 1/2 | 11 1/2 | 10 | 5 |
| 2 | 17 | 3/8 | 25 | 2 1/2 | 20 | 15 1/2 | 18 | 14 | 11 | 6 |

The thread angle is low enough to make the gears self-locking, thus avoiding a mechanical brake. Some hoists have, beside the worm gears, a pair of spur gears to get greater speed reduction between motor and hoist drum. For foundry work, the worm gear hoist can be fitted with special control. A non-spring-return controller used with an over-sized motor gives a speed range from $^1/_{10}$ to full speed, which is useful in handling flasks and molds. A jointed control rod worked by one hand permits operator to use the other hand to guide the load. Fig. 16 shows a 1 $^1/_2$-ton worm-geared hoist made by Electro-Lift, Inc. Overall length is 26 $^1/_4$ in.; distance from bottom of rail to highest position of hook, 14 in. Corresponding dimensions in the 6 $^1/_2$-ton size are 37 and 23 in. respectively.

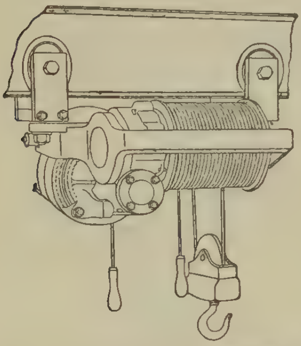


Fig. 16. 1 $^1/_2$-Ton Worm-geared Hoist

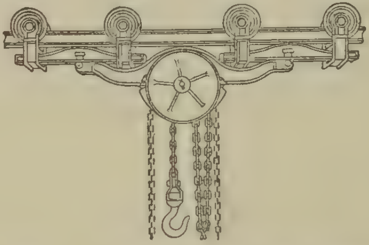


Fig. 17. Tramrail Chain Hoist Carrier

**HOIST CARRIERS.**—Hoists with top hook or clevis may be used as fixed hoists or attached to a commercial trolley. More head-room is required than when trolley and hoist are combined, and wiring is less neat. Most electric hoists can be made with top connections to fasten to an 8-wheel push-type trolley, which are strong, light and can pass curves of small radius. Fig. 17 is typical; overall length is 66 in.; distance bottom of rail to hook, 21 in.

Plain push-type trolleys for I-beam tracks are useful for moderate distances. If load or hook hangs out of reach, or must be spotted accurately, it is better to use a hand-chain geared trolley. If the hoist must travel much, or if the loads are hot, heavy or bulky, the trolley can be motor-driven. (See Fig. 18.) Control cords may be kept away from the hoist by carrying controller on a separate trolley (Fig. 19) when hoist operates through a wide hatchway, from vats or beyond the edge of a platform.

Electric hoists up to 2 tons capacity especially for use with Cleveland tramrail are mounted on 4-wheel hand-propelled carriers for 500-lb. hoists, and 8-wheel carriers for 1- or 2-ton hoists. Fig. 20 shows an electrically propelled carrier mounted with an electric hoist coupled to a trailing operator's platform. Maximum overall length, with single motor drive is 12 ft. 3 in.; operator's platform to bottom of rail, 6 ft. 11 in. minimum; hook to bottom of rail, 29 in. Carriers also are made for pendant cord control, and also with enclosed operating cab. Travel speeds are up to 600 ft. per min. and the carrier can climb grades, run through switches and pass short radius curves. Most builders supply cab-operated machines to travel on single I-beam tracks. For heavy loads and high speeds, the double T-rail track, shown in Fig. 7, is preferable to the lower flange of the I-beam.

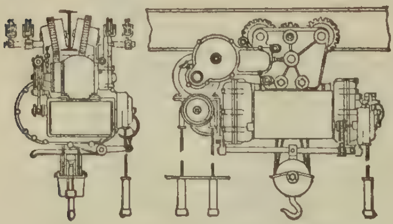


Fig. 18. Motor Driven Trolley Hoist

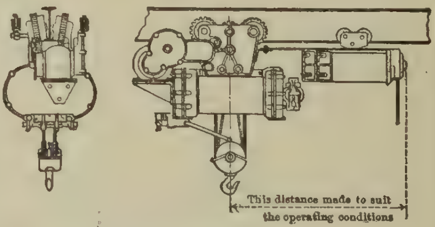


Fig. 19. Motor Driven Trolley Hoist with Separate Controller Trolley

**TELPHER HOISTS.**—In Europe, the track rail for outdoor work usually is a T-rail on a longitudinal supporting beam. The traveling frame, with cab and hoist or grab bucket hangs from the trolleys by hook-shaped hangers which permits some side swing. In England, this is known as the Telpher system. A German monorail carrier of this type, Fig. 21, has no operator but runs under an automatic block control system. Cab control, however, permits higher traveling speed, requires fewer cars, and has less apparatus to maintain.

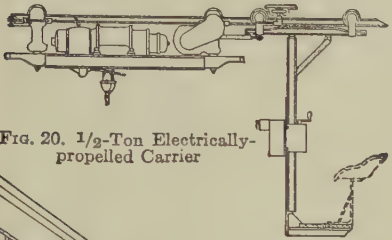


Fig. 20. 1/2-Ton Electrically-propelled Carrier

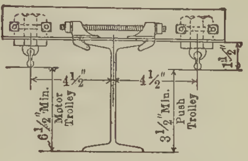


Fig. 22. 8-shaped Conductors and I-beam Track

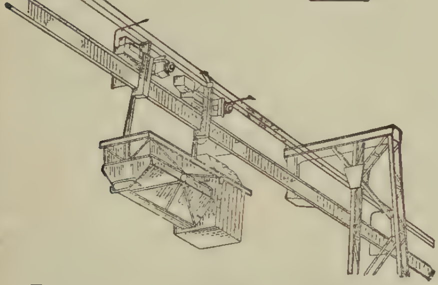


Fig. 21. Automatically Controlled Telpher

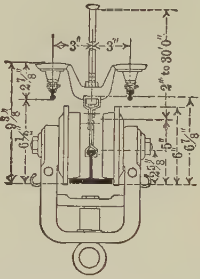


Fig. 23. Tramrail and Electric Conductor Support

**ELECTRIC CONDUCTORS AND COLLECTORS.**—If travel of hoist is short, a flexible cable can be so supported as to allow the necessary movement. If cable drags or interferes, a spring-actuated reel can pay out and take up cable automatically. Such reels are made by Appleton Electric Co., Chicago; J. L. Gleason Co., Boston, and some hoist builders. For long travel, shoe or wheel collectors bear against conductors. These may be copper wire, angle iron, flat bar on edge, or an 8-shaped bar. Fig. 22 shows the latter with an I-beam track, and Fig. 23 a special track and conductor support made for Cleveland tramrail. The American monorail (see p. 23–44) uses small channel conductors on ebonite blocks mounted on the hanger bolts.

To use two conductors with 3-phase current, the primary current is brought through a special transformer. One phase of the secondary current is grounded and connected to the track which is separately grounded. The conductor bars transmit the other two phases. The Cleveland Tramrail Co. in one of its bulletins states that this method of transmitting current to tramrail carriers does not improve operating conditions of alternating-current tramrail motors. Such motors are fundamentally unsuited for conditions requiring delicate speed control and should not be used for extremely severe service. This reference to alternating-current motors applies only to tramrail service, where loads are light and lift is short, and not to hoist and crane service in general.

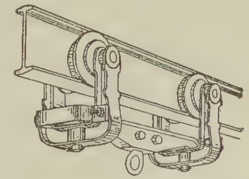


Fig. 24. Special I-Beam Tramrail Track

**TRAMRAIL SYSTEMS.**—The advantages of tramrail systems are: 1. Little machinery required, cost and maintenance low. 2. No floor space required. 3. Switches permit many tracks to be served by few carriers. 4. Overhead tracks for storage of suspended loads avoid the use of floor space. 5. Standard carriers are available for all kinds of bulk and package material.

Louden Machinery Co., Fairfield, Iowa, makes a standard track of I-section, with wide flanges to carry trolley wheels, and for heavier loads a track similar to a 6-in. I-beam with strengthened flanges, one flange wide the other narrow. They are 6 to 16 ft. long, carried by hangers spaced to suit the load. Trolley wheels are drop-forged, ball-bearing, to carry loads up to 5000 lb. See Fig. 24. Trolleys can be driven by motor or operated by hand.

Richards-Wilcox Co. makes an I-beam track (Figs. 8 and 22) for heavy work, and a pressed sheet steel track (Fig. 25) for lighter work, in 10 or 12 ft. lengths of No. 10, 13 or 14 gage steel. Trolley wheels for the size shown are 3 in. diam., ball or roller bearing and are assembled in groups of 4 to 16 with equalizer bars to carry loads up to 3000 lb. The track is made with switches, cross-overs, turntables, drop sections, hangers and brackets to suit different conditions.

Fig. 26 shows American Monorail track, made by American Monorail Co., Cleveland, Ohio, in lengths up to 16 ft., and clamped to drop-forged suspenders. The track is made with staggered joints. It assembles 2 in. wide, 2 1/2 in. deep and will carry loads up to 4000 lb., depending on spacing of hangers and number of trolley wheels. For spans in the open, or where hanger bolts cannot be used, the halves of the rail are welded to the web of a T-beam. Trolley wheels are drop-forged, 4 in. diam., ball bearing, and are assembled in groups of 4 or 8 in carriers with drop-forged wheel yokes, equalizers and load bars.

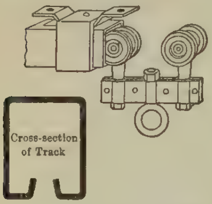


Fig. 25. Pressed Steel Tramrail Track

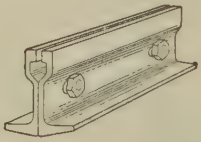


Fig. 26. American Monorail Track

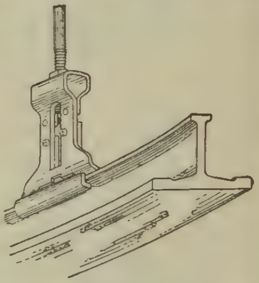


Fig. 27. Cleveland Tramrail Track Support

Fig. 27 shows Cleveland track. It is about 3 × 3 in., 7 lb. per ft. Hanger rods, spaced 3 to 10 ft., support the rail, a clamp on the lower end gripping the rail. A ball-and-socket joint at the upper end permits some free movement, allowing track to adjust itself to load, giving trolley wheels a fair bearing on tread of the rail. On curves, where the load tends to swing outward the rail is free to tilt slightly and wheels on the inside of the curve are not overloaded. Trolley wheels are 3, 4 or 5 in. diam., ball-bearing. Curves, switches, turntables, transfers and fire-door arrangements have been standardized. This sytem uses hand- or motor-propelled carriers, with or without hoists, to carry loads of 500 to 4000 lb. Carriers are made for suspended tiering machines and furnace chargers. Foundry ladles for one-man operation have hoist control buttons on the tilting handle of the ladle. Power-propelled carriers with suspension frames for ladles have controllers for tilt, travel and hoist in the operator's cab or platform. This system also provides automatic dispatch with distance control and special devices for high-speed or continuous travel. See Figs. 28 and 29. With any tramrail system it is possible to arrange track turn-outs to register with various positions of transfer bridges or overhead cranes, enabling carriers to cover a large area.

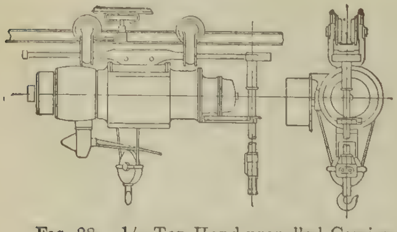


Fig. 28. 1/2-Ton Hand-propelled Carrier

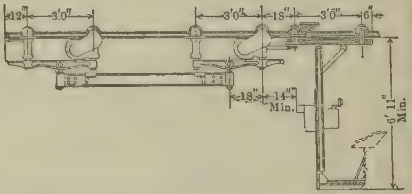


Fig. 29. 2-Ton Electrically-propelled Carrier

## 3. POWER-OPERATED HOISTS

**POWER-OPERATED HOISTS.**—Steam, gasoline and electric hoists for mine, quarries, building erection, logging and grab bucket work are made by manufacturers [3], [21], [29], [49], [56], [96] (see p. 23–81) and others. Single-drum hoists are used for cargo hoisting and building construction, double-drum hoists for general hoisting, and grab bucket work with straight booms. To operate grab buckets on derricks, three drums generally are used, for the holding line, closing line and for luffing the boom, respectively. For operation of grab buckets, see p. 23–20. Swing boom derricks require an extra pair of drums to swing the boom. See p. 23–24. On small rigs, the bucket-holding line can be wound on a drum, separate from the main hoist, operated by a counterweight rope on the same drum and controlled by a brake. With one of these, a hoist with a single drum for the closing line will work a grab bucket (George Haiss Mfg. Co.).

Small winches, winding drums, car pullers and drag scrapers, made to operate by compressed air, are furnished in sizes from 750 to 2000 lb. by manufacturers [42], [86]. (See p. 23–81.) In the Sullivan Turbinair portable hoist, Fig. 30, the air motor is in the hollow of the drum. A 3-Hp. hoist will wind 350 ft. of $^5/_{16}$-in. rope and exert a pull of 1000 lb. at 130 ft. per min. It weighs 175 lb., and is 20 × 15 × 12 in. overall. The handles control clutch and brake. These machines are made with single or double drums or winch heads. They can be bolted to wood or steel plate bases, or clamped to pipe posts or to the masts of derricks. Air-driven hoists mounted on trailer trucks with gasoline engine, compressors and air tanks also are available.

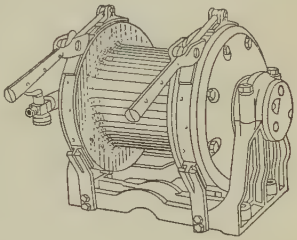


Fig. 30. Single Drum Turbinair Portable Hoist

Many makers of electric hoists supply winches or pulling machines for bolting to floor or ceiling, or mounted on a truck or skids to form a portable winch. American Engg. Co., Philadelphia, makes single-drum pullers of this type in 3 sizes. The smallest is 21 × 23 in., 15 in. high, has a 2-Hp. motor, will wind 250 ft. of $^3/_8$-in. rope, and exert a pull of 1000 lb.

**CAR PULLERS OR SPOTTERS** for moving or spotting freight cars are made by various manufacturers. The Caldwell car spotter (Link-Belt Co.) is a compact machine of the vertical capstan type, with gearing enclosed and running in oil. Two sizes, 5000 lb. and 10,000 lb. pull, are made. The operator throws a few wraps of rope around the capstan and maintains a hand pull on the slack rope as it pays off. See Fig. 31.

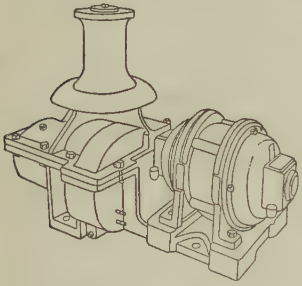


Fig. 31. Car Spotter

**AIR HOISTS FOR SHOP USE.**—Direct lift hoists, Fig. 32, are made by manufacturers [23], [37], [61]. (See p. 23–81.) Cylinders, in diameters up to 19 in. and lengths to suit the work, have lifting capacities up to 10 tons at 80 lb. per sq. in. air pressure. They are made single-acting for simple rapid lifting, or as balanced-pressure cylinders, in which air acts at full pressure on one side of the piston but is varied on the other by a hand-operated valve. The piston moves under a difference of pressure, and speed and travel are accurately controlled. Air cylinders can be hung from trolleys if the travel is not too great.

Advantages claimed for direct acting air hoists are: 1. They will work under conditions of dust, moisture, fumes, etc., injurious to other hoists. 2. Few parts to get out of order; repairs can be made by any mechanic. 3. Accuracy of control. 4. Low first and operating cost, including maintenance and depreciation.

**GEARED HOISTS** operated by compressed air are made by manufacturers [42], [97]. (See p. 23–81.) A balanced 2-, 3-, or 4-cylinder engine drives a wire rope drum or chain pocket-wheel, through gears enclosed in a tight lubricant-filled case. Sizes range from 500 lb. to 10 tons capacity. They weigh 2 or 3 times as much as hand chain hoists but are lighter than the ordinary electric hoists, especially in smaller sizes. They are useful in shops and foundries where compressed air is available and where the hoist can be hung from a fixed point or a trolley with travel not too long for the air hose. They are not easily affected by heat and dust in foundries, and since the valve control is accurate and delicate, patterns, cores and flasks can be handled with certainty and safety. Similar hoists with engines of the square piston type are made by Dake Engine Co., Grand Haven, Mich.

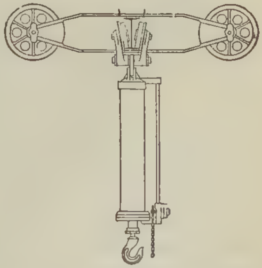


Fig. 32. Direct-lift Air Hoist

# GENERAL CONVEYING AND ELEVATING

## 1. COAL HANDLING IN BOILER HOUSES

**BASIC FACTS FOR BOILER HOUSE DESIGN.**—(Abstracted from Catalog 50 of R. H. Beaumont & Co., Philadelphia.) Assuming that boilers are run at 150% of rated capacity, and a load factor of 66 percent and that the coal consumption is 3.5 lb. of coal per boiler Hp.-hr., then 1000 boiler Hp. requires $1000 \times {}^3/_2 \times {}^2/_3 \times 3\,{}^1/_2 = 3500$ lb. per hr. = $1\,{}^3/_4$ tons per hr. Coal handling machinery should be able to handle a one-day supply in ${}^1/_2$ of an 8-hr. shift. Its capacity therefore should be $10\,{}^1/_2$ tons per hr. per 1000 boiler Hp. The supply bin should hold at least enough for a one-day, or better, a two-day run. The coal larry should be able to handle the daily supply in not more than 8 hr. Stoker hoppers should contain a 3-hr. supply if possible. Good quality bituminous coal produces 10% ash; poor quality, 20%; anthracite culm, 60%. If ashes are handled only on day shift, and in 5 or 6 hr., ash hoppers under the stokers should hold a 16- to 20-hr. run of ashes. If ashes are handled on two or three shifts, ash hoppers can be smaller, height and cost of building less, and hourly capacity of ash handling machinery also can be smaller. Ashes best can be discharged from a storage bin to railroad cars or trucks by daylight The bin should hold a one-day run. If ashes are to be moved in railroad cars the bin should hold a carload plus a margin for uncertainty of car supply.

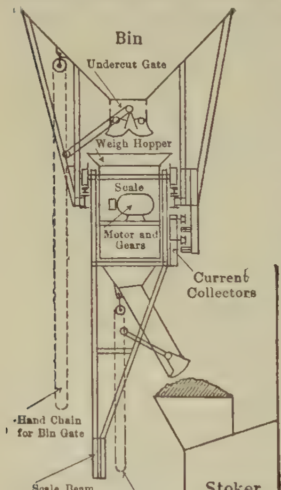


Fig. 1. Floor-operated Weigh Larry

**COAL HANDLING IN BOILER HOUSES.**—Whether it will pay to install coal handling machinery in an old boiler house depends on amount of coal used, probable remaining life of old boiler house, how coal is received, and the ease or difficulty of adapting machinery to the place. All new boiler houses using coal should be designed for coal handling machinery and storage. Even in boiler houses using as little as 5 or 10 tons per day, a simple coal elevator or conveyor usually will pay its way.

Nearly all modern boiler houses use mechanical stokers or burn pulverized fuel, and must have machinery for crushing the coal, elevating and storing it, and feeding it to stokers or pulverizers. Both systems require crushed coal. In small plants it usually is possible to store enough coal in a covered overhead bin to supply boilers regularly and have enough reserve for ordinary contingencies of interrupted coal supply or accident to machinery. In large and important plants reserve storage is necessary to avoid risk of boiler shut-down and also to permit coal to be bought at favorable times and prices. The reserve storage is usually a pile in the open, formed by dumping from an elevated trestle, or by a crane or conveying machinery, or spreading by means of a drag scraper. Fig. 4, p. 23–16, shows a boiler house with an external storage bin and a reserve storage, spread and reclaimed by a drag scraper.

**DESIGN OF BIN.**—In the structural design of a new boiler house, the size and location of storage bin must be determined first. It should hold at least enough coal to carry the plant during interruption to car service or repairs to coal handling machinery. If the bin is directly in front of the boilers, the bottom should be high enough from the floor to avoid interference when boiler tubes are cleaned or replaced. Bins should not obstruct the light. Spouts from stokers should be at 45° or more from horizontal. If the bin is in the house and runs its full length, bottom gates placed close together, serving a traveling hopper, permit more stored coal to be drawn off than would flow to spouts placed to match stokers. The traveling hopper generally is a weigh-hopper to check weight of coal delivered to each boiler. If the bin is outside or at one end of the house, its height is fixed by the traveling hopper, but the boiler house structure can be designed without reference to the bin or its weight. The roof can be lower, and size and shape of bin itself can be independent of restrictions imposed by boiler house dimensions. An outside bin generally is short and can be filled by a simple elevator or skip hoist, whereas a long bin in a boiler house requires more machinery. The structure supporting an outside bin also can house coal handling machinery, and also may carry ash handling machinery and ash storage. In small plants, one skip hoist or one pivoted bucket carrier may handle both coal and ashes.

Fig. 1 shows a weigh-larry operated from the floor; others are cab-operated. Some control opening and closing of bin gates to avoid spilling coal on the floor. The gate will open only when larry is under it and must close before larry can move. Larries are made from 1/2 ton to 10 tons capacity, or more, by most makers of coal handling machinery. In plants with not more than 3 or 4 boilers it is possible to use a separate stationary weigh-hopper at each stoker spout. The investment in scale mechanism, however, is greater and probably there is no saving in labor.

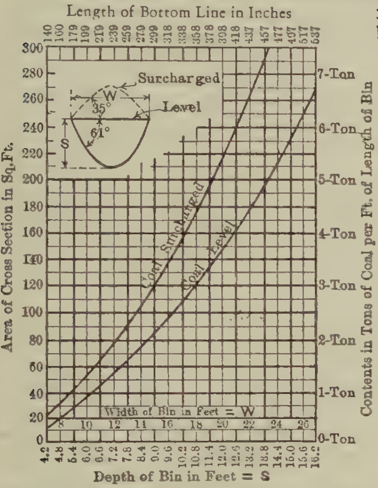


FIG. 2. Capacity of Coal Storage Bins

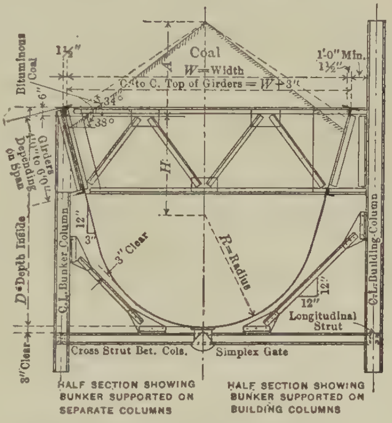


FIG. 3. Beaumont Suspended Coal Bunker

**Shape of Coal Storage Bin.**—In some boiler houses a round tank with a conical bottom, holding 5 to 20 tons, opposite each stoker or pair of stokers is sufficient. When greatest possible storage in boiler house is necessary, the bin may be continuous for the entire length of boiler or of house. Continuous bins can be built most economically as part of the boiler house structure and hung either from roof trusses or supported on building columns. The slope of the bottom always should assist movement of coal toward the spout, and also lessen the danger of fire in deep bins. The best bins, in which weight of structure is low per ton of contents, are the round-bottom suspension type. The steel sides act in direct tension to carry the weight as well as to resist thrust of the coal. Fig. 2 (Link-Belt Co.) shows relations between dimensions of such bins and their capacities in tons of coal, based on a bin depth of 0.6 bin width and with level fillings or a surcharge at a slope of 35°. Fig. 3 (Beaumont) and Table 1 give dimensions of bins where depth is about 0.8 × width, and capacities based on a surcharge at an angle of 38°. The last

**Table 1.—Suspended Steel Coal Bunkers**

(R. H. Beaumont & Co., Philadelphia)

| Tons per ft., Continuous Loading | Tons per ft., Loaded Every 8 ft. | Dimensions (See Fig. 3) | | | | | Tons Lost in 2 Ends |
|---|---|---|---|---|---|---|---|
| | | *W* ft. | *R* | *H* | *D* | *A* | |
| 1.85 | 1.60 | 10 | 4′–0″ | 3′–6″ | 8′–0″ | 3′–4 5/8″ | 0.8 |
| 2.15 | 1.90 | 11 | 4′–4 13/16″ | 3′–10 3/16″ | 8′–9″ | 3′–8 5/8″ | 1.0 |
| 2.55 | 2.30 | 12 | 4′–9 1/8″ | 4′–2 3/8″ | 9′–6″ | 4′–0 3/4″ | 1.5 |
| 3.00 | 2.75 | 13 | 5′–2 3/8″ | 4′–6 5/8″ | 10′–3″ | 4′–4 3/4″ | 2.2 |
| 3.50 | 3.25 | 14 | 5′–7 3/16″ | 4′–10 13/16″ | 11′–0″ | 4′–8 7/8″ | 3.2 |
| 4.00 | 3.75 | 15 | 6′–0″ | 5′–3″ | 11′–9″ | 5′–0 7/8″ | 4.5 |
| 4.55 | 4.30 | 16 | 6′–4 13/16″ | 5′–7 3/16″ | 12′–6″ | 5′–5″ | 6.1 |
| 5.15 | 4.90 | 17 | 6′–9 5/8″ | 5′–11 3/8″ | 13′–3″ | 5′–9″ | 7.8 |
| 5.75 | 5.50 | 18 | 7′–2 3/8″ | 6′–3 5/8″ | 14′–0″ | 6′–1 1/8″ | 10.0 |
| 6.45 | 6.20 | 19 | 7′–7 3/16″ | 6′–7 13/16″ | 14′–9″ | 6′–5 1/8″ | 12.2 |
| 7.10 | 6.85 | 20 | 8′–0″ | 7′–0″ | 15′–6″ | 6′–9 1/4″ | 14.7 |

column refers to the fact that full length of bin cannot be surcharged unless the ends are built higher than the bin body.

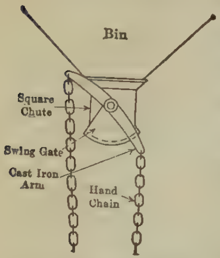


Fig. 4. Swinging Undercut Bin Gate

It is unnecessary to line round-bottom suspension bins with reinforced concrete to prevent corrosion. Coal does not rub the plates except at gate openings where there is some steel reinforcement. Bins for washed anthracite or high-sulphur coal are coated inside with a plastic bituminous cement to resist corrosion.

**Bin gates** should be of the undercut type, Fig. 1 or Fig. 4. Slide gates may stick and are hard to work. Spouts should be at least 12 in. diam., or better, 12 × 12 or 16 in., as wet coal is less apt to stick in large spouts. Openings from bins to stoker spouts should always be controlled by a gate to permit removal of spout for repairs or cleaning or replacement of boiler tubes.

**PULVERIZED COAL IN BOILER HOUSES.**—Two principal systems are: 1. The bin system. 2. The unit pulverizer system.

**Bin System.**—Coal from storage is dried, pulverized, conveyed by a pneumatic system (see p. 23–67) if distance is long, or by screw conveyor if distance is short, to a bin at the boiler. An adjustable feeder under the bin delivers pulverized coal to a blast pipe, thence to the burners. Advantages of this system are: 1. Feed to furnaces can be varied instantly. 2. Coal can be pulverized in off-peak period. 3. Machinery is kept away from the boiler (see H. Kreisinger, *Trans.* A.S.M.E., 1930, FSP–52–36).

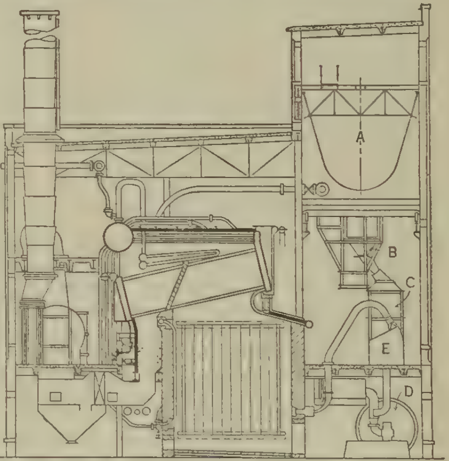


Fig. 5. Kennedy-Van Saun System of Pulverized Fuel

**The Unit Pulverizer System** stores no pulverized coal. The pulverized coal is burned as it is made; it is unnecessary in most cases to dry the coal. Fig. 5 shows the Kennedy-Van Saun system. Crushed coal is stored in bin *A*, drawn off and weighed by larry *B*, stored in small hopper *C*, delivered by an adjustable feeder to tube mill *D*, then as fine dust through blower *E* to the furnace. Pulverizing, conveying and blowing equipment is made by manufacturers [31], [35], [43], [45], [75] (see p. 23–81).

**HANDLING COAL FROM CARS.**—The problem in most boiler houses is to take run-of-mine bituminous coal from railroad cars, crush it and elevate it to bins. In small plants, small anthracite is burned, or lumps of soft coal are broken on a grating over the track hopper or on the boiler room floor by the fireman. The apparatus comprises a track hopper delivering to a simple elevator, Fig. 6, or to an inclined conveyor, Fig. 7. A pile so formed will serve one or two boilers by direct spread on the floor, but for more boilers, a distributing conveyor, Fig. 8, is advisable. To distribute coal in an overhead bin, a scraper conveyor, a belt conveyor or a pivoted bucket carrier can be used in combination with any of the above methods. If the return run of a pivoted bucket carrier can be placed below the boiler room level, it also may be used to remove ashes to a bin.

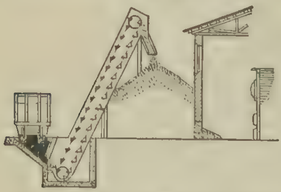


Fig. 6. Track Hopper and Elevator

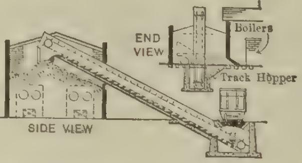


Fig. 7. Track Hopper and Conveyor

If coal is to be crushed, flow from track hopper must be controlled by a feeder. For small anthracite, it generally is sufficient to use a slide gate. For soft coal, smaller than run-of-mine, where flow is direct from track hopper to elevator or conveyor, and where the rate of handling is slow, an undercut gate, hand controlled, can be used. To avoid risk of flooding conveying system, and to prevent breakdowns, a feeder generally is advisable.

**TRACK HOPPERS** can be made as part of a concrete foundation, especially in small sizes and where no feeder is used. That shown in Fig. 6 has a hand-operated undercut gate. If a feeder is used, it is most convenient to attach the parts to a steel hopper. If proper access is provided to back and both sides of a feeder, concrete construction becomes costly. Steel track hoppers usually are of $^1/_4$-in. plate with angle stiffeners. At track level they should be at least 10 ft. wide to give room for lump between track girders and hopper sides. Hoppers above ground level or on trestles should be 14 ft. wide to catch flow from car doors and to prevent spill. In general, hoppers 14 ft. wide are advisable for rapid unloading of bottom-dump cars. Length of hopper along tracks should be at least 12 ft., but for quick work with the largest cars 24 ft. is better, as it will take half the discharge of a car at one setting. Link-Belt Co. has made hoppers 40 to 44 ft. long to empty a car at one setting. Such hoppers have a double-acting plate-feeder working parallel to the track. Track hoppers for skip hoists usually have automatic gates to fill the skip to capacity without spilling. See p. 23–50.

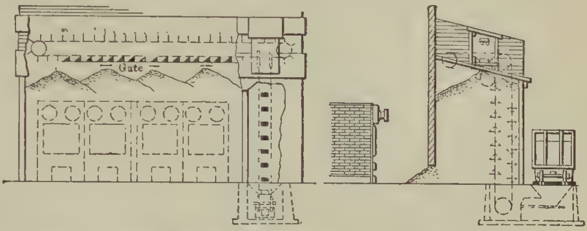


Fig. 8. Track Hopper, Elevator and Distributing Conveyor

Track girders consist of double I-beams, plate-and-angle girders, or Bethlehem wide-flange beams. They must be designed for dead loads, live loads and impact to meet railroad specifications, and generally are subject to the approval of railroad engineers. See Stephens-Adamson Mfg. Co., catalog No. 30 for design based on such specifications.

The time required to spot a car, open doors, get coal out, close doors and move car away varies with the size of car and the weather. Under favorable conditions, the total time for a 50-ton car may be 30 min.; for a 100-ton car, 45 min.; that is, rate of delivery to a one-track hopper varies from about 2 tons a minute in good weather, to, say, 1 ton in bad freezing weather. A one-track hopper will feed a continuous conveying system if average hourly capacity is not over 100 tons. The size of hopper is not important if its area at track level allows the car to be unloaded without difficulty. If a continuous system must deliver more than 2 or 3 tons per min., it may be necessary to dump on two tracks or to use a rotary car dumper. If track hopper delivers to a car or skip, volume of the hopper must be such as will keep skip supplied during those intervals when no coal enters the hopper. A proper size is: Hopper capacity, tons, = 0.80 × rated capacity of skip hoist, tons per hr. Fig. 9 shows a rotary car dumper feeding a skip hoist.

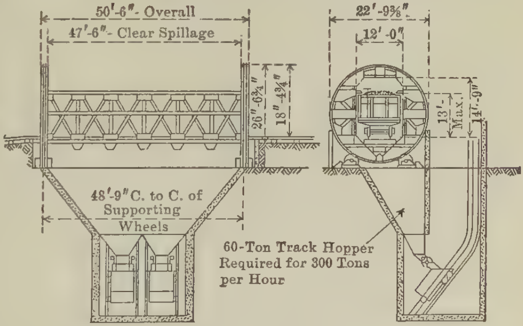

50′-6″- Overall
47′-6″- Clear Spillage
26″-6¾″
18″-4¾″
48′-9″ C. to C. of Supporting Wheels
22′-9⅜″
12′-0″
13′- Max.
14′-9″
60-Ton Track Hopper Required for 300 Tons per Hour

Fig. 9. Rotary Car Dumper Feeding Skip Hoist

**COAL CRUSHERS.**—The kind, size and speed of crusher depends on the kind of coal, whether hard or soft, size of largest pieces, size to which it is to be crushed and quantity per hour. For dimensions, capacity and power required by single- and double-roll crushers, see catalogs of makers of conveying machinery. For crushing finer than 1 in. in large capacities, hammer mills or ring roll mills often are used. To prepare coal in large quantities for subsequent pulverization, the Bradford breaker often is used instead of roll crushers. As made by the Pennsylvania Crusher Co., Philadelphia, and others, it is a steel cylinder 6 to 12 ft. diam., 8 to 20 ft. or more long, with a shell of perforated steel plate. Coal fed in at one end of the cylinder is lifted up and tumbled until it is small enough to pass the perforations. The hard slaty refuse and stray iron are rejected at the other end of the cylinder. The machine acts both as a crusher and as a separator and protects pulverizing and conveying machinery from damage.

**SKIP HOISTS** comprise a single-drum winding machine, a wire rope, a skip or bucket which takes its load below, and dumps it above, together with the necessary lead sheaves, skip filling device and electrical control apparatus for the winding machine. Skip hoists are simple in construction and relatively cheap to maintain and operate. Carrying capacities can be made to suit the smallest or largest boiler houses. The path of the skip can be vertical, inclined or a combination of both. Carrying capacities depend on size of skip, speed of travel, method of loading, whether or not the skip is counterbalanced, and whether operation is full or semi-automatic. Makers' catalogs give tables of expected capacity under various conditions. Skip hoists have been well standardized by manufacturers [11], [50], [98], (see p. 23–81), and other makers of conveying and hoisting machinery. Their catalogs give detailed information about track hoppers, loaders, skips, steel structures, winding machines and electrical control apparatus. Fig. 10 shows a skip hoist where an apron feeder was used to reduce the depth of pit under the track hopper. This arrangement is not recommended. It is simpler and better to feed directly from the track hopper through a chute.

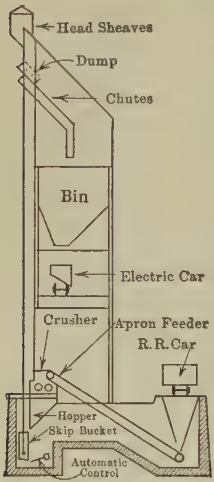


Fig. 10. Skip Hoist

**Skip Hoist with Traverse Motion.**—A form of skip hoist, first used to load anthracite or coke to silo bins in retail coal yards, is made by manufacturers [33], [50], [99]. (See p. 23–81.) A latched-bottom bucket takes its load of 1 or 2 tons from a track hopper with an automatic feed chute. When hoist is started the bucket rises, comes in contact with a trolley carriage, moves horizontally to a chosen point, then lowers. When bottom of the bucket touches material in the bin, the doors open and contents slide out. The bucket then returns for a new load. Some of these hoists have full-automatic push button control. See F. D. Campbell, *Trans.* A.S.M.E., 1931, MH–53–1; also W. W. Sayers, *Trans.* A.S.M.E., 1931, MH–53–8.

## 2. CARRIERS AND ELEVATORS

**PIVOTED BUCKET CARRIERS** consist of round-bottom buckets, pivoted between two long-pitch roller chains, arranged to dump on a horizontal run by tipping the bucket. They can be run vertically, horizontally, or in an inclined path. Suitably combined with crusher and feeder, they fit many arrangements of railroad track, boiler house and storage bin. Where lift, travel and capacity required are not too great, a pivoted bucket carrier is economical, and convenient to install and operate. If carrier is not used for coal more than an hour or two a day, it may in some cases also be used for ashes. If ashes are in large clinkers some other system will be better.

Most carriers of this type have overlapping buckets which can be fed continuously from a chute. The chains have hardened pins and bushings, self-oiling rollers and other refinements. The difference between different makes is chiefly in the suspension of the buckets, and making them lap properly at corner turns. Sizes of buckets, capacities, clearances, power required are practically the same for all such carriers made in the U. S. See Tables 2 and 3.

The horsepower required at the head-shaft of a pivoted bucket carrier is

$$\text{Hp.} = AL + BR \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where $L$ = total vertical lift, ft., $R$ = total horizontal run, ft.; $A$ and $B$ are factors whose value is given in Table 3. Add 10% for each cast gear drive, and 5% for each drive through belt, chain or cut gears.

Pivoted bucket carriers are quiet in operation, convey and elevate at the same time and thus avoid transfers and deep pits. They require little power. They cannot be fed on a vertical run nor under a turn wheel. The dump must always be on a horizontal run. Catalogs of manufacturers [17], [44], [50], [85], [89], (see p. 23–81), give dimensions of drive groups, corner turns and clearances.

**CENTRIFUGAL DISCHARGE ELEVATORS.**—The simplest form of elevator consists of malleable-iron buckets fastened at intervals to a single chain. It is called a "centrifugal discharge" machine because the contents of the buckets are thrown out at the head-wheel by centrifugal force. A full, clean discharge into the head-chute depends chiefly on the size of the head-wheel, its speed and the bucket spacing. Table 4 gives the result of practical experience with vertical centrifugal discharge coal elevators.

Belts with buckets attached are used for fine, dry or gritty materials, but chain is better for the rough work of elevating coal. Fig. 11 shows Style A malleable-iron buckets, used in coal elevators. The buckets will not discharge clean if set too closely together. For analysis of conditions of pick-up and discharge of centrifugal bucket elevators see Belt Conveyors and Belt Elevators, by F. V. Hetzel. (John Wiley & Sons.)

Table 5 gives the closest possible spacing for the speeds given in Table 4. If spacing does not match pitch of chain, increase it to a multiple of that pitch. If the coal is lumpy, increase the spacing. To determine the size of chain, working strengths alone, as given in makers' catalogs, are not a suitable guide A link should be chosen which bears a proper proportion to the dimensions of the bucket, and which has a bucket attachment large enough for the bucket. K 1 attachments have two bolt holes; K 2 attachments have four bolt holes and are advisable for buckets 8 × 5 in. and larger. The greater the projection of the bucket, the greater should be the pitch of the link to give the bucket a proper support in picking up its load. Table 6 represents good practice.

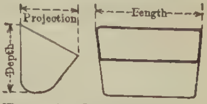


Fig. 11. Malleable Iron Elevator Bucket

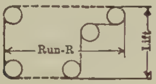
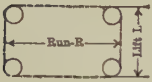
### Table 2.—Standard Sizes of Pivoted Bucket Carriers

(Link-Belt Co., Philadelphia)

| Size of Bucket, in., Length × Width | Pitch of Chain, in. | Carrying Capacity of Bucket, cu. ft. | Coal Capacity, tons per hour | Speed, ft. per min. |
|---|---|---|---|---|
| 18×15 | 18 | 0.68 | 15– 20 | 30–40 |
| 18×21 | 18 | 0.94 | 20– 30 | 30–40 |
| 24×18 | 24 | 1.68 | 40– 50 | 40–50 |
| 24×24 | 24 | 2.24 | 55– 70 | 40–50 |
| 24×30 | 24 | 2.80 | 75–100 | 40–50 |
| 24×36 | 24 | 3.36 | 90–120 | 40–50 |
| 30×24 | 30 | 3.50 | 95–120 | 45–60 |
| 30×30 | 30 | 4.37 | 110–160 | 45–60 |
| 30×36 | 30 | 5.25 | 140–190 | 45–60 |
| 36×36 | 36 | 8.50 | 210–330 | 50–80 |

### Table 3.—Horsepower Factors of Pivoted Bucket Carriers

(Jeffrey Mfg. Co., Columbus, Ohio)

| | Size of Bucket | | Loaded on One Vertical and Both Horizontal Runs | | | |
|---|---|---|---|---|---|---|
| | | | Power Factors | | | |
| | | | Material 50 lb. per cu ft. | | Material 100 lb. per cu. ft. | |
| | Length, in. | Width, in. | *A* | *B* | *A* | *B* |
| | | | For Lift *L*, ft. | For Run *R*, ft. | For Lift *L*, ft. | For Run *R*, ft. |
|  | 18 | 18 | 0.044 | 0.017 | 0.088 | 0.023 |
| | 24 | 18 | .074 | .039 | .148 | .051 |
|  | 24 | 24 | .097 | .045 | .194 | .060 |
| | 24 | 30 | .120 | .051 | .240 | .069 |
| | 30 | 30 | .198 | .061 | .396 | .085 |
| | 30 | 36 | .238 | .070 | .476 | .100 |

### Table 4.—Speeds for Vertical Centrifugal-discharge Coal Elevators

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Diameter, head wheel, in. | 18 | 21 | 24 | 27 | 30 | 36 | 42 |
| Revolutions per minute | 46 | 43 | 41 | 39 | 37 | 34 | 32 |
| Speed of chain, feet per minute | 217 | 237 | 257 | 276 | 290 | 320 | 352 |

Note.—Permissible variation not over a few r.p.m. either way.

### Table 5.—Minimum Spacing Style A Buckets for Clean Discharge

| | | | | | |
|---|---|---|---|---|---|
| Projection of bucket, in. | 4 | 5 | 6 | 7 | 8 |
| Closest spacing, in. | 10 | 12 | 14 | 17 | 20 |

### Table 6.—Pitch of Chain for Style A Malleable-iron Buckets

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Size of bucket, in., length × projection | 6×4 | 7×4 1/2 | 8×5 | 10×6 | 12×7 | 14×8 | 16×8 |
| Pitch of malleable-iron chain, in. | 2.6 | 2.6 | 3.1 | 3.1 | 4.7 | 6 | 6 |

### Table 7.—Weights and Capacities, Style A, Malleable-iron Buckets

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Size of bucket, in., length × projection | 6×4 | 7×4 1/2 | 8×5 | 10×6 | 12×7 | 14×8 | 16×8 |
| Weight, lb. | 2.2 | 3 | 4 | 6.4 | 9.6 | 13.2 | 21 |
| Coal capacity, lb. (*C* in formulas) | 1.5 | 2.2 | 3 | 5.2 | 8.3 | 13.9 | 18.9 |

Table 7 gives weights and capacities of buckets used in simple coal elevators. The diameter of the foot-wheel of a centrifugal discharge coal elevator should be at least four times the pitch of chain used. If the wheel is too small, buckets will not fill properly. The following formulas are applicable to bucket elevators:

**Notation.**—$C$ = lb. coal per bucket (see Table 7); $D$ = diam. of foot-wheel, in.; Hp. = horsepower to be delivered to the elevators; $L$ = height of lift, ft.; $P$ = pull in chain, lb., due to pick-up of coal at foot of elevator; $S$ = spacing of buckets, in.; $T$ = travel of chain, ft. per min.; $U$ = unbalanced pull in chain due to weight of coal in buckets; $W$ = capacity of elevator, lb. coal per minute; $X$ = theoretical horsepower required.

$$P = 12\,DC/S \qquad [2]$$

$$U = 12\,LC/S \qquad [3]$$

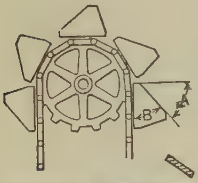


Fig. 12. Continuous Bucket Elevator

Combining [1] and [2],

$$X = 12\,(C/S)(L + D)(T/33{,}000) \qquad [4]$$

Allowing 30% for losses in power transmission,

$$\text{Hp.} = 0.0005(CT/S)(L + D) \qquad [5]$$

$$W = 12\,CT/S \qquad [6]$$

The head-chute of a centrifugal discharge coal elevator should have a slope of at least 35 deg. and its upper end should not be above a line drawn at 45 deg. through the center of the head-wheel.

The lower wheel of a centrifugal discharge coal elevator generally is enclosed in a cast-iron or steel plate box, called a boot, which serves as a support for shaft bearings and also confines material to the path of the buckets. Boots with fixed bearings generally are not used for coal elevators, those with adjustable bearings being convenient. For sizes and dimensions, see makers' catalogs. For diameter of head-shaft, see p. 23–56.

**Advantages and Limitations.**—The centrifugal discharge elevator is cheap and simple, takes up little space, does not require deep or large pits, and can be encased easily. It will give good service when worked for a few hours each day, but for lumpy material, large capacities and long hours of service, it is better to use a heavier machine at slower speed. Such machines comprise continuous bucket elevators, perfect-discharge elevators, and V-bucket elevators.

**CONTINUOUS BUCKET ELEVATORS** consist of single or double chains with an endless series of steel buckets so shaped that in passing over the head-wheel each bucket acts as a deflector or chute for the discharge of the contents of the following bucket. Single strand machines use K 2 attachments fastened to the backs of the buckets, the depth of the bucket being 2, or rarely 3, pitches of the chain. Angles usually are fastened to the bucket ends to guide the vertical runs. Long buckets for large capacities require two chains, fastened to the ends. The chains usually are of steel, with a pitch equal to the depth of the bucket. In this position the chains guide the vertical runs. For inclined runs, they can be fitted with rollers that run up and down on suitable tracks. Buckets for vertical elevators handling crushed coal, Fig. 12, are made with the angle $B$ in the bottom equal to 55 deg., thus giving a chute or discharge angle of 35 deg., which is steep enough for crushed coal. If the coal is all fine or wet, $B$ should be 45 deg.; angle $A$ then is 45 deg., giving a better delivery into the head-chute. When the elevator is inclined 25 or 30 deg. from the vertical, as in Fig. 13, $B$ can be increased to 70 deg. to give more bucket capacity, and for ordinary crushed coal the discharge will be clean. This is one advantage of inclining such an elevator; another is that material can be delivered directly to the buckets by a chute without spill. These buckets will pick up soft material like crushed coal from a boot, but not hard lumpy material like stone. Stone and ores should be handled in inclined elevators, with the feed directly from a chute set 2 or 3 buckets above the foot-wheel, and the foot of the elevator must be free and open to allow spilled material to fall clear of the buckets and be shoveled up. The head-chute never must be above a line drawn at 45 deg. through the center of the head-wheel. Speeds range from 90 to 200 ft. per minute in handling coal.

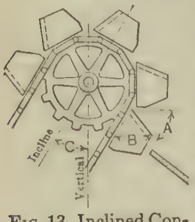


Fig. 13. Inclined Continuous Bucket Elevator

For horsepower required by continuous bucket elevators, use formula [5]; for capacities use formula [6]; for diameter of head-shaft, see p. 23–56.

The continuous-bucket elevator is especially adapted for handling stone, but in elevating crushed coal for boiler houses it has the merit of giving high capacities at speeds much lower than those of centrifugal discharge elevators. For handling run-of-mine or large lump coal the V-bucket elevator is generally better.

**PERFECT-DISCHARGE ELEVATORS** have two strands of chain fastened to the ends of the buckets. The buckets are not continuous and the speed is slow as compared with centrifugal discharge elevators. In order to effect a clean discharge, the chains are deflected backward under the head-wheels by a pair of supplementary wheels in such a way that the contents of each bucket will fall clear of the bucket ahead. Fig. 14 shows the location of the deflector wheels with reference to the head-wheels. Table 8 is based on good practice and gives dimensions for elevators using malleable-iron, Style A buckets and with wheels arranged according to Fig. 14. These elevators often are built with sheet steel buckets for larger capacities of coal, at speeds between 100 and 150 ft. per minute.

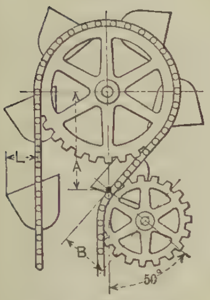


Fig. 14. Perfect-discharge Elevator

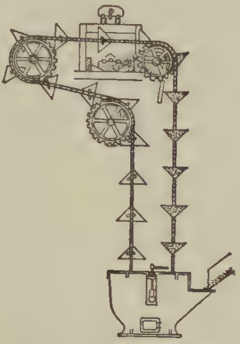


Fig. 15. V-Bucket Carrier

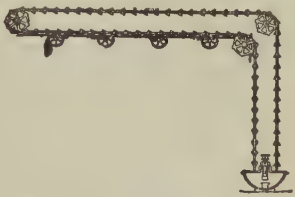


Fig. 16. V-Bucket Carrier with Boot

Measure *A* down from center of head-wheel. Draw line 50 deg. from vertical. Measure *B* from intersection to center of deflector.

For horsepower use formula [5]; for capacities use formula [6]; for diameter of head-shaft, see p. 23–56.

**Advantages of the Perfect-discharge Elevator.**—1. It has two chains, and as usually designed, has a large margin of safety between the working strain and the breaking strength of the chains. 2. The pick-up of material is easy. 3. On account of the slow speed, the chains, buckets, wheels, gears and bearings last longer and require less attention and repair. As an offset, the machine costs more than a single-strand elevator.

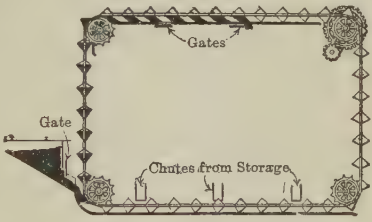


Fig. 17. V-Bucket Carrier in Boiler House

**THE V-BUCKET CARRIER GRAVITY DISCHARGE MACHINE** is used extensively for handling coal, either lump or crushed. It consists of two roller chains fastened to the ends of V-shaped buckets which carry the coal on vertical or inclined runs and push it along a trough on the horizontal run. Gates in the trough permit the coal to be discharged where it is wanted. When arranged as in Fig. 15, or Fig. 16, it will pick up from a boot; with the path shown in Fig. 17, it will pick up at any point on the bottom run as well as at the corner turn. In boiler house work, machines arranged as in Figs. 15 and 16, will take coal from a track-hopper, a feeder or a crusher, elevate it and discharge into a belt or flight-conveyor for distribution in overhead bins. The machine arranged, as in Fig. 17, can form a storage pile near the boiler house, or by discharge into an overhead conveyor, transfer direct from cars to the boiler house. Coal is reclaimed from storage by delivery into the lower run located in a tunnel beneath the pile. It is then elevated and discharged to the overhead conveyor for distribution in the boiler house.

Table 8.—Data for Perfect-discharge Elevators, Style A Buckets

| | | | | | | |
|---|---|---|---|---|---|---|
| Projection of bucket, in. | 5 | 6 | 7 | 8 | 9 | 10 |
| Center of chain from lip, in. (*L*, Fig. 14) | 3 | 3 3/8 | 4 1/2 | 5 3/8 | 5 7/8 | 6 1/4 |
| Diameter of head-wheel, in. (Fig. 14) | 22 | 24 | 26 | 28 | 30 | 32 |
| Diameter of deflector wheel, in. (Fig. 14) | 14 | 15 | 16 | 17 1/2 | 19 | 20 |
| Revolutions per min. of head-wheel | 19 | 19 | 19 | 19 | 19 | 19 |
| Speed of chain, ft. per min. | 109 | 119 | 130 | 140 | 149 | 158 |
| Spacing of buckets, in. | 17 | 19 | 21 | 22 | 23 | 25 |
| Buckets per min. | 77 | 75 | 74 | 76 | 77 | 76 |
| Distance *A* (Fig. 14), in. | 18 | 19 | 21 | 23 | 24 | 26 |
| Distance *B* (Fig. 14), in. | 7 | 7 1/2 | 8 | 8 3/4 | 9 1/2 | 10 |

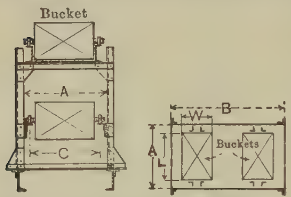
Figs. 18 and 19 show typical cross-sections. Table 9 gives capacities and dimensions for sizes commonly used in boiler houses. Table 10 gives the horsepower required for some of the sizes.

The 12-in. and 16-in. buckets sometimes are carried on malleable-iron roller chains, but steel bar chains generally are used for the larger sizes, and are better for the small sizes also. Take-ups or adjustable bearings always should be placed at a corner where the buckets pass empty, not at a corner around which coal travels. Vertical runs always should be encased to confine the spill and to avoid exposure to rain and snow.

**Advantages of V-bucket Carrier.**—1. Strong and rugged; will carry any coal up to run-of-mine size, at speeds up to 120 ft. per minute. 2. Will receive on any run or from a boot. 3. Handles coal without breaking it. 4. Gives larger capacities than pivoted bucket carriers.

**Disadvantages.**—1. Requires more power on horizontal runs than other types. 2. Not suitable for ashes or hard and gritty material. 3. As compared with other elevators, it cannot discharge over a head-wheel, but always must discharge on a run between two sets of wheels.

**HORIZONTAL DISTRIBUTION BY BELT OR FLIGHT CONVEYOR.**—Inclined conveyors sometimes are used for elevating coal when the boiler house is at a distance from the track. If the house is not too long, the inclined conveyor may be carried over a hump into the horizontal, but if the house is long, or if its axis is not in line with the conveyor from the track, a separate conveyor must be used for distribution in the house. A similar condition exists in boiler houses where the ash disposal is by car or hoist. In such cases, where a pivoted-bucket or a V-bucket carrier cannot well be used for distributing, a simple coal elevator or a hoist is used for lifting the coal.



Figs. 18 and 19. Cross Sections of V-Bucket Carrier

Whether to use a belt conveyor or a flight conveyor depends partly on the structures and operating conditions, and partly on the personal preferences of engineers and owners. Considering only physical conditions, belt conveyors may be regarded as best: 1. Where the bin is long, say, 150 ft. or more. 2. Where the capacity is more than 100 tons an hour. 3. Where noiseless operation is essential.

A flight conveyor may be best: 1. Where there is no room to admit a belt-tripper. 2. Where the first discharge is less than 20 ft. from loading point of the conveyor, a distance too short to admit a tripper. 3. When the conveyor is shorter than 50 ft. center to center. 4. Where separate spaced bins are used, instead of one continuous bin. 5. Where the conveyor is exposed to rain, snow or freezing.

**Table 9.—Capacities and Variable Dimensions of V-bucket, Gravity-discharge Machines**

(Link-Belt Co., Philadelphia)

| Size of Bucket, in. $L \times W$ | Capacity in Tons per Hour at 100 ft. per min. | | | | See Figs. 18 and 19 | | |
|---|---|---|---|---|---|---|---|
| | Spacing of Buckets | | | | *A* | *B* | *C* |
| | 18 in. | 24 in. | 36 in. | 48 in. | in. | ft. in. | in. |
| 12×12 | 30 | 22 | 15 | 11 | 20 3/4 | 4 0 1/2 | 15 1/8 |
| 16×12 | 40 | 30 | 20 | 15 | 24 3/4 | 4 0 1/2 | 19 1/8 |
| 16×15 | 60 | 45 | 30 | 22 | 24 3/4 | 4 8 | 19 1/8 |
| 20×15 | 78 | 59 | 39 | 29 | 28 3/4 | 4 8 | 23 1/8 |
| 24×15 | 90 | 68 | 45 | 34 | 32 3/4 | 4 8 | 27 1/8 |
| 20×20 | 140 | ...... | 70 | ...... | 32 1/2 | 5 9 1/4 | 24 1/2 |
| 24×20 | 165 | ...... | 82 | ...... | 36 1/2 | 5 9 1/4 | 28 1/2 |
| 30×20 | 200 | ...... | 100 | ...... | 42 1/2 | 5 9 1/4 | 34 1/2 |
| 36×20 | 250 | ...... | 125 | ...... | 48 1/2 | 5 9 1/4 | 40 1/2 |

**Table 10.—Horsepower of Elevator Conveyors with V-buckets**

(Link-Belt Co., Philadelphia)

| Size of Bucket, in. | 12 by 12 | | | 24 by 15 | | | 36 by 20 | |
|---|---|---|---|---|---|---|---|---|
| Spacing, in. | 18 | 24 | 36 | 18 | 24 | 36 | 18 | 24 |
| Hp. for each 10 ft. vertical lift | 0.46 | 0.35 | 0.23 | 1.35 | 1.0 | 0.67 | 3.8 | 1.9 |
| Hp. for each 100 ft. horizontal, running empty | 1.2 | 1.1 | 0.9 | 1.7 | 1.5 | 1.3 | 2.4 | 1.6 |
| Hp. for each 100 ft. horizontal, handling anthracite | 2.5 | 2.1 | 1.5 | 5.3 | 4.3 | 3.1 | 12.4 | 6.6 |
| Hp. for each 100 ft. horizontal, handling bituminous | 3.2 | 2.6 | 1.9 | 7.4 | 5.8 | 4.2 | 18. | 9.4 |

Add 5% for each turn.

**Table 11.—Dimensions and Capacities, Single-strand Coal Conveyors**
(Link-Belt Co., Philadelphia)

| Size of Flight, in. | 14×6 | 19×8 | 24×10 |
|---|---|---|---|
| Coal per flight, lb. ($M$) (See formula [7]) | 30 | 50 | 85 |
| Ordinary speed, ft. per min. ($T$) (See formula [7]) | 75–150 | 75–150 | 75–150 |
| Ordinary spacing, in. ($S$) (See formula [7]) | 18– 36 | 18– 36 | 18– 36 |
| Weight one flight complete, lb. | 21 | 25 | 37 |
| Weight usual chain, per ft., lb. | 7 | 8 | 9 |
| Dimension $A$, in. (See Fig. 20) | 23 3/4 | 29 | 33 |
| Dimension $B$, in. (See Fig. 20) | 7 3/8 | 9 1/4 | 11 1/4 |
| Dimension $C$, in. (See Fig. 20) | 18 1/4 | 23 3/8 | 28 3/8 |
| Dimension $D$, in. (wood) (See Fig. 20) | 9 3/4 | 11 3/4 | 11 3/4 |
| Dimension $E$, in. (steel) (See Fig. 20) | 7 | 9 | 12 |

**Table 12.—Dimensions and Capacities, Double-strand Coal Conveyors**
(Link-Belt Co., Philadelphia)

| Size of Flight, in. | 16×8 | 20×10 | 24×10 | 30×12 | 36×12 |
|---|---|---|---|---|---|
| Coal per flight, lb. ($M$) (See formula [7]) | 55 | 86 | 103 | 150 | 180 |
| Ordinary speed, ft. per min. ($T$)(See formula [7]) | 75–150 | 75–150 | 75–150 | 75–150 | 75–150 |
| Ordinary spacing, in. ($S$) (See formula [7]) | 18– 36 | 18– 36 | 24– 36 | 24– 36 | 24– 36 |
| Weight one flight, lb. | 8 | 16 | 19 | 28 | 32 |
| Weight usual chain, single, lb. | 10 | 11 | 12 | 16 | 18 |
| Dimension $A$, in. (wood) (Fig. 21) | 17 | 21 | 25 | 31 | 37 |
| Dimension $C$, in. (wood) (Fig. 21) | 28 | 32 | 36 1/2 | 48 1/2 | 54 1/2 |
| Dimension $F$, in. (wood) (Fig. 21) | 10 3/8 | 12 1/8 | 12 1/8 | 14 5/8 | 14 1/4 |
| Dimension $J$, in. (wood) (Fig. 21) | 17 | 21 | 25 | 31 | 37 |
| Dimension $B$, in. (steel) (Fig. 21) | 16 1/2 | 20 1/2 | 25 | 30 1/2 | 36 1/2 |
| Dimension $D$, in. (steel) (Fig. 21) | 25 | 29 | 33 1/2 | 39 | 45 |
| Dimension $G$, in. (steel) (Fig. 21) | 6 | 8 | 8 | 10 | 10 |
| Dimension $K$, in. (steel) (Fig. 21) | 16 1/2 | 20 1/2 | 25 | 30 1/2 | 36 1/2 |
| Dimension $E$, in. (either) (Fig. 21) | 17 | 21 | 25 1/2 | 31 | 37 |

For either machine, these limits are not hard and fast, and between them there are chances to exercise personal preference or judgment. Both machines have been on the market for many years and have been standardized. There is no " universal " conveyor; for handling heavy or gritty ores, as in the Western smelters, or in conveying heavy tonnages of coal over long distances, as in by-product coke plants, a flight conveyor is not to be considered, but in many boiler houses it makes the best distributor, even though it requires more power. The older forms of flight conveyor were noisy, but modern roller-flight conveyors, or double-strand roller-chain conveyors, are by comparison almost noiseless. They are strong and rugged, easy to load and discharge, and will work under conditions and lack of care that would be harmful to belts. Some engineers have a prejudice against scraping the coal in a trough. It does consume more power, but it does not harm even anthracite. As to wear on the trough, the item of replacement is not an important one.

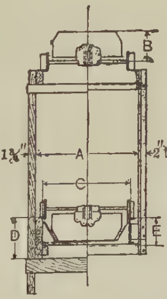


Fig. 20. Single Strand Flight Conveyor

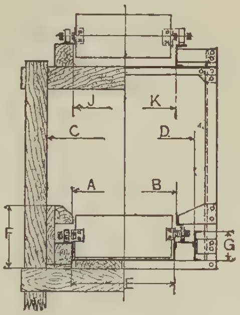


Fig. 21. Double Strand Roller Conveyor

**FLIGHT CONVEYORS FOR BOILER HOUSES.**—Two forms are in general use. Fig. 20 shows a single strand of chain carrying flights with supporting rollers. Fig. 21 shows flights fastened to two strands of chain with rollers mounted on the chain joints. The first is used for crushed coal; the other will carry any size of coal up to run-of-mine. Dimensions and capacities are given in Tables 11 and 12. All flights will carry the stated quantities of bituminous coal on the level, or on inclines up to 20 deg. For a slope of 25 deg., deduct 10%; for 30 deg., deduct 20%; for 35 deg., deduct 30%.

$$\text{The capacity in tons of coal per hour} = 0.36\,MT/S \quad . \quad . \quad . \quad . \quad . \quad [7]$$

**CHAIN PULLS AND HORSEPOWERS.**—The general formula for any ordinary flight conveyor, apron conveyor, bucket elevator or bucket carrier is

$$\text{Pull} = P = X + Y + Z, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

where $X$ = pull to move the weight of material, coal, etc., on the loaded run; $Y$ = pull to move the conveyor parts on the loaded run; $Z$ = pull to move the conveyor parts on the empty run.

$X = WL(F \cos \alpha + \sin \alpha)$; $Y = W_1 L(f \cos \alpha + \sin \alpha)$; $Z = W_1 L(f \cos \alpha - \sin \alpha)$, where $W$ = weight of coal, etc., lb. per ft.; $W_1$ = weight of conveyor parts, lb. per ft.; $L$ = length of conveyor, ft.; $\alpha$ = angle of inclination from the horizontal; $F$ = coefficient of friction for the material (coal, etc.); $f$ = coefficient of friction of moving conveyor parts.

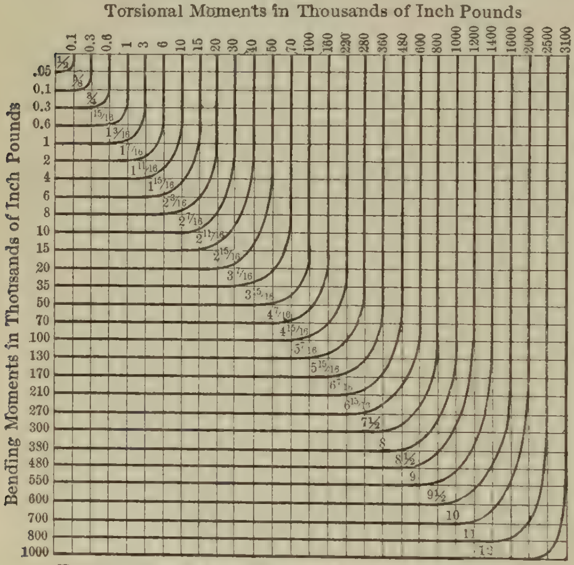


FIG. 22. Bending and Torsional Stress of Head-shafts

For anthracite, $F$ = 0.33; for bituminous coal, $F$ = 0.50. For conveyor parts, sliding in a trough or on tracks, $f$ = 0.33; for conveyor parts, rolling on tracks, $f = Nd \div D$, where $D$ = diameter of chain roller, in.; $d$ = size of hole in roller, in.; $N$ = factor determined by the kind of hole and the grade of lubrication. For a smooth hole, not lubricated, $N$ = 0.5; for a smooth hole, greased, $N$ = 0.33; for a bored hole, greased, $N$ = 0.25; for a bored hole, well oiled, $N$ = 0.20. For vertical lifts, $\cos \alpha = 0$; for horizontal runs, $\sin \alpha = 0$. For ordinary bucket carriers, apron conveyors, etc., where the material is carried, and not scraped, $F = f$.

If material is carried down hill or lowered, $\sin \alpha$ becomes negative and then

$$X = WL(F \cos \alpha - \sin \alpha);\ Y = W_1 L(f \cos \alpha - \sin \alpha);\ Z = W_1 L(f \cos \alpha + \sin \alpha).$$

For combined elevator conveyors of complicated path, make a diagram of the path, calculate the pull for each run separately, note it on the diagram with a plus sign if positive and with a minus sign if the pull is in the direction of motion of the machine. Add the several values, allowing at each turn, where pull is transmitted, 5% of the accumulated pull up to that point. Note $P$, the maximum pull where the chain enters the driving sprocket wheel, and $P_1$, the pull on the chain leaving the wheel. Then $P$ determines the stress in the chain, and $(P - P_1)$ is the torsional pull on the driving-wheel. From this

$$\text{Hp.} = T(P - P_1)/33{,}000 \quad \ldots\ldots\ldots \quad [9]$$

where $T$ = chain speed, ft. per min. For the size of the motor, increase horsepower by 10% for each pair of cast-tooth gears, and by 5% for each drive through belt, chain or cut-gears.

**SIZES OF SHAFTING FOR ELEVATORS AND CONVEYORS OF ALL KINDS.**—Head-shafts and important turn-shafts should be calculated for torsion and bending. From formula [9], $(P - P_1)$ is the torsional pull; this multiplied by the radius of the sprocket wheel gives the torsional moment in inch-pounds. The load which produces bending is the resultant of two forces. One of these is $(P - P_1)$; the other is the weight of the shaft, wheels and gear. In a vertical machine, the resultant is the algebraic sum of the forces. For other machines the weight of shaft, etc., is ignored, if it is proportionately small; otherwise the resultant is found by means of a diagram of forces. The shaft then may be considered as a beam supported by its bearings, and the bending moment in inch-pounds determined. Observing where the bending stress and the torsion stress act together on the same cross-section of the shaft, the shaft diameter may be obtained from the diagram, Fig. 22.

**CHAINS FOR FLIGHT CONVEYORS.**—Malleable-iron hook-joint chains are used for simple flight conveyors without rollers when the pull is not over 1500 lb. For roller flights, malleable-iron, closed-joint chains, or chains of the steel bar type are common. For double-strand conveyors with flights 20 × 10 in. or smaller, and of moderate length, malleable-iron roller-chains are used; for larger flights, and for heavy duty, steel bar-chains are better. These are made in pitches from 9 to 24 in. and of a strength to meet any service.

## 3. ASH HANDLING

**HANDLING ASHES IN BOILER HOUSES.**—For pneumatic system, see p. 23–67; for hydraulic methods, see p. 23–70; for skip hoists, see p. 23–50; for pivoted bucket carriers, see p. 23–50.

In small plants, ashes often are collected by hand car and taken to a waste dump, or discharged into a skip or elevator for storage in a bin. Ashes elevators which work only an hour or two per day are of the centrifugal discharge type (see p. 23–50) with a chain strong enough to withstand grit and corrosion. If the machine must work longer, it is better to use a slower speed elevator or a skip hoist. For handling large clinkers, old brick, soot and other boiler house refuse, a car and skip system is better than any bucket elevator. For high lifts a skip hoist is better under any conditions.

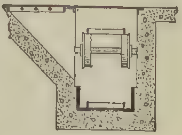


Fig. 23. Ashes Conveyor

A simple ashes conveyor used in some small boiler houses is shown in Fig. 23. A wide malleable-iron chain runs at about 5 ft. per min. in a heavy cast-iron trough and partly drags, partly carries the ashes to the foot of an elevator. It is cheap, requires no lubrication, and lasts long enough to pay for itself in some places.

## 4. ELEVATORS FOR MATERIALS OTHER THAN COAL

It is necessary to consider whether material to be handled is fine or in large pieces, abrasive or non-abrasive, friable or non-friable, wet or dry, whether or not it must be handled at slow speed. It is not possible to give exact rules, but for most cases the following is useful:

1. Ashes, crushed ores, small coal, foundry sand, dry clay, gypsum, acid phosphate and similar materials. Use centrifugal discharge elevators with malleable-iron buckets on single-strand closed-joint, or steel-bushed malleable-iron chain.

2. Grains, feeds, and other free-flowing materials. Use centrifugal discharge elevators with chains lighter than those in (1), or rubber belts, with malleable-iron or sheet-steel buckets.

3. Cement, crushed coal, coke, slag, ores, glass house cullet and other material, not too large, which should be handled at relatively slow speeds. Use single-strand continuous-bucket elevators with steel buckets on closed-joint or steel-bushed malleable-iron chains. These elevators are vertical, or nearly so, and have a boot and casing.

4. For materials in (3), handled in large quantities or to high lifts, use double-strand continuous-bucket elevators with steel buckets on steel chains.

5. Heavy coarse material, ores, crushed stone, gravel, and rock which cannot be picked up in a boot. Use inclined continuous-bucket elevators with one or two strands of chain or a rubber belt.

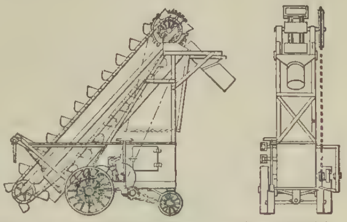


Fig. 24. Self-propelled Loading Machine

Elevators for handling coal in boiler houses and for many other materials have been classified and standardized by manufacturers [44], [50] (see p. 23–81), and other makers of conveying machinery, whose catalogs give dimensions, capacities, speeds, power required, and details of casings, boots, etc. For belt elevators, see handbooks issued by rubber belt makers, Goodyear Rubber Co., Akron, O.; Goodrich Rubber Co., Akron, O.; Manhattan Rubber Co., Passaic, N. J., and others.

**PORTABLE ELEVATORS OR LOADERS** are bucket elevators with a motor or gasoline engine mounted on wheels or caterpillar treads, and arranged to pick up their load at ground level and deliver it to wagons or trucks through a chute, screen or short belt conveyor. Some loaders for hard coal, coke, or excavated gravel have shaking screens. In coal yards, they save time and labor and reduce cost of handling by keeping trucks at work. A man will shovel a ton of coal from ground to wagon in 20 min.; a loader in 3 min. or less. Four men load a 5-ton truck by hand in 30 min.; one man with a loader in 5 min. The system of using loaders in connection with ground storage is preferred by many to storage in overhead bins with chutes to wagons. First cost and depreciation are less and there is less machinery to maintain. For large capacities, or where a machine handles sand, stone or gravel, a self-propelled machine, Fig. 24, is an advantage. The loader backs into the pile by its own power, which is quicker and easier than breaking down the pile by hand. Loaders that work on roads or in yards with soft floors should have caterpillar treads.

Portable elevators are made by manufacturers [9], [36], [50] (see p. 23–81), and others. These machines in modified form excavate earth, dig trenches for pipe lines, remove snow from city streets. A development by Barber-Greene is a portable road building machine that travels under its own power, picks up sand and gravel, mixes it with bituminous binder, spreads, tamps and levels it to make a finished road at the rate of 1 mi. per hr.

**VIBRATING CONVEYORS,** mounted on roller races for use in mines are described below. Similar conveyors with the trough mounted on spring legs (Link-Belt Co.) are used for sugar and other materials not easily handled by screw, flight or apron conveyors.

Vibrating pipe conveyors made by Traylor Vibrator Co., Denver, Colo., are used to convey roasted ores, sintered concentrates and other hot, dusty and gritty materials. The pipes are 10 to 26 in. diam., hung by flexible suspension from above. Vibrating units are clamped to the pipe at intervals, each consisting of an armature and a stator supplied with alternating current. At 60 cycles per sec. the armature is attracted 3600 times per min. against the flexure of powerful bar springs. The inertia of heavy weights attached to the frame causes the pipe and armature to vibrate in unison. The plane of vibration is inclined to the axis of the pipe, and the material receives a component of motion, which moves it through the pipe. See A. F. Beasley, *Engg. and Min. Jour.*, March 24, 1930.

**The Redler Conveyor** (Stephens-Adamson Mfg. Co.) consists of a flexible chain or wire rope carrying pusher bars or links at short intervals. These move in a closed box, imbedded in the material conveyed. By a proper arrangement of turns, horizontal and vertical runs can be combined, and the machine acts both as elevator and conveyor. Moving parts do not completely fill the cross-section of the box and the conveyor is said to be free from danger of overloading.

## 5. COAL CONVEYING IN THE MINE

Coal conveying in a mine starts at the working face. Coal which has been shot down can be loaded by hand on a low portable scraper, shaker or belt conveyor, discharging to a longer conveyor or a mine car. Fig. 25 shows a scraper conveyor, comprising a drive section with motor and gearing, a foot section with take-up, and intermediate cross-sections 6 ft. long, bolted together. Wide link chain runs in a trough 14 in. high, Fig. 26. Belt conveyors are similar in cross-section. Shaker conveyors, Fig. 27, have troughs in sections bolted together. Each section rests on a pair of rollers in curved pieces of track, shaped to give a jumping motion to the coal when the assembled length is reciprocated by a crank at the driving end. Speed and throw of crank are varied to suit length of travel, slope, and kind of coal.

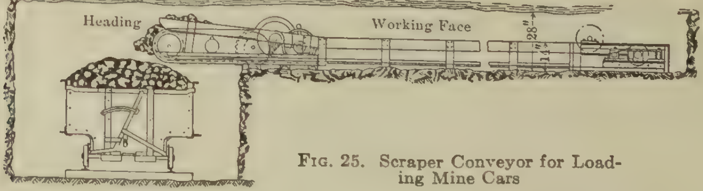


Fig. 25. Scraper Conveyor for Loading Mine Cars

Another type to make loading of mine cars easier is a chain conveyor with a low end, an inclined portion, and a horizontal top portion extending over one or more mine cars, all mounted on wheels. Other machines, to displace hand shoveling altogether are made by Jeffrey Mfg. Co., Columbus, O. A shovel machine (Myers-Whaley Co., Knoxville, Tenn.) thrusts a scoop under the loose coal, lifts it and tips it backwards into a conveyor discharging into the mine car. A loader made by Joy Machine Co., Pittsburgh, Pa., has pickpointed arms which dig into the coal and pull it into path of the conveyor chain. Several machines with wire rope drum and a scraper that drags the coal from the working face are made by manufacturers [34], [42], [86] (see p. 23–81).

U. S. Bureau of Mines, *Bull.* 343, Dec., 1930, describes and illustrates these machines. For English machines, see *Iron and Coal Trade Rev.*, No. 123, 1931. For German machines, see *Fördertechnik*, Jan. 16, March 13, May 8, July 3 & 31, Oct. 23, 1931. For a flexible conveyor for long face mining, see *Coal Age*, May 7, 1925.

**COAL CONVEYING BY MINE CAR HAULAGE.**—Robert Peele (Mining Engineers' Handbook) says: Locomotive haulage is justified at mines with easy track grades and large output. Compared with haulage by animals, the high speed, large tonnage per trip, and few men required, may balance the larger interest and depreciation charges. Compared with rope haulage, especially if there are branches or many curves, it is more flexible and easier to extend or maintain. It requires less head-room, heavier rails and easier curves than animal haulage, but cannot compete with it on thick seams in daily output of less than about 175 net ton-miles on grades below 1.25%, nor with rope haulage on grades averaging 4% against or 5% with loads.

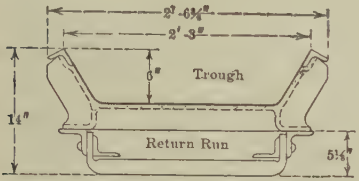


FIG. 26. Cross-section of Scraper Conveyor

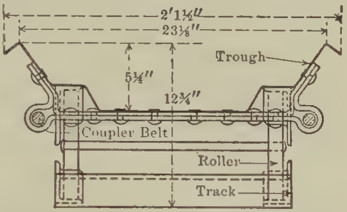


FIG. 27. Cross-section of Shaker Conveyor

**Gasoline Locomotives** have the flexibility of steam locomotives in radius of travel, and for mining work are geared for speeds of 3 to 12 miles per hr. They have no open fire, consume no fuel when idle, and are less dangerous than steam locomotives in gaseous mines. Exhaust gases and cost of gasoline are factors to consider.

**Compressed-air Locomotives** are flexible in operation within the radius of travel from charging stations for which they are designed, require no fire, do not spark nor emit noxious exhaust, but aid ventilation and are the safest type for gaseous, dry-timbered or poorly ventilated workings.

**Electric Storage-battery Locomotives** are flexible, cheaper to install than trolley or air locomotives, simple to operate and safer than any other type except compressed-air. They are suitable for short hauls, in temporary workings, as in gathering, or for infrequent long hauls where trolley lines are not warranted or where low-roofed wet workings make them dangerous. Grades must be easy.

**Electric Trolley Locomotives** are the prevalent types in mines. They are compact, simple to operate at high efficiency and usually have lowest power cost of all types. Disadvantages are: Cost and danger of trolley circuit; danger of sparking at commutator or trolley wheel; necessity of wired and bonded track.

**Rope Haulage** conditions are large tonnage and grades which make locomotive haulage inefficient. Straight haulways and uniform grades are desirable but not necessary. See Rope Haulage, p, 23–13.

**UNDERGROUND CONVEYING BY BELT CONVEYOR.**—A belt conveyor system may handle large quantities of materials in mines at a lower cost than any system of haulage. At the Colonial mines of H. C. Frick Coke Co., near Brownsville, Pa., trains of cars in the mine are turned over in a rotary dumper; the coal falls into a bin and is fed by feeders to a 60-in. belt which delivers to a series of 19 48-in. belts run at 500 ft. per min., and of lengths from 321 to 2439 ft. The total run is 4 $1/2$ miles; the total lift is 357 ft.

The system was planned and installed after the estimate had shown a probable saving of 3 cts. per ton, 10,000 tons a day, based on the following comparisons between the belt system and haulage by electric locomotive: 1. First cost, belts 25% less. 2. Depreciation, belts 15% more. 3. Cost of power, belts 100% more. 4. Cost of labor, belts, 50% less. The belt system was started in 1924. Up to July, 1932, it had carried over 25,000,000 tons of coal, and 8 of the original belts were still in use. Records and tests have shown the cost of power for the belt system to be less than $1/2$ of that estimated, the saving in labor to be greater than estimated and saving in cost of conveying to be over 3 cts. per ton. For a description of this system, see *Coal Age*, Dec. 25, 1924; F. V. Hetzel, Belt Conveyors and Belt Elevators, p. 106. A belt conveyor similar to the above is used at the Palmer mines of H. C. Frick Coke Co. It has 12 belts with a total length of travel of over 3 miles.

For uses of belt conveyors in and about mines, see catalogs of manufacturers [50], [77], [85] (see p. 23–81).

**RETARDING CONVEYORS.**—In parts of West Virginia and Kentucky, coal veins are on the hills and railroad tracks in the valley. An old way to get coal down is in coupled mine cars lowered down the slope by a tail rope. A broken car coupling or rope may cause a runaway resulting in great damage or loss of life. Coal sometimes is lowered in a large skip (monitor or gunboat) but unless it is shipped as run-of-mine, breakage due to filling and emptying the skip is objectionable. Wire rope tramways have been used for capacities of a few hundred tons per day (see p. 23–14). For larger capacities, retarding conveyors are used.

**The Wire Rope Retarder**, Fig. 28 (Jeffrey Mfg. Co., Columbus, O.) comprises an endless wire rope to which 12 in. diam. cast-iron discs are clamped at intervals of 3 or 4 ft. On the carrying (lower) run the discs travel in a V-shaped trough with rounded bottom. On the return they run in a small guide trough. The rope, $^7/_8$ to $1\,^1/_8$-in., generally is made with 6 flattened strands (see p. 6–40) arranged alternately, regular lay and lang lay so that the rope does not twist or rotate. The ends are joined in a split clamp disc. Rope retarders can be used on any slope up to 35 deg. and up to 1000 ft. long. The best slope for soft coal is about 26 deg. For slopes over 30 deg., hinged baffle-boards over the trough stop lumps if they start to avalanche. A straight run is preferable, but if slope must change from steep to flat, a large radius curve (1000 ft.) should be made in the trough, or one conveyor must be made to discharge into another. As the wire rope is submerged in coal, wheels or tracks cannot be used to hold the rope down. At terminals, the rope runs over toothed wheels. On all but flat slopes, brakes prevent the conveyor drifting when power is off. On steep slopes a hand-operated brake also is used. Rope retarders have been used for 50 years or more. When properly designed and built they are good and relatively inexpensive. Carrying capacity is 100 tons to 200 tons per hr.

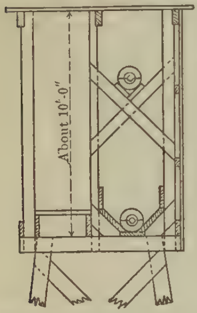


Fig. 28. Wire Rope Retarder

**Chain and Flight Retarding Conveyors** consist of two strands of steel roller chain attached to ends of steel flights, running in a steel trough. The chains roll on tracks and the path of the conveyor can be made to suit the slope and contour of the ground. Fig. 21 shows a cross-section. A 36 × 12 in. flight will carry 360 tons per hr. at 100 ft. per min. Some retarders have 48 × 12-in. or 60 × 16-in. flights for large carrying capacity. Fig. 29 shows the upper end of a retarding conveyor where a shaking screen separates the fines before the coal goes to the picking bands. With such an arrangement, refuse is thrown out on the slope of the hill.

**CLEANING AND SIZING OF BITUMINOUS COAL.**—The handling of bituminous coal for cleaning and sizing in one modern tipple may be described as follows: The tipple building is a steel structure spanning 5 railroad, and 2 larry tracks, that run to coke ovens. Coal runs from lower end of retarding conveyor over a bar screen which removes slack, then over a double deck shaking screen which takes out more slack and separates the rest into lump, egg and nut. These sizes discharge separately to three steel apron conveyors which are placed parallel to the railroad tracks and located directly over them. For about 30 ft. the apron conveyors are horizontal. Men on each side pick out refuse. For the remainder of their length, conveyors are on hinged booms, which by electric hoists, can be lowered or raised so that coal is loaded into railroad cars without much breakage. When booms are horizontal, coal can be discharged into mixing chutes, arranged so that lump and egg, for instance, or egg and nut, can be loaded into cars. Refuse picked out of the coal is dropped into vertical chutes, whence it passes into conveyors under tipple floor and into a car for disposal. Everything below nut size from shakers and bar-screen is conveyed to a screen house over the larry tracks. A rotary screen takes out the pea size and drops it into a bin for shipment by car or truck. Slack falls into a bin, and either is taken in larrys to coke ovens or conveyed across the tipple for mixing with any of the larger sizes on the railroad loading track. Fig. 30 shows the arrangement of loading boom and picking table in a small tipple.

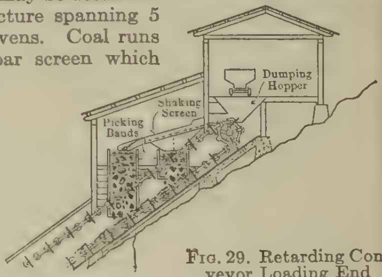


Fig. 29. Retarding Conveyor Loading End

The Knox-GON swiveling loading boom (Link Belt Co.) has a frame pivoted at the receiving end, and is placed over the center line between two tracks. It can be swung to serve gondolas on either track or to discharge to a Manierre type loader for loading box cars on either track. This machine (Manierre Engg. Co., Milwaukee) has a short belt or apron conveyor mounted on a hinged and jointed boom adjustable to various angles. It is capable of being moved into working position through a car door, or swung out to give car clearance. It will handle any kind and size of lump or fine material and deposit it in either end of a box car with a minimum of breakage.

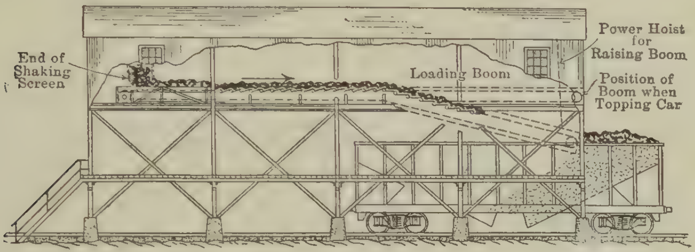

End of Shaking Screen
Loading Boom
Power Hoist for Raising Boom
Position of Boom when Topping Car

FIG. 30. Picking Table and Loading Boom in Coal Tipple

The rescreening loading boom (Link Belt Co.) has two chains with cross bars running in a trough with screen bars in the bottom. Its advantages are lighter construction, shorter boom and less drop of coal at delivery end.

# BELT CONVEYORS

For fundamentals of design and operation, see F. V. Hetzel, Belt Conveyors and Belt Elevators (John Wiley & Sons). In its simplest form a belt conveyor comprises two end pulleys, a moving belt, and idlers to support the belt. The lower idlers have flat-faced rolls; the upper or carrying idlers generally are arranged to curve or trough the belt to make it carry more and avoid risk of spill. There is not much difference between 3- and 5-roll troughing idlers in carrying capacity. Three-roll idlers cost less, have fewer bearings, lubrication is more certain. The best troughing idlers have ball or roller bearings; the saving of power, reduced cost of maintenance and longer life of belt often pay for the extra cost of the improved idlers. On grain conveyor belts, the best practice is to run the belt nearly flat with concentrator or "tuck-up" idlers at intervals to give a slight troughing.

Troughing idlers are spaced 4 to 5 $1/2$ ft. apart on narrow belts, according to weight of material carried; on wide belts, 3 to 4 ft. apart. Return idlers are spaced 9 or 10 ft. apart.

For detailed information about belt idlers and other parts, see catalogs No. 1615, Link-Belt Co., Chicago; No. 82, Robins Conveying Belt Co., New York; No. 30, Stephens-Adamson Mfg. Co., Aurora, Ill., and other manufacturers of conveying machinery.

**THE CONVEYOR BELT** may be of solid woven fabric, or of cotton duck with the plies stitched together. Both kinds are impregnated with oils or compounds to resist moisture. Rubber belts, however, are in more general use. They are built up with layers of cotton duck cemented together by rubber compounds, and then vulcanized. Lengthwise threads of the cotton duck are heavier than crosswise to give tensile strength with flexibility for troughing. An extra thickness or cover of rubber on carrying side of the belt resists abrasion and increases its life. For crushed coal, the cover may be $1/16$ in. thick, but for run-of-mine coal and abrasive ores a thicker cover is required to avoid puncture by sharp pieces, and to resist wear occasioned by making material at the loading point acquire the belt velocity.

**CARRYING CAPACITIES OF BELT CONVEYORS** in tons of coal per hour for troughed belts are given in Table 1, which is based on formula, cu. ft. per hr. at 100 ft. per min. = $3.2W^2$, where $W$ = width of belt, in. Belts carefully loaded with run-of-mine coal can run 500 ft. per min.; higher speeds in Table 1 refer to crushed coal. If material is not too lumpy and if feed is steady and continuous, belts wider than 18 in. can be loaded to a greater capacity, *viz.*, 24 in., 5% more; 30 in., 10%; 48 in., 16%; 60 in. 20%.

On narrow belts, a relatively greater percentage of lumps comes in contact with the belt surface in loading. Wear from abrasion in loading is proportional to speed of belt. Hence, it is injurious to run narrow belts at high speeds. At the same time, part of the efficiency of the belt as a conveyor is due to the ability to run faster than a chain conveyor, whose speed seldom exceeds 200 ft. per min. At speeds much lower than that a belt is not so economical in handling bulk materials, although it may be preferred for other reasons. Width of the belt depends on lumpiness of material carried. Table 2 gives values ordinarily considered safe and proper.

**HORSEPOWER OF BELT CONVEYORS.**—A formula based on tests of belt conveyors (Goodyear Rubber Co., 1930) is in three parts.

$$\text{Hp. of empty conveyors} = C(L + L_0)(0.03\,QS)/990 \quad [1]$$

$$\text{Hp. to carry tonnage on a level} = C(L + L_0)T/990. \quad [2]$$

$$\text{Hp. to raise tonnage} = +(TH/990)\text{; to lower tonnage} = -\ (TH/990) \quad [3]$$

The algebraic sum of the three items gives horsepower at head pulley; $C$ = a friction factor depending on kind of idlers used, $=0.05$ for plain bearings, grease lubricated; $=0.03$ for ordinary ball or roller bearings, grease lubricated; $=0.022$ for high-grade ball or roller-bearings, pressure lubricated; $L$ = length of conveyor, ft., center to center; $L_0$ = arbitrary length factor, depending on grade of equipment used, $=100$ where $C = 0.05$; $=150$ where $C = 0.03$; $=200$ where $C = 0.022$. $Q$ = (weight of moving parts of idlers + weight of belt reduced to lb. per ft. of center to center length of conveyors), (see Table 3); $S$ = belt speed, ft. per min.; $T$ = tons of material per hr.; $H$ = height, ft., through which material is raised or lowered.

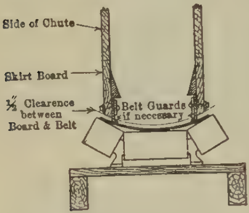


FIG. 1. Skirt Board Construction

If a tripper is used, add 1 Hp. for narrow belts, and up to 3 Hp. for wide belts. To determine size of motor, add 3% to 5% for each belt drive, chain drive or gear reduction.

**BELT THICKNESS.**—In general, for durability and long life, tension in a rubber or cotton belt should not exceed 24 lb. per in. of width per ply of thickness, if duck is the usual 28-oz. duck. In a belt conveyor, this 24 lb. cannot all be used to move the loaded belt. Some of it must be used to supply enough tension to prevent slip on the drive pulley. This tension is applied by the take-up. The smaller the arc of wrap on the drive pulley and the lower the coefficient of friction between belt and pulley, the greater must be the tension. By increasing angle of wrap and lagging drive pulley, more of the strength of the belt can be usefully employed.

After calculating horsepower by method given above, the number of plies of belt to move the loaded conveyor may be found by

$$N = \text{number of plies} = \text{Hp.} \times 33{,}000/SWP, \quad [4]$$

where $S$ = belt speed, ft. per min.; $W$ = belt width, in.; $P$ = useful net tension per inch of width per ply of thickness. The relation between belt tension, belt velocity and horsepower is given on p. 24–14. The value of the coefficient of friction $f$ in the fundamental formula there given may be taken as $f = 0.30$ for conveyor belt on bare iron pulleys, and $f = 0.34$ for belt on lagged pulleys. Values of $P$ may be taken as follows: 180° wrap on iron pulley, $P$ = 14 lb.; 180° wrap on lagged pulley, $P$ = 16 lb.; 240° wrap on lagged pulley, $P$ = 18 lb.; 420° wrap on 2 lagged pulleys geared together, $P$ = 24 lb.

Table 1.—Tons (2000 Lb.) of Coal per Hour at Various Belt Speeds

| Width of Belt, in. $W$ | Belt Speed, ft. per min. = $S$ | | Width of Belt, in. $W$ | Belt Speed, ft. per min. = $S$ | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 300 | 350 | | 300 | 350 | 400 | 450 | 500 | 550 | 600 |
| 12 | 34 | ... | 24 | 139 | 162 | 184 | ..... | ..... | ..... | ..... |
| 14 | 49 | ... | 30 | 216 | 252 | 288 | 324 | ..... | ..... | ..... |
| 16 | 62 | ... | 36 | 311 | 363 | 414 | 466 | 518 | ..... | ..... |
| 18 | 80 | 92 | 42 | 423 | 493 | 564 | 635 | 705 | 775 | ..... |
| 20 | 96 | 112 | 48 | 552 | 644 | 737 | 830 | 920 | 1013 | 1105 |

Table 2.—Maximum Size of Material to Give Capacities in Table 1

| Size of Material, in. | Belt Width, in. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 14 | 16 | 18 | 20 | 24 | 30 | 36 | 42 | 48 |
| 90% of mixture not over | 2 | 2 1/2 | 3 | 3 1/2 | 4 1/2 | 6 | 7 1/2 | 9 | 10 1/2 |
| 10% " " " " | 3 | 4 | 5 | 6 | 8 | 11 | 14 | 17 | 20 |

Table 3.—Values of Factor $Q$ in Belt Conveyor Horsepower Formula

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Width, in. .......... | 12 | 14 | 16 | 18 | 20 | 24 | 30 | 36 | 42 | 48 | 54 | 60 |
| $Q$ ................ | 15 | 16 | 17.5 | 19 | 21 | 25 | 32 | 40 | 49 | 59 | 69 | 81 |

If the belt is made of 32-oz. duck, values of $P$ may be multiplied by 1.125. Horsepower may be calculated from the fundamental formula for belt tension, *viz.*, Hp. $= PV/33{,}000$, where $P$ = net effective tension in belt $= (T_1 - T_2)$, the tension in the tight and slack sides, respectively; $V$ = velocity, ft. per min.

Values of $N$ obtained from the formula should be compared with tables of maximum and minimum thicknesses of belts as given in handbooks of belt makers. If the belt is too thin it will deflect too much; if too thick, it will not trough properly.

Rubber belts are made in different grades, differing as to weight of duck, quality of "friction" used to cement plies together, and thickness and quality of rubber covers. For detailed information, see F. V. Hetzel, Belt Conveyors and Belt Elevators, pp. 41, 110. Goodyear's 1930 Handbook gives considerable detailed information concerning belts, tandem drives, stresses in inclined conveyors, etc. See also Goodrich's Manual of Belt Practice, No. 2800.

**DIAMETER OF BELT PULLEYS.**—The larger the pulley, the less the stretching of the rubber "friction," or of the stitching which holds the plies of the belt together, and the longer the belt will last. The diameter of driving pulleys in inches should be at least five times the number of plies in the belt. The diameter of foot pulleys and snub pulleys should be four times the number of plies, and of tandem pulleys, geared together, driving thick belts with heavy rubber covers, at least six times the number of plies.

**LOADING BELTS.**—To reduce the wear on the belt surface, the material should be fed by a chute of such length and slope as will give to the material the velocity of the belt, if possible. To prevent scatter, and to pile the load properly, skirt-boards 5 to 7 ft. long are used. They must be carefully designed. If not well supported they may sag and cut the belt. Fig. 1 shows the usual construction in wood.

**DISCHARGE FROM BELT CONVEYORS.**—The simplest belt conveyor discharges over the head-pulley, as in Fig. 2. For intermediate discharge at one or more points, fixed trippers may be used, as in Fig. 3, but they double or triple the wear on the belt surface whenever they feed material back on the belt, and by repeated reverse bending over pulleys they tend to separate the plies of the belt. For most conditions of intermediate discharge, a traveling or movable tripper, Fig. 4, is preferable. Some materials, like foundry sand or wood chips, can be carried on flat belts, and at slow speeds the discharge can be divided at several intermediate points by "strike-off" boards set at various positions above the belt, or all can be thrown off at any point if the boards are hinged so as to swing into and out of position. This method is not good with lumpy material.

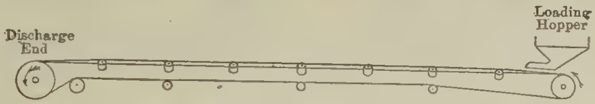


FIG. 2. Belt Conveyor Discharging Over Head Pulley

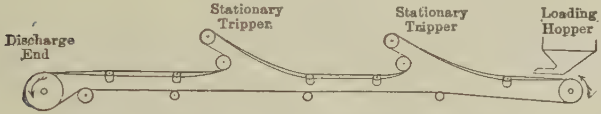


FIG. 3. Belt Conveyor with Stationary Trippers

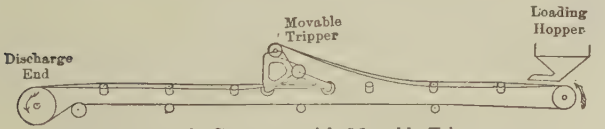


FIG. 4. Belt Conveyor with Movable Trippers

**INCLINED BELTS.**—With uniform feed, fine coal may be carried on a belt at an angle from the horizontal of 22 deg.; but when mixed with lumps, at an angle of not over 20 deg. Run-of-mine coal, coal and coke can be carried at an angle of not over 18 deg., crushed stone and ore, at not over 18 deg., and tempered foundry sand at not over 24 deg. On the other hand, numerous substances under certain conditions can be carried at not over 15 deg.

**ADVANTAGES OF BELT CONVEYOR.**—1. Low power consumption. 2. Noiseless operation. 3. Adaptable to all sorts of material. 4. Suitable for high capacities and long distances beyond the range of other conveyors. 5. Simplicity of construction.

**LIMITATIONS OF BELT CONVEYOR.**—Some comparative disadvantages are stated on page 23–61. The belt is the expensive item and must be protected from the following sources of injury: 1. Poor arrangement of feed. 2. Chute or skirt-board sagging on belt. 3. Lumps falling on return belt and jamming between foot pulley and belt. 4. Belt running crooked and fraying the edges. 5. Oil or grease getting on rubber belt. 6. Continued exposure to great heat or cold, or direct sunlight.

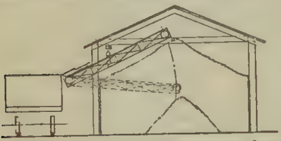


FIG. 5. Portable Conveyor for Car Unloading

When belts are inclined, especially at or near the maximum angle, there is a shift or slip of material in passing over each troughing idler. This adds to the wear on the carrying surface, and with gritty materials it may be serious. Short belts, steeply inclined, are apt to be short-lived under such conditions, because the wear from the pick-up at the loading point is already large, due to the fact that each foot of belt passes the loading point frequently. In long belts the wear from pick-up is distributed over a greater amount of belt surface and the bending over pulleys does not occur so often. Hence long belts last proportionately longer than short belts.

**STANDARD BELT CONVEYORS,** 16 to 48 in. wide for various lengths and capacities are shown in catalogs of manufacturers [44], [50], [85] (see p. 23–81), and other makers of conveying machinery. For many conditions of ordinary service, time and expense can be saved by buying a conveyor from a catalog rather than designing one.

**Sectional Belt Conveyors** 18 and 24 in. wide (Barber-Greene Co.) consist of standard drive and take-up groups with machinery parts mounted on steel frames. Conveyor length is made up of light steel frames of various standard lengths, which, with belts, idlers and guards weigh not more than 60 lb. per ft. They will span 25 to 50 ft. in the clear.

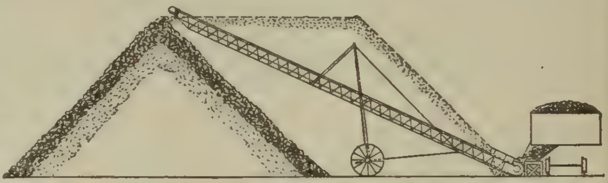


FIG. 6. Portable Conveyor for Piling

**Portable Conveyors** of the same light construction, listed by the same makers, are used to receive concrete from mixers and deliver it to forms in building construction. By having one conveyor discharge to the next, concrete can be distributed over large areas. In some cases, 15 conveyors have been used for a distance of 500 ft. or more. The 24-in. belt runs over 2-roll idlers to form a deep trough. A flexible scraper under each head pulley removes concrete from the belt. Frames are made up to 50 ft. long and are mounted on large caster wheels. For handling concrete on belt conveyors, see *Trans.* A.S.M.E., MH–52–10, 1930.

Other portable belt conveyors are made by the companies above named and also by manufacturers [33], [72], [77], [82] (see p. 23–81). Fig. 5 shows one which stores in a shed, material shoveled over side of a car. The frame is hung from a chain block and trolley on an overhead track that runs the length of the shed. Motor- or gasoline-driven conveyors on wheeled frames, Fig. 6, with 12-, 14- or 16-in. belts on frames 20 to 40 ft. long, will receive material from bottom-dump or side-dump cars and form storage piles or deliver to bins or wagons. Several machines can be placed in tandem to deliver to a distance, or a high pile can be formed by arranging machines as in Fig. 7. These machines are light, a 16 in. × 30 ft. size weighing less than a ton. Heavy machines of this type are made by Barber-Greene Co., with 18- or 24-in. belts on frames up to 60 ft. long, with a hand-power hoist for adjusting height of boom.

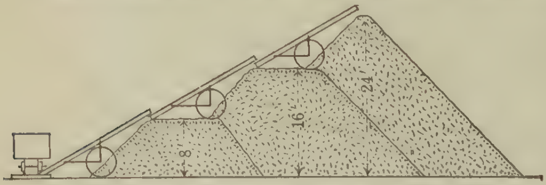


FIG. 7. Portable Conveyors Arranged for High Piling

**CAR LOADER BELTS.**—Short belt conveyors mounted on wheeled frames easily handled by one man, and run at 1200 ft. per min. or more will throw fine material a distance of 15 to 20 ft. The machine is placed in position at door of a box car, receives material from a chute, throws it into a pile at each end of the car and distributes it for proper loading. It saves the labor of several men. Loaders of this type are built by manufacturers [9], [50], [85] (see p. 23–81).

**CAR UNLOADER BELTS** (Barber-Greene) are mounted on a shallow steel frame, 4 or 5 in. deep, that can be laid across a track above the rails, or in a shallow pit below them, to receive discharge from a drop-bottom car. They are used for free-flowing materials without lumps. The service is severe, but the car can be unloaded anywhere on the track and a deep pit is unnecessary.

**WEIGHING ON BELT CONVEYORS.**—The Merrick Weightometer (Merrick Scale Mfg. Co., Passaic, N. J.) for weighing material in transit consists of: 1. A part of the upper run of the belt suspended from counterbalanced weighing levers. 2. A totalizer driven by the moving belt which continuously multiplies weight by travel to give tonnage passing over the conveyor. This is recorded on a counting machine or chart recorder.

**STEEL BELTS.**—The Sandvik belt (Sandvik Steel, Inc., New York) is a tempered, polished, cold-rolled steel band, about $^1/_{32}$ in. thick, made up to 32 in. wide and up to 375 ft. long. It runs flat over rollers, spaced 6 to 12 ft., or slides on wooden strips. These belts handle clinker, phosphate rock, sugar, clay, foundry sand and similar wet, sticky mixtures that do not discharge clean from fabric or rubber belts. They discharge either over the head pulley or by scrapers over the belts. The scraper is simple, cheap, and works well with materials that cannot be discharged cleanly by a belt tripper. The belts are light, run quietly, and when supported on rollers are said to take less power than rubber belts on troughing idlers. Belts wider than 32 in. can be made by a method of longitudinal jointing.

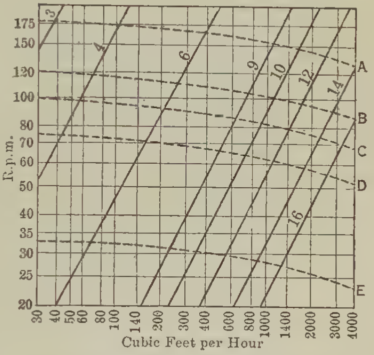


Fig. 1. Capacities of Screw Conveyors

**STEEL MESH BELTS** are used for conveying through acid or alkaline baths, chemical solutions, and scalding tanks, and in processes of drying, baking, annealing, spraying, etc., for which fabric belts would be unsuited. They are made in the U. S. by Cambridge Wire Cloth Co., Cambridge, Md., and Wickwire-Spencer Steel Co., New York. They consist of woven wire mesh, with or without edge reinforcement, or of coils of wire interlocked and flattened. Sometimes the coils are joined over wire rods which may be bent at the ends to form a protective edge or border. These belts can be made of copper, brass or the various alloys which impart strength or resist heat, corrosion, or the action of chemicals. They can be made with steel plate selvages or partitions that will bend around driving drums. Belts can be woven up to 12 ft. wide. Those made for lehrs and annealing ovens will work regularly in temperatures over 1000° F.

**LONG DISTANCE CONVEYING BY BELT.**—See p. 23–20 and p. 23–59. For a recent installation (Jeffrey Mfg. Co.) of long, wide belts of large carrying capacity, see *Engg. News-Record*, Aug. 1, 1935.

## SCREW CONVEYORS

Screw conveyors consist of a helical blade of metal fastened to a central pipe or shaft, and set in a trough or box. The blade generally is of steel plate, but depending on service and nature of material handled, it can be cast iron, cast steel or an alloy that resists corrosion and oxidation. The trough generally is U- or V-shaped with a rounded bottom. Some materials can be conveyed through a round pipe if the length is short enough to permit the screw conveyor bearings to be kept outside the pipe.

Bearings are of various kinds, babbitt, bronze, chilled cast iron, hardwood. Success with screw conveyors often depends on using the proper kind of hanger bearings and the right kind of trough. It is advisable to consult the makers on these points.

**CONVEYING CAPACITIES.**—Fig. 1 shows curves developed by Link-Belt Co., Chicago, for capacities of screw conveyors. Diagonals 3, 4, 6, etc., are diameters of screw, in. Curves *A*, *B*, *C*, etc., are advisable speed limits, r.p.m., but rating should be multiplied by a capacity factor *X*, given on p.23–66.

**Class 1.**—Dry, free-flowing, not gritty; as grain, malt, flour, meal, pulverized coal; advisable speeds not beyond curve *A*; $X = 1.00$.

**Class 2.**—Shelled corn, clean cotton seed, fine coal, crushed ice, pulverized minerals that are not abrasive; advisable speeds not beyond curve *B*.; $X = 0.75$.

**Class 3.**—Wet malt, starch, sawdust, ground cork, carbon black, dry paper pulp stock, non-abrasive minerals, with occasional lumps (not over 25%) of size not over $^1/_8$ diameter of screw; advisable speeds not beyond curve *C*; $X = 0.66$ to $0.75$.

**Class 4.**—Damp, trashy cotton seed, dry clay, raw sugar, foundry sand, fertilizers, crushed gypsum, lime, and other abrasive materials with lumps (not over 50%) of size not over $^1/_6$ diameter of screw; advisable speeds not beyond curve *D*; $X = 0.50$ to $0.66$.

**Class 5.**—Very gritty materials, as ashes, wet or dry, silica sand, pulverized flint or quartz, flue dust, fly ash; advisable speeds not beyond curve *E*; $X = 0.30$ or less.

The capacity factor *X* refers to screw conveyors with intermediate hanger bearings. If conveyor is short, has no hanger bearings, and is properly fed, capacity may be 50% greater than use of the factor would indicate.

Screw conveyors at slopes up to 10° or 15° have about the same capacity as level conveyors. At greater slopes, capacity tends to fall off, but for such conditions, screws with short-pitch flights can be used. Screws at steeper angles should be referred to the makers. Vertical screws, run at the proper speed and properly fed, will elevate ground cork and some other light materials in closed pipes. In "batching" plants that serve concrete mixers, Link-Belt vertical screws have been used to elevate cement 20 or 30 ft. They require more power than a bucket elevator and are apt to get out of order from cement working into the step bearing. The advantage is that they are light and portable.

**HORSEPOWER OF SCREW CONVEYORS.**—If a screw conveyor is crooked and out of line, if speed is excessive, if hangers and bearings are not suited to the work, if discharge outlets choke, then the power consumed cannot be estimated. For screws working under proper conditions, Link-Belt Co. gives the following general formula:

$$\text{Hp.} = 1 + (ALN + CWLF)/1{,}000{,}000 \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where $L$ = length of conveyor, ft.; $N$ = r.p.m. of screw; $C$ = cu. ft. conveyed per hr.; $W$ = weight of material, lb. per cu. ft.; $A$ and $F$ are factors. For conveyors with lubricated bearings, values of $A$ are as follows:

| Diam. of screw, in. | 3 | 4 | 6 | 9 | 10 | 12 | 14 | 16 |
|---|---|---|---|---|---|---|---|---|
| *A* | 15 | 21 | 33 | 54 | 66 | 96 | 135 | 186 |

For bearings not lubricated, multiply *A* by 1.75; for chilled iron bearings, multiply *A* by 2.5.

Factor *F* varies with kind of material (see classification under Conveying Capacities), and within a given classification with the resistance material offers to moving in the trough and past hanger bearings. Values of *F* are:

| Material Class | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| *F* | 0.4–0.6 | 0.4–0.9 | 0.6–1.2 | 1.4–2.0 | 2.0–5.0 |

For specific values of *F*, see Link-Belt Co. 1933 catalog.

**SCREW CONVEYOR FEEDERS.**—For withdrawing material from a bin at a definite and regular rate a screw has the following advantages over other feeders: 1. Compact, leak proof, dust tight. 2. Relatively cheap in first cost and operation. 3. Can be fitted to bin bottoms of any shape. 4. Acts as a shut-off valve as well as a feeder.

Fig. 2 shows a short feeder of diameter *A* in a closed pipe for the length *D*. The pipe prevents material from flooding out of the bin and acts as a shut-off when screw is idle. Screw is tapered over length *C*, which is attached to the bin, and the trough is flared. Diameter *B* is one size smaller than *A*. The screws are made with heavy steel blades on heavy pipe. Table 1 gives dimensions of the three sizes used for feeders. The carrying capacity for fine material, or material containing not more than 25% of lumps, is about 0.02 cu. ft. per revolution for 9-in.; 0.04 cu. ft. for 12-in.; and 1.1 cu. ft. for 16-in. screws.

For longer feed screws, where an intermediate bearing must be used, sizes are either 9 in. or 12 in. A U-shaped trough is used beyond the length *D*, Fig. 2.

**Table 1.—Dimensions of Screw Conveyor Feeder**

| Size of Conveyor, in. | Max. Size of Lumps, in. | Dimensions, in. (See Fig. 2) | | | |
|---|---|---|---|---|---|
| | | *A* | *B* | *C* | *D* |
| 9XX | 1 1/2 | 9 | 6 | 24 | 24 |
| 12XXX | 2 | 12 | 9 | 36 | 36 |
| 16XXX | 2 1/2 | 16 | 12 | 48 | 48 |

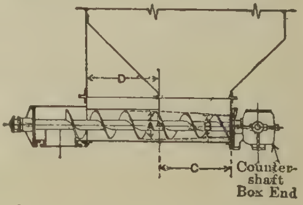


Fig. 2. Screw Conveyor Feeder

It contains the hanger bearing and a screw of full diameter. Clearance between screw and inside of trough should be 3 to 4 in. Length $D + C$ has a screw of a reduced diameter $B$. The rated capacity is the same as for short feed screws, but the lumps may be 50% of total volume.

For cement, pulverized coal and materials which at times flow like fluids, screw feeders are inclined upward at 20 or 30°, with outlet of the trough higher than the intake point. Screws generally have double flights.

**THE THRUST IN SCREW CONVEYORS** should be taken on end castings. These are of various kinds, with ball or roller thrust bearings for heavy loads. Driving ends with bevel gears are general. The Link-Belt Co. places the motor worm gearing and oil-tight gear box on a base plate; gear box is bolted to end of the trough.

**TROUGHS** are well standardized. For cement and similar dusty materials no attempt is made to bolt on a dust tight cover. The cover plate is flanged downward all around and rests in grooves formed by Z-bars on top edge of trough. Dust collecting in the grooves seals cover against further dust emission.

Screw conveyors originally were used for conveying grain but their use has been greatly extended. With properly designed screws, troughs, bearings and drive ends, they handle a wide variety of materials, some of which cannot be conveyed by any other kind of conveyor. For details of screw conveyors and accessories, see catalog of Link-Belt Co., and other makers of conveying machinery.

## PNEUMATIC AND HYDRAULIC CONVEYING

**PNEUMATIC TUBE CONVEYORS** used for transmitting small carriers between departments of shops, newspaper offices, industrial plants, etc., are made by manufacturers [32], [46], [65] (see p. 23–81), and others. The conveyor tube is round or oval, the carriers are short, of a cross-section to fit the tube, and are propelled at 20 or 30 ft. per sec. by vacuum or air pressure. The round tubes are $1\,^1/_4$ in. diam. or larger; the oval tubes are 3 × 6 or 4 × 7 in. Short lengths of small tube, up to 200 ft., can be served by a foot-power pump, but usually a motor-driven blower is used. For a history and description of this system, see *Trans.* A. S. M. E. 1929, MH–51–10.

**PNEUMATIC CONVEYING FOR BULK MATERIAL** was first used in England, developed chiefly in Germany, and now finds extensive use in the U. S. The suction or vacuum system, Fig. 1, comprises an exhauster or pump to produce a partial vacuum, a pipe line through which mingled air and material move, a receiver or cyclone to separate the material from the air, and usually a bag filter to clear the air and keep dust from the exhauster, and prevent it discharging to atmosphere. In a pressure system, the pump delivers air to a pipe line. Material is fed from a supply bin through an air-tight rotary valve into the pipe, moves with the air to a cyclone or separator. The air is discharged to a filter or to atmosphere. The material settles to the bottom of the cyclone.

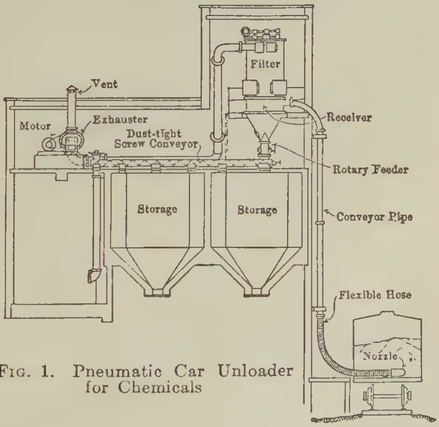


Fig. 1. Pneumatic Car Unloader for Chemicals

**Carrying Capacity** depends on amount of air pumped and difference in air pressure between the two ends of the system. Theoretically, almost any material can be conveyed, but feasibility and cost are determined by practical considerations: 1. Material too large requires large-diameter pipes and a large air pump. 2. Material too dusty may require too large a filter. 3. Material too fragile may be reduced in value by breakage at pipe turns, or dust produced by breakage may require a costly filter. 4. If air pressure is kept low enough to avoid damage to material or to avoid excessive wear at pipe turns, carrying capacity may be too low to be economical.

**The Cost of a Pneumatic System** may be higher than that of a mechanical system. Power cost may be 5 or 10 times as much, but a pneumatic system often is justified by other considerations: *A.* In boats, a flexible suction hose is more easily handled than an elevator leg, takes up less room, cleans up better, requires less shoveling, can work under closed hatches in wet weather. *B.* In unloading railroad box cars, time often is saved, labor costs reduced, delays and demurrage charges avoided. Avoidance of dust sometimes is important. *C.* In handling bulk materials in manufacturing plants, labor often is saved, less dust is produced and less space occupied. Changes in the pipe system to reach new points of intake or discharge are not difficult, and, what often is most important, conditions of safety and health for workmen are better.

A combination of pneumatic handling with mechanical conveying often is desirable. In Fig. 1 material is drawn from the cyclone separator to a distributing screw conveyor. At the New York State Barge Canal terminal in Brooklyn, grain is lifted 80 ft. vertically from barges, and discharged to a distributing belt conveyor.

Fig. 2 shows a Fuller conveyor system on a movable tower for unloading small size soft coal at a cement plant. Coal, at rate of 100 tons per hr., passes through the cyclone separator, is discharged to a short belt conveyor, then dropped on the ground to be distributed later over the storage area by a drag scraper. The tail pulley of the drag line is attached to the rear of the tower and moves with it. See Drag Scraper System, p. 23–16. The boat also carries cement in bulk from the cement plant. At destination, the cement is unloaded by a Fuller-Kinyon pump system on the boat. See below.

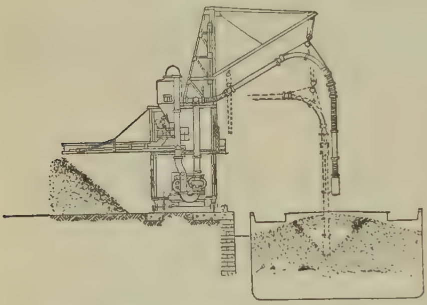


Fig. 2. Fuller Pneumatic Conveying System

Pneumatic systems for conveying bulk material are made by Dust Recovery and Conveying Co., Detroit; Brady Conveyors Corp., Chicago; Federal Pneumatic Systems, Inc., Chicago; Holly Pneumatic Systems, Inc., New York; United Conveyor Corp., Chicago; Fuller Co., Catasauqua, Pa.

**SUCTION SYSTEMS** for collecting and conveying chips and sawdust from wood working machines and dust from grinders are made by B. F. Sturtevant Co., Boston; Allington and Curtis Mfg. Co., Saginaw, Mich.; Kirk and Blum Mfg. Co., Cincinnati, and also by makers of blowers and exhausters. A Kirk and Blum installation in Camden, N. J., handles over 1,000,000 cu. ft. of air per min., collects 150 tons of sawdust daily, and has filters with an area of 200,000 sq. ft. of cloth surface. Similar systems for taking ground coal, rock, etc., from pulverizing mills are made (see p. 23–81) by manufacturers [31], [35], [38], [73], [91], and several makers of grinding mills.

For precautions against fires in blower systems, see publications of National Fire Protection Assoc., Boston. For safety codes for preventing dust explosions, see Bull. 433, U. S. Dept. of Labor, 1927.

**GRAIN UNLOADING.**—Suction apparatus for unloading grain from boats is used at many ports in Europe. The whole system, including power plant and separators, generally is mounted on a barge or float that works on the harbor side of the ship. It transfers grain to river barges while the ship discharges other cargo on the wharf. In the U. S., the largest suction system for unloading grain (1935) is the one in Brooklyn mentioned above; it lifts 6500 bushels per hr. At the public grain elevator, New Orleans, where it was necessary to provide for a 21 ft. range of water level, and for boats differing greatly in width of beam and depth of hold, a pneumatic unloader of 6000 bushels per hr. capacity has been used since 1917.

**THE FULLER-KINYON CONVEYING SYSTEM,** Fig. 3, is not strictly a pneumatic conveyor. At one end of a pipe line a short impeller screw 1 is driven by the motor 2. Material is fed through hopper 3. At discharge end of cylinder in which the screw revolves, compressed air enters through rings 4 and 5. The material when mixed with air acts as a fluid. In the fluent condition it is forced through the pipe line 6 by the displacement of material fed in by the screw and also by the expansion of the compressed air used to create the fluent condition.

Practically any dry pulverized material in quantities up to 300 tons per hr. can be transported by this system over distances of 3000 ft. and elevations of 150 ft. or more. The system is widely used in cement mills, and for other pulverized materials as lime, soda ash, starch, kaolin, bauxite, etc. The pipe line may range from 2 to 8 in. diam. according to amount of material. Branch pipes are connected to the main line by 2- or 3-way valves, usually operated by hand, but sometimes by automatic and remote control.

Bins can be filled in desired sequence; the system will continue to work automatically so long as any bin has room to receive material. Deliveries under automatic control have been made to 48 bins in one system. To mix and blend raw materials in cement making, the discharge from several pipe systems may be controlled automatically according to predetermined time intervals. Hot materials up to 1000° F. are successfully conveyed. Pipe lines may be water jacketed for cooling cement in process of manufacture. Steam jacketed lines also are in use.

Fig. 4 shows a Fuller-Kinyon portable pump, used chiefly on construction jobs to unload bulk cement from box cars or barges, to convey it to mixer bins or storage, and to reclaim it from storage to the bins. The pump is a motor-driven self-contained unit. Clutches control the power drive to the traction wheels. Cement is discharged through flexible hose which may be connected to any of the steel pipes leading to delivery point.

Several boats on the Great Lakes are equipped with the L. D. Smith system of drag scrapers (see p. 23–17) and a Fuller-Kinyon pump. Cement is pumped ashore at rate of 1400 barrels per hr. through a flexible hose to the pipe lines which distribute cement to silos or pack houses.

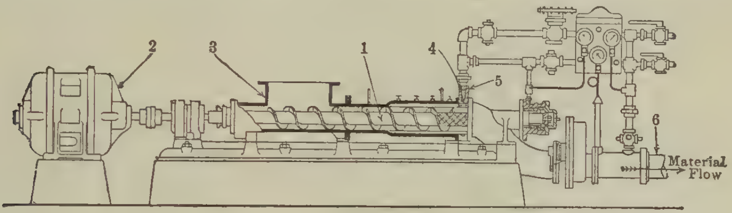


Fig. 3. Fuller-Kinyon Pump

The Fuller-Kinyon system often is cheaper and more convenient than any combination of mechanical elevators and conveyors. For short distances power consumption may be more, but for average or long distances it is apt to be less. Its advantages are: Pipe can be run anywhere over, under or around obstacles; material conveyed is always enclosed and risk of fire, explosion, or injury to workmen is low. For temporary jobs of road making or building construction, its erection, dismantling and moving are relatively cheap.

**TRANSFER OF MATERIALS UNDER DIRECT PRESSURE.**—The Fluxo pump (F. L. Smidth & Co., New York) is used in some cement plants to transfer cement or dry pulverized raw materials to and from storage bins. A carriage runs on a track at one side of a row of silos or storage bins, and connects with a pipe in the base of the silo by a flexible hose. On the carriage are two tanks, a vacuum pump with an air filter, an air compressor and an electro-pneumatic control apparatus. When the current is being drawn into one tank by suction, it is being expelled from the other by air pressure. The operation is reversed by the control apparatus, the tanks alternately filling and emptying. The flow, from silo to transport line to which the machine is connected, is continuous.

The Kennedy transport system (Kennedy-Van Saun Engg. Corp., New York) consists of a single tank, to which cement, or other pulverized materials flow, suitable valves, air under pressure and a transport pipe line. At a predetermined weight of material in the tank, springs on which the tank rests deflect, the supply is shut off, the bottom valve opens, and air admitted to top of the tank forces the material out through the pipe line. The cycle automatically repeats.

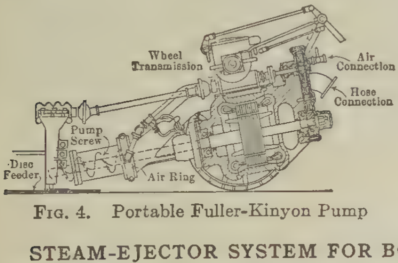


Fig. 4. Portable Fuller-Kinyon Pump

**STEAM-EJECTOR SYSTEM FOR BOILER HOUSE ASHES.**—The Steamatic ash conveyor system (United Conveyor Corp., Chicago) has in many cases replaced the steam-jet system of handling ashes. In the Steamatic system, air flow is produced by a ring-nozzle ejector and a Venturi tube placed at the discharge end. The following claims are made: 1. Low steam consumption. 2. Relatively little wear on pipe lines and pipe turns. 3. Since ashes are in contact with steam for only 3 ft. of travel, they enter the bin dry, and do not pack, or freeze in winter. 4. Less noise. 5. No danger of explosion in storage bin. This system requires steam at 70 lb. per sq. in. or higher, will carry up to 200 tons of ashes per day up to 300 ft. distance and lift through 100 ft. Pipes are

4, 5, 6, 8 in. diam. The 8-in. size handles large clinkers and is most economical of steam per ton of ashes carried. Small pipes may be used as branch lines for fly ash and soot.

The Nuveyor system (United Conveyor Corp.) is similar in principle to the Steamatic. The pipe line is the same, but to produce suction a positive air exhauster of the Roots blower type or a multi-jet exhauster is used. In both cases ashes pass to a cyclone separator; then by an automatic timing device, which acts every 90 sec., a release valve opens, suction stops for 5 sec. and ashes in the cyclone drop into the storage bin. Fig. 5 shows such a system with an air washer between cyclone and exhauster. No dust enters the latter and discharge to atmosphere is clean. The air washer acts also as a silencer. The Nuveyor will handle 30 tons of ashes per hour, is quiet in operation and ashes entering the storage bin are dry. The system is operated by remote-control buttons at points where ashes are fed to pipe line.

**ASH DISPOSAL BY WATER FLOW.**—In the Hydrojet system (Allen-Sherman-Hoff Co., Philadelphia) ashes accumulated under boiler furnaces are dislodged by a water jet at 100 lb. per sq. in., and moved through a trench below the basement floor by similar jets. Fig. 6 shows the system applied to the ash pit of a chain grate stoker. The whole apparatus is enclosed. The jet breaks down the pile of ashes; the mixture of water and ashes flows into the trench under the floor, lined with U-shaped cast-iron plates; nozzles at intervals deliver jets in the direction of flow. Outside the boiler house the discharge can flow to waste land or to a sump from which ashes are removed by grab bucket or pumped to a drainage bin for removal by car or truck. Advantages claimed for this system are: 1. No escape of dust, gas or steam into basement. 2. Men cannot be gassed or burned. 3. No leakage of air from basement into furnace to disturb combustion. 4. Operation is intermittent; time is available for repairs when necessary. 5. Overhead ash hoppers and cars are not used; boiler house can be built 10 ft. lower.

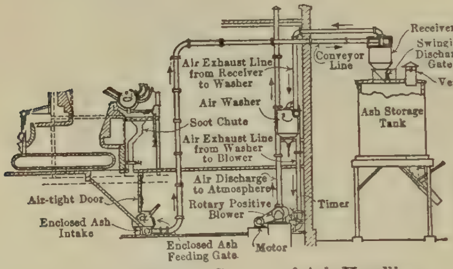


Fig. 5. Nuveyor System of Ash Handling

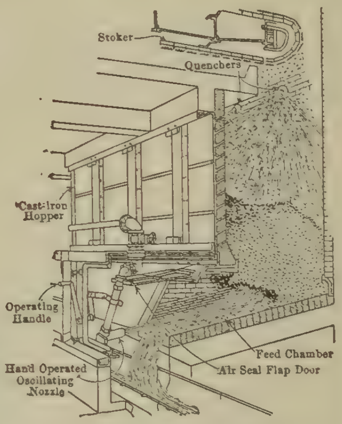


Fig. 6. Hydrojet System of Ash Removal

The Hydrovac dust handling system (Allen-Sherman-Hoff Co.) for boiler houses removes siftings under stokers, fly dust from boiler settings, soot and floor sweepings, by means of air flow under vacuum produced by high pressure water jets in a Venturi tube. The siftings can be blown into a furnace, refuse can be collected in a tank for removal or discharged into the water trench of the Hydrojet system.

For mechanical methods of ashes handling, see p. 23–57. For hydraulic ash handling systems see *Trans.* A.S.M.E. 1928, MH–50–12.

# PACKAGE HANDLING

**TYPES OF PACKAGE ELEVATORS.**—1. For boxes, barrels, bales and sacks, the rigid arm elevator, Fig. 1, will discharge over head wheel only. 2. The automatic arm elevator, Fig. 2, has carriers arranged to lift and lower; lifting side is so hinged that it will dump a barrel or sack forward when a lever on the arm contacts with an adjustable cam on the lifting side of elevator. On lowering side, the arm is rigid. Packages going down are deposited on skids at unloading point. 3. Suspended tray elevators, Fig. 3, have trays so hung on pivots between two chains that center of gravity of loaded tray is well below points of suspension. These trays, when made with finger bottoms, will pick up packages from skids on the up run and deposit them on skids on the down run. Solid bottom trays require loads to be put on or taken off by hand. Suspended tray elevators run at not over 30 ft. per min. 4. Elevators with corner-hung trays, Fig. 4, are similar to [3] but the chains are not directly opposite each other and pivots are not in line. This makes tray more stable in passing around turns and on straight runs, but wheels are out of line and the drive is not so simple. 5. Tray elevators with selective discharge: Style A, long trays with fingered bottoms with one loading compartment for each discharge point. Receiving fingers at each floor are set to register with compartment for that floor. Style B, fingered trays dumping forward on up run. Tray guides have a throw switch at each floor to tilt the tray. Switches are controlled by latches on tray which are set by an indicator at loading point.

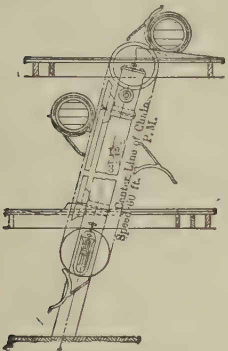


Fig. 1. Rigid Arm Elevator

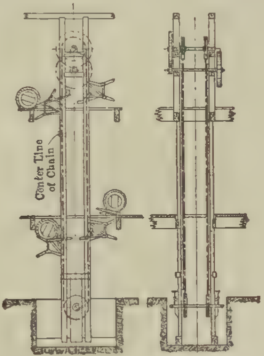
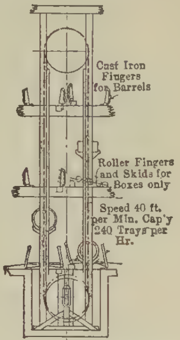


Fig. 2. Automatic Elevator


Cast Iron Fingers for Barrels
Roller Fingers and Skids for Boxes only
Speed 40 ft. per Min. Cap'y 240 Trays per Hr.
Fig. 3. Suspended Tray Elevator

Package elevators in these various styles are made by nearly all builders of conveying machinery. Much ingenuity has been spent on special forms. Elevators for trays of dishes, elevators to deliver to or to take from roller conveyors, elevators for selective discharge are made (see p. 23–81) by manufacturers [46], [51], [55], [65], and others.

**PACKAGE CONVEYORS.**—1. Packages of uniform size, not too heavy, can be moved in a trough by pushers attached to a

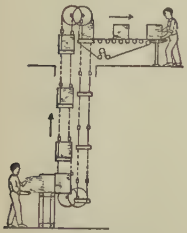


Fig. 4. Corner Hung Tray Elevator

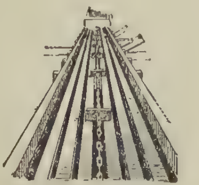


Fig. 5. Single Chain Package Conveyor

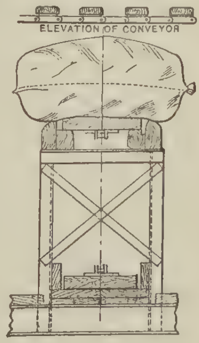


Fig. 6. Slat Conveyor

single chain. See Fig. 5. 2. Packages can be carried on slats which slide on both runs, Fig. 6. 3. Slat conveyors on roller chains will handle packages of any size or weight. Discharge is generally over the end, but movable plows, Fig. 7, will deflect boxes sideways for intermediate discharge. 4. Belt conveyors are used in shops, assembly departments, stores, post-offices, etc., for handling stacks of light package goods of all kinds. For this work the conveyor has straight tubular idlers and the edges of the belt are supported to prevent sagging between idlers. Fig. 8-A shows the construction with steel guards used in department stores, the goods being delivered by chute from above. Fig. 8-B shows the construction where packages are tossed on the belt from below. For light work the belt may slide on a hard wood plank. 5. Packages of uniform size can be carried long distances and at any angle of slope by wheeled carriages attached to a single chain, Fig. 9. 6. Inclined conveyors, Fig. 10, are used for handling and piling bags and bales. They have a belt to which cleats are attached, or else push-bars mounted on two strands of chain.

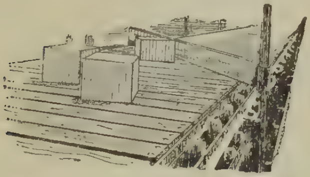


Fig. 7. Slat Conveyor on Roller Chain

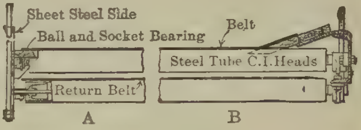


Fig. 8. Belt Conveyor for Packages

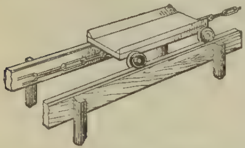


Fig. 9. Package Carrier

Package conveyors are built by all makers of conveying machinery. For information concerning them and special conveyors to work with gravity roller systems, see catalogs of the several makers. Selective discharge from belt or roller conveyors is described by W. O. Hildreth, *Trans.* A.S.M.E, MH–51–8, 1929. For selective discharge actuated by photo-electric relay, see *Matl. Handling and Distrib.*, May 1931, and other issues in 1931 and 1932. The electrical apparatus is made by General Electric Co.

A novel form of portable flexible conveyor for goods in bales and sacks is the Twinveyor (Clark Tructractor Co., Battle Creek, Mich.). It comprises several 8-ft. sections coupled end to end by flexible joints. Each section is made of two parallel tubes 4 in. diam. geared together and driven by a 5-Hp. motor, with a blunt screw thread on the outside. As the screws revolve the bag or bale resting on them acts as a nut and travels at 90 ft. per min. The original installation is described by J. T. Buzzo, *Trans.* A.S.M.E., MH–50–2, 1928.

**THE MONORAIL CHAIN CONVEYOR** consists of a power-driven chain suspended from trolleys on a continuous track, usually overhead. The chain joint pins are in a vertical plane. It bends around sprocket wheels or nests of rollers, or it can bend around tracks curved in a vertical plane if the proper chain is used. Loads are carried on hooks, racks, trays, slings, etc., hung from the trolleys. Speed generally is less than 75 ft. per min. The chain may be in one plane or several. It may serve several floors of one building or connect several buildings. Work can be done on suspended pieces as they travel along to dipping vats, baking ovens, etc. Monorail chain conveyors do work for which floor trucks and freight elevators otherwise would be required. They occupy no floor space, require little power and provide a steady movement of parts from one operation to another and to final assembly, inspection and shipment.

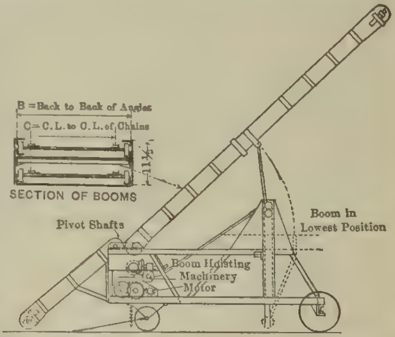

B = Back to Back of Angles
C = C.L. to C.L. of Chains
11½"
SECTION OF BOOMS
Pivot Shafts
Boom in Lowest Position
Boom Hoisting Machinery Motor

Fig. 10. Inclined Conveyor for Piling Packages

Fig. 11 shows a light trolley with 3 1/4-in. pressed steel ball-bearing wheels with two styles of hangers. The hangers are forgings which pass through the middle of a rivetless drop-forged chain. The pins in this chain have double heads and there is clearance enough in the joints to allow side flexure. These trolleys also are made with 4 wheels.

Fig. 12 shows a heavy trolley with 5-in. chilled roller-bearing wheels, running on a double angle track. The forged hanger occupies one pitch of a steel roller chain, and has a transverse joint to permit chain to roll around curves in a vertical plane without side strain on chain pins. Trolleys and hangers of this type (Palmer-Bee Co.) will carry loads up to 3000 lb.

**Calculation of Monorail Chain Conveyor.**—(Adapted from Link-Belt Data Book). 1. Make a layout, and scale off amount of chain required. 2. Assume a chain speed and a spacing of loads, as determined by time required to travel a certain distance, or the number of pieces to be delivered per min. 3. Sketch tray or carrier and make the horizontal and vertical turn so that carriers clear each other even if they swing. See Fig. 13. 4. From catalogs of makers select a trolley suitable for the load. These have been standardized. 5. As a first approximation, assume horsepower pull for a single-plane conveyor to be 5% of total moving weight (chains, trolleys, carriers, loads), about 7% for a conveyor with a few easy vertical curves, and 10% for one with many curves. 6. Select a chain strong enough for (5), and suited to the trolley. 7. Locate drive where sprocket wheel has at least 90° arc of contact with chain; 180° is better. The take-up should be where chain is under light load. Vertical turns work best if chain passing them is not under heavy tension. 8. For more accurate determination of chain tension and horse-power, calculate chain pull for each run and each turn. Thus, $C_H = WLX$; $C_C = YC$, where $W$ = weight of moving parts, lb. per ft., empty or loaded as the case may be; $L$ = length of run, ft.; $X$ = 0.03 for ball- or roller-bearing trolleys; = 0.05 for bronze-bushed trolleys; $C_H$ = pull for each straight horizontal run, lb.; $C_C$ = pull for each corner, take-up idler or roller turn in horizontal plane, lb.; $C$ = chain pull at turn, lb.; $Y$ = 0.03 for sprocket wheel or roller turn; = 0.02 for a traction wheel turn. The pull for a double curve in passing from one level to another is

$$C_V = WSX + CZ + HW,$$

where $W$ and $X$ are as above; $S$ = horizontal length of double curve, ft.; $H$ = height of double curve, ft., taken as (+) when travel is up, (−) when travel is down; $C$ = chain pull at entrance to the curve, lb.; $Z$ = friction factor due to radial pressure of trolley on curve; for ball or roller bearings $Z$ = 0.001 per deg. of inclination from the horizontal of tangent between curves; for bronze-bushed trolley wheels, $Z$ = 0.002. 9. To algebraic sum $C_H + C_C + C_V$, add 3% for friction at drive wheel. Check this against strength of chain, assumed in (6). 10. The radial pressure on each trolley is $P = BC/R$, where $B$ = trolley spacing, in.; $R$ = radius, in., to center of chain on curve; $C$ = maximum chain pull at any curve, lb. Check this with permissible trolley load. If too great, space trolleys closer or make radius longer. The longer the radius the better will be the operating results.

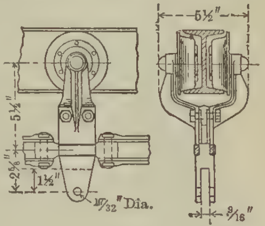


Fig. 11. Monorail Conveyor

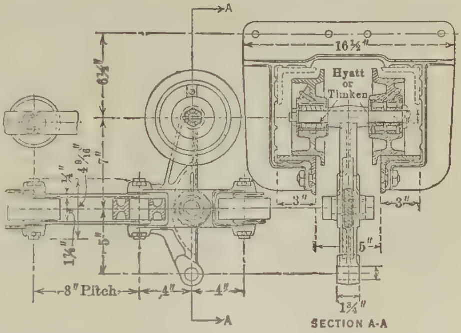


Fig. 12. Palmer-Bee Overhead Conveyor

Where conditions make it difficult or impossible to drive a chain by a sprocket wheel, an auxiliary drive acting on the straight run of chain may be used. The auxiliary chain is short and has pushers which engage links of the conveyor chain. The Webb system of using a number of these devices synchronized, and with automatic speed control, is described by H. C. Rundle, *Trans.* A.S.M.E, MH–53–7, 1931.

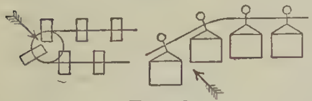


Fig. 13

Monorail chain conveyors are made (see p. 23–81) by manufacturers [50], [57], [69], [88], and other makers of conveying machinery.

**ROLLER CONVEYORS,** Fig. 14, will convey down slopes of 1 1/2% to 5%, boxes, packages, cans, bottles, lumber, steel products, almost anything that has a smooth face or side. The goods travel over rollers of steel tubing fitted with ball bearings. Capacities range from a few pounds up to 4 tons per roller. Some conveyors have spaced wheels instead of rollers. Rollers are assembled in straight sections, curves, switches, $Y$-assemblies and roller spirals. They can be set on floors supported from walls or hung from ceilings. Roller conveyors have been highly developed and have a wide field of usefulness. For detailed information, see catalogs and bulletins of manufacturers [46], [51], [55], [65], [83]. (See p. 23–81.) For a description of an elaborate system of Logan Co. conveyors with a new form of switch, see C. K. Pevear, *Trans.* A.S.M.E., MH–52–6, 1930.

**SPIRAL CHUTES** are made by manufacturers of roller conveyors, and by Hazlett Spiral Chute Co., Oaks, Pa. Some are made with roller beds, others have steel plate bottoms. They carry boxes, packages and even some kinds of unwrapped goods from upper to lower floors. Some are made with two or more spirals. They provide means for delivering different classes of goods to separate unloading stations at the lowest floor of a building, or goods can be passed from one floor to another on one spiral without interfering with goods on other spirals. Fig. 15 shows diagrammatically a chute with 6 spirals serving 13 floors of a clothing factory. The letters *A*, *B*, *C*, *D*, *E*, *F*, indicate separate spirals and the arrows show inlets and outlets. *G*, *H* and *J* are straight chutes in the system, not part of the spiral chutes, but all combine to provide a gravity flow for raw materials from the 13th and 11th floors to manufacturing floors below, and finally as finished goods to the shipping room in the basement.

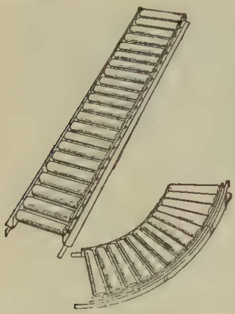


Fig. 14. Roller Conveyor

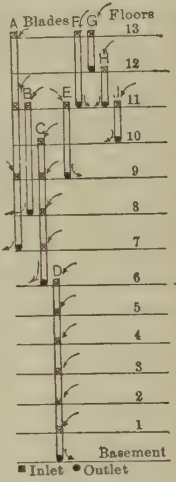


Fig. 15. Scheme of Spiral Conveyor System

**PROCESSING OVENS.**—Monorail and other forms of chain conveyors are used to carry cores, lithograph plates, and enameled metal parts through drying and baking ovens, and to carry metal parts through ovens in which they are dipped and drained. Such ovens are made by Young Bros. Co., Detroit; Foundry Equipment Co., Cleveland. Other forms of processing ovens are equipped with roller beds for handling cores and other parts on pallets. Some carry the work on trays suspended from two chains.

The conveyor tunnel kiln made by Swindell-Dressler Co., Pittsburgh, conveys tile, refractory brick, and similar ceramic materials through a firing kiln by means of rocking frames which support the materials. In other forms of the Dressler kiln, castings, steel plate or other heavy materials are annealed while they move through the kiln on a train of cars propelled by a ram.

# VEHICLES

## 1. TIERING MACHINES, LIFT TRUCKS AND SKIDS

**TIERING MACHINES, Portable Elevators, Stackers** for piling cased goods, barrels, bales, rolls of paper, etc., are made in a variety of styles and sizes by manufacturers [10], [25], [48], [74], (see p. 23–81) and others. For light service or occasional use they are operated by hand-crank, fitted with a safety brake to prevent it from flying back and injuring operator. For regular service, an electrically-driven machine is more useful. These machines also are used to handle dies to the level of press-beds, put up and take down overhead motors and shafting, support rigging and platforms for painting, cleaning and repair work, etc. Fig. 1 shows a hand-power stacker for lifts of 4 to 12 ft. The frame is hinged so that the machine can pass through a doorway. Similar hand-power and electric machines with telescopic frames are made for 2000 lb. loads and 12 ft. lifts.

The Revolvator (Revolvator Co.) has a turn-table base which locks in any of 4 positions. It is easy to load and convenient to use in narrow aisles or to stack goods against a wall. Revolvators, hand-driven or electric, will lift 5000 lb. loads to heights of 18 ft. See Fig. 2. They travel on 4 wide-face roller-bearing wheels, but when they are used for stacking, the load is transferred from wheels to legs so that the frame cannot roll.

**SKIDS AND LIFT TRUCKS.**—The following is taken from Simplified Practice Recommendation R95–30, U. S. Dept. of Commerce, 1931: The skid platform, or skid, is a flat, rectangular platform mounted usually on two parallel runners at or near the sides, or four legs near corners of platform. It is handled by a lift truck comprising two parallel, horizontal frames, on 3, 4 or 6 wheels. The upper frame, or platform, is so mounted that it may be raised above its lowest position. The truck, with lifting platform in the lowest position, is run under the skid; the lifting platform is raised, lifting skid supports clear of the floor. The truck then may be moved, carrying the skid. When skid is to be set down, the lifting platform is lowered until the skid supports rest on the floor. With lifting platform in its lowest position, the lift truck may be withdrawn.

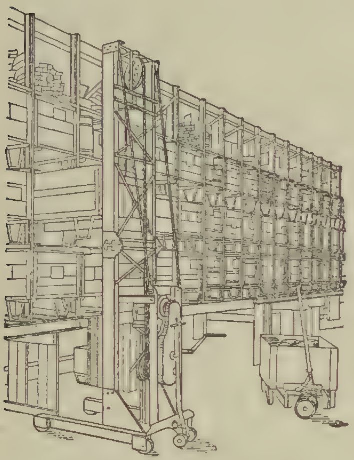


Fig. 1. Hand Power Stacker

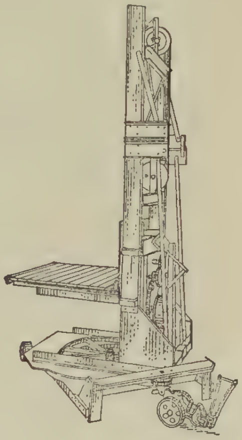


Fig. 2. Revolvator

Various attachments to skids have been developed to carry different commodities, including sides and ends permanently fastened to skid, superimposed removable boxes, fixed shelves and racks, removable vertical posts at sides or ends, and special attachments for particular commodities. Plain skid platforms carry paper, hides, rubber, machinery, sacked goods, boxed, crated and barreled goods and numerous other materials.

Advantages of handling goods on skid platforms include savings in time and labor, avoidance of loss or damage from piecemeal handling, and facility in handling and storing in large units without breaking bulk. Indirect advantages are simplified cost and inventory accounting and greater mobility of stores, supplies and materials. The paper and other industries began the shipping of goods on skids. Advantages of " skid shipment," include all advantages of intraplant skid handling, plus savings in packing material, labor, cost of loading and unloading, and elimination of breaking bulk in rehandling. Savings actually made range from 25 to 90%.

**Clearance and Overall Dimensions.**—To standardize dimensions of skid platforms to allow them to be used interchangeably in different plants and by all makes of lift trucks, a committee of manufacturers and users of skids and trucks, working with the U. S. Dept. of Commerce, adopted (1931) the following: Clear height under bottom of platform, either 8 or 12 in.; least width between legs or runners, 29 in. Overall dimensions: No. 1, 54 in. long, width not over 33 nor under 32 in.; No. 2, 60 in. long, 42 in. wide.

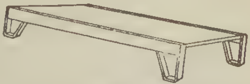


Fig. 3. Steel Frame Skid

**Skid Construction.**—To withstand rough handling, skid platforms should be strongly built. Fig. 3 shows one style (Lewis-Shepard Co.). The frame is arc-welded steel; the top is wood; the open sides allow a lift truck to be pushed in from any of the 4 sides. Other skids are: Built-in roller-beds (Yale & Towne); all-steel platforms of one steel plate bent to form top and sides (Truscon Steel Co., Cleveland); mounted-on dump-hoppers (Wilder-Strong, Monroe, Mich.). Skids which do not require a lift truck are made by manufacturers [22], [50] (see p. 23–81). They have 2 wheels in back and legs in front. A separable handle, fitted with a wheel, is thrust under the platform and engages a pin; when the handle is pushed down the front of the platform is lifted, the weight is taken off the legs and the platform is on rollers. Boxes, on skid legs or runners, in which small parts are handled and stored, as in Fig. 1, are made by the companies named above.

**LIFT TRUCKS.**—Lift trucks in 3 or 4 styles are made to handle loads from the lightest up to 5 tons. For the heaviest service, some trucks have a hydraulic lift operated by the pull-handle, which raises skid platform to clear the floor 3 in. or more. See Fig. 4. The "Jacklift" truck, Fig. 5, has a step-by-step ratchet lift; it is made in several models with lifts from 2 1/2 to 6 in. for loads up to 5 tons. The "foot-lift" model is worked by a treadle. Similar lift trucks are made by manufacturers [10], [48], [74], [95] (see p. 23–81), and others. They differ chiefly in lift and release mechanisms. All makers will furnish trucks with long frames for handling long narrow articles, core racks, drying frames, etc. Trucks with flanged rollers to run on tracks also are listed.

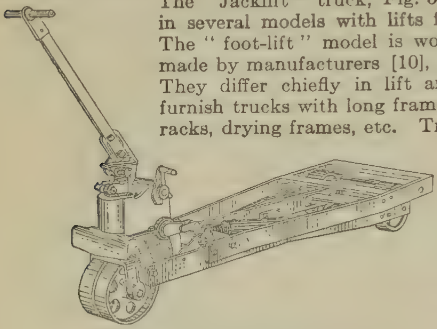


Fig. 4. Hydraulic Lift Truck

*Trans.* A.S.M.E., MH–50–4, 1928, MH–51–9, 1929, contain papers on handling materials on skids. These papers and the discussion of them bring out the following advantages of the system: 1. Easier handling. 2. Lower costs. 3. Time saved. 4. Less material carried in stock and inventory. 5. Greater safety to men. 6. Saving in packing for shipment. 7. Lower freight charges.

Storage systems using steel racks, to which box-skids or platform-skids are brought by lift trucks, and stored by a stacker are made by manufacturers [10], [48] (see p. 23-81), and others. Fig. 1 shows the parts of such a system. Material can be stored ceiling-high, floor space is saved and all skids easily can be reached and removed.

## 2. ELECTRIC TRUCKS AND TRACTORS

**ELECTRIC TRUCKS, TRACTORS AND TIERING MACHINES.**—Bulletins of the following companies (see p. 23–81) show these machines and describe the uses to which they are put: [5], [6], [7], [27], [58], [95]. Developments have been the use of stronger steel frames, roller and ball bearings, better motors, enclosed gearing, better lubrication, large capacities, and adaptation to handling a greater variety of products. Many forms of tractors, lift-trucks, tiering machines and crane-trucks are in use. See *Trans.* A.S.M.E., MH–53–4, 1931. Various types are in use as follows:

1. **Trucks with Fixed Platforms** used in shops and warehouses to carry loads up to 6000 lb., and also in railroad service to handle baggage. The operator stands on a hinged platform at one end, one hand on the steering-lever, the other on the controller handle, his foot on the brake pedal. If he steps off, the brake is set and the current shut off. Fig. 6 shows a small truck. These travel 4 or 5 mi. per hr. loaded, steer on all four wheels, are reversible and can be handled in small spaces and narrow aisles. Trucks with good springs and 20-in. wheels can run over rough floors and block pavement.

2. **Trucks with Lift Platforms,** used principally to handle material on skid platforms (see p. 23–74), are built to lift and carry loads from 4000 to 20,000 lb. In low-lift trucks a cam or link motion in the base frame raises the platform to give floor clearance. On high-lift trucks the platform is raised by cables or chains, by a screw working in a movable nut, or by hydraulic cylinders. All truck builders furnish machines to handle standard skids (see p. 23–75) as well as special or higher skids. Wheels under platforms of lift-trucks for low skids seldom are over 10 1/2 in. diam. and roughness and unevenness of floor affect them more than trucks with larger front wheels. For effect of bad floors, see p. 23–78.

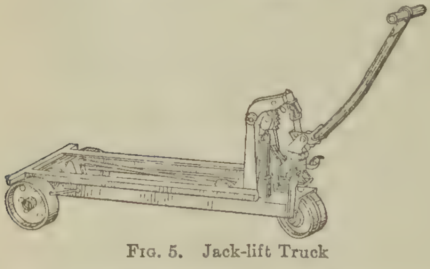


Fig. 5. Jack-lift Truck

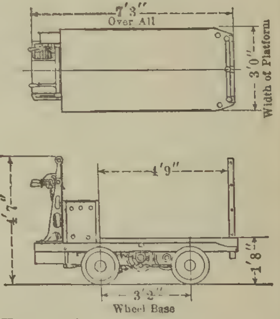


Fig. 6. Electric Platform Truck

Fig. 7 shows a 4000-lb. low-lift truck which will lift 4 inches, skids 8 in. high. Fig. 8 shows a high-lift truck which will raise a 6000 lb. load 6 ft. The platform is 55 × 27 in. High-lift trucks pile and tier goods, serve presses and handle freight to and from box cars. Trucks of this type are made with special platforms or attachments to do such work as the following: *a.* Pick up bales from floor by thrusting forks under them (called *chiseling*). *b.* Handle 5-ton reels of wire by pulling them from floor to a tilting platform by a power winch. *c.* Carry piles of tin-plate with a top clamp to keep pile from shifting. *d.* Place heavy stamping dies in position on a press platen.

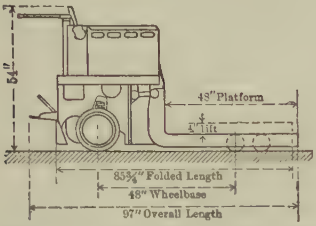


Fig. 7. 2-Ton Electric Lift Truck

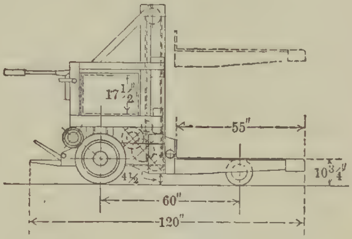


Fig. 8. 3-Ton Electric High-lift Truck

**THE FORK TYPE LIFT TRUCK.**—The forks, Fig. 9, can be lowered to the floor to pick up material resting on battens, or even chisel under some kinds of packages on the floor. Machines of this type are compact enough to work in and out of box cars, will handle loads of 1/2 to 3 tons or more.

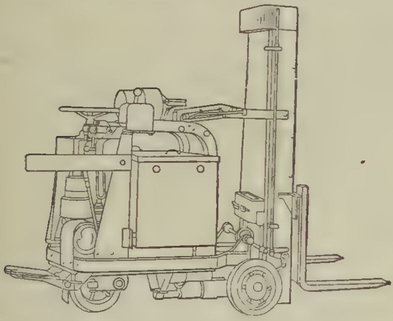


Fig. 9. Fork Type Lift Truck

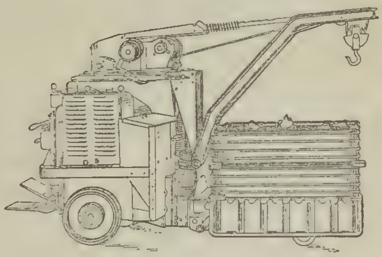


Fig. 10. Electric Lift Truck with Crane

When fitted with telescoping uprights they will lift and tier to heights of 6 ft. A truck which acts as a tractor, lift-truck, crane-truck, and portable crane is listed in four sizes; crane capacity up to 2000 lb.; platform capacity up to 10,000 lb. See Fig. 10.

Storage battery cranes for shop service are made by manufacturers [7], [27] (see p. 23–81), and others. They have low bases fitted with booms, fixed or telescopic, rotating and with luffing motion. Capacities are up to 6000 lb. at 7 ft. radius, but with wheel base short enough to permit maneuvring in shop aisles.

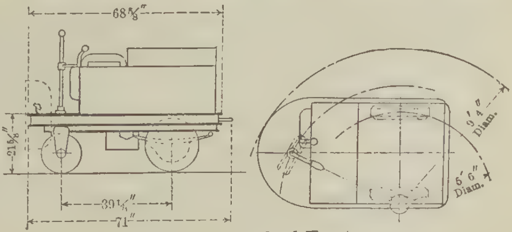


Fig. 11. 3-Wheel Tractor

**ELECTRIC TRACTORS** are built in various sizes with various kinds of storage batteries by makers mentioned above. The smallest tractors, Fig. 11, have three wheels, two for drive, one for steer. Intermediate sizes steer on two wheels; the largest and most powerful on all four wheels. Some makers list tractors with 4-wheel drive. Some tractors have flanged wheels to run on narrow-gage tracks.

Tractors are economical where quantity of material handled is enough to warrant use of trains of trailers, and where movement of trains can be arranged according to a timetable. In a railroad shop in Milwaukee, covering 3 sq. mi., the stores department and various shops are served by trains of trailers drawn by storage battery tractors. They make the circuit every 45 min. and stop at 11 stations en route. Where possible, material is kept on skids, handled, moved and delivered to trailers by lift-trucks. Empties are placed at the various stations as required, to be removed by a later train when loaded. Many railroad stations use several tractors, with 50 to 100 trailers per tractor. Advantages of this system, as compared with 2-wheel hand-pushed trucks, are: 1. 50% or more time saved in getting freight from wagon or truck to freight car, or *vice versa*. 2. More systematic loading of freight cars and more prompt dispatching. 3. Less damage to freight, fewer complaints from shippers. 4. Saving in cost of freight handling sufficient to pay for entire investment in a few years; in some cases in a few months.

Trucks and trailers are made by many manufacturers; some (see p. 23–81) who specialize in trailers for haulage by electric or gasoline powered tractors are [5], [19], [24], [58], [62], [84], [95].

**Table 1.—Cars and Trailers for Different Services**

(Easton Car & Construction Co., Easton, Pa.)

| Moved by | Conditions and Loads | Style of Car (see note) Choice 1st | 2d | 3d |
|---|---|---|---|---|
| Power | Accurate trailing | A | B | C |
| " | Accurate trailing and reverse | A | D | ...... |
| " | Good or rough runways, heavy loads on 2 or 3 trailers | C | A | D |
| " | Good runways, heavy loads on 4 or more trailers | B | A | C |
| " | Rough runways or dirt floors, heavy loads on 4 or more trailers | C | A | ...... |
| " | Rough runways or dirt floors, light loads | A | C | ...... |
| " | In conjunction with considerable hand trucking | B | C | D |
| Hand | Good runways, light loads | B | D | C |
| " | Poor runways, dirt floors, heavy loads | C | ...... | ...... |

Note.—A—4-wheel-steer trailer; B—Caster trailer or truck; C—Fifth-wheel trailer or truck (Wagon-type); D—Balanced trailer or truck. On right-angle turns in 7 ft. aisles, caster trailers will creep outward 3 or 4 in.; fifth-wheel, 7 to 10 in.; balanced trailers with cross-chain connections. 7 to 10 in.; 4-wheel steer trailers will not creep. The larger the radius of turn, the less the creep.

**Four-wheel Steer Trailers** guide well but do not turn sharp corners readily and are awkward to move by hand.

**Caster Trailers** carry most of the load on two large wheels and steer by swivel casters at front or rear. They move easily, steer well and can be maneuvred in narrow aisles. They are the best for general use.

**Fifth-wheel or Wagon-type Trailers** or trucks are for heavy loads and rough service. They are not reversible. They can be made with large wheels for use on bad paving or rough floors.

**Balanced Trailers** or trucks are like warehouse hand trucks, carry the load on two large wheels under middle of the body and have single or double casters at the ends. They turn easily but do not guide well at high speed.

On smooth new floors, trucks and trailers with steel wheels start easier and roll easier than rubber-tired wheels, but on most floors and on all runways, ramps and paved yards the rubber-tired or canvas-cushion tired wheels have the advantage. The pull is less, goods are not shaken or damaged, wear and tear on the truck is less, noise is less and damage to floors is eliminated.

CONDITION OF FLOORS AND RUNWAYS.—Table 2, based on tests of trailers fitted with roller bearings and solid rubber tires shows that power required to keep them

**Table 2.—Rolling Resistance of Trailers on Various Floors and Roads**

(Plain Bearings)

| Kind of Surface | Average Rolling Resistance, lb. per ton |
|---|---|
| Hard plank; wood block; concrete; asphalt in good condition | 20 |
| " " " " but worn | 40 |
| Macadam in good condition; gravel rolled hard | 35 |
| " poor condition | 70 |
| Dry earth, hard, dusty | 60 |
| Ordinary earth road | 100 |
| Muddy earth road; frozen ruts; loose gravel; loose sand | 200 |
| For each 1% of up-grade, add 20 lb. per ton. | |

moving varies greatly. Incidental disadvantages of poor floors and runways are: 1. Tractor operator will run slowly to prevent goods being jolted off trailers. 2. Cost of repair and maintenance of equipment is increased. 3. Labor cost due to delay and stoppages is increased. Checkered steel plate or hard wood plank laid lengthwise to form runways or ramps often will prove profitable. On rough floors, the advantage of ball or roller bearings tends to disappear, especially at speeds above 3 mi. per hr. On good floors the rolling resistance of 8-in. wheels fitted with roller bearings is four times that of 20-in. wheels. See Hellmich, in *Zeit.* V.D.I., Mar. 4, 1933.

**COST OF HANDLING BY ELECTRIC TRUCKS.**—C. B. Crockett and H. J. Payne (*Trans.* A.S.M.E., MH–50–3, 1928), give the following as average costs per year: Depreciation of electric truck, $300; depreciation of storage battery, $200; depreciation of charging equipment, $15; interest, maintenance and storage charges, $245; electric current for charging batteries, $225; maintenance and repairs, $225; total, $1210. This sum is approximately equal to a man's wages. To it add wages of operator, and total cost of operating an electric truck is, roughly, two men's wages. If an electric truck can save the labor of only two men, its direct saving is doubtful; if it eliminates 3 or more men, it can show a profit. Labor cost alone is not all that should be considered. If trucks speed up handling, reduce lost time and delays, and increase output of a plant, cost of truck equipment must be balanced against cost of increased plant necessary for increased production. Material kept moving reduces inventory, material in process and money tied up thereby. These differences should be credited to the truck system.

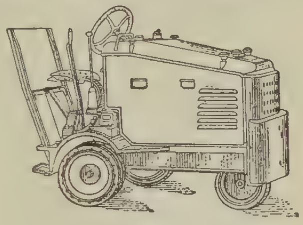


FIG. 12. Gasoline Tractor

The Crockett-Payne diagrams (see above) show: 1. Trucking by hand may be cheaper than haulage by truck if the work takes only $^1/_7$ of full time, *i.e.*, 1 $^1/_5$ hrs. per day, or 44 full days per year. 2. Fixed charges of electric trucks may be higher than those of gasoline trucks, but under favorable conditions cost of power and repairs is less. 3. For long hauls, or 24-hr. service, cost of handling by gasoline truck may be less than that for haulage by storage battery truck. 4. For bad floors or badly paved yards, haulage by gasoline truck may be cheapest. Electric haulage is particularly suited to transfer of material of considerable volume and weight between departments of shops and warehouses, where haulage can be done on a regular schedule that allows 5 to 7 hr. for recharging batteries; where floors inside or paving outside are fairly good; where quiet operation is an advantage and where fire risk and noise of gasoline tractors may be objectionable.

**GASOLINE TRACTOR TRUCKS AND TIERING MACHINES.**—For general haulage and service in shops, warehouses and freight houses, gasoline tractor trucks are made in sizes corresponding closely to storage battery trucks. Clark Tructractor Co. makes tractors in various sizes. The smallest, Fig. 12, weighs 2700 lb., has a draw-bar pull of 2000 lb., and can haul 25 tons on trailers over good floors. It is 39 in. wide, 78 in. long, 59 in. high, has a 2-wheel drive, 1-wheel steer and will turn on a radius of 51 in. Another model for travel over very rough floors or paving has four wheels, and a turning radius of 92 in. In both, the engine is 4-cylinder, rated at 12 Hp. It has an automatic speed governor, dry plate clutch, and 4-speed gear box. Diameter of driving wheels is 21 in.; of steering wheel, 15 in. Tires are solid rubber or pneumatic. A condensing muffler prevents escape of hot gases. A larger 4-wheel tractor has a draw-bar pull of 3600 lb. in low gear and a turning radius of 108 in. Diameter of driving wheels is 32 in.; of steer wheels, 24 in. The motor is rated at 40 Hp. The chassis of this machine can be provided with a shovel or a crane. The crane tractor has an 18-ft. pivoted boom working in one plane that will lift 4000 lb. 12 or 13 ft. from the floor. With boom lowered the machine will go through a 5 × 8-ft. doorway.

Clark Truclifts are made in two models: 1. Front wheel drive with rear wheel steer. 2. Rear wheel drive with 4-wheel steer. The latter will make sharp turns, can be more easily handled in narrow aisles and into and out of box cars. High-lift models will tier 3 ton loads up to 6 ft. high. The lifts are hydraulic. Lifting heads can be fitted with rams or forks like those on some storage battery machines, and the columns can be made to tilt to carry the load safely. Low-lift models will handle loads up to 4 tons.

Tractors for shop use built up around a Ford AA engine are made by Whitehead and Kales Co., Detroit; Mercury Mfg. Co., Chicago; W. F. Hebard & Co., Chicago. These tractors have relatively large rubber-tired wheels, and can travel up to 30 mi. per hr. Some have a box body. Some of the companies named above make gasoline-powered trucks with lift platform to handle U. S. standard skids.

Other gasoline machines suited to outdoor work or on roads have developed from agricultural or road tractors. International Harvester Co., Chicago, makes several types with steel wheels, rubber-tired wheels, or crawler treads. Some can be fitted with digging and excavating buckets, road scrapers, and earth elevators for road work. Similar machines with caterpillar treads are made by Caterpillar Tractor Co., Peoria, Ill.; Allis-Chalmers Mfg. Co., Milwaukee; J. I. Case Co., Racine, Wis., and others. The Roadmaster (Bucyrus-Erie Co.) is a revolving crane with a luffing boom on the chassis of a commercial tractor, with wheels or crawler treads. The Roustabout crane (Hughes-Keenan Co., Mansfield, O.) is a similar machine. It can be furnished with winch heads or power take-off. With boom lowered it will go through a 5 1/2 × 7 1/2-ft. doorway.

**USE OF GASOLINE-POWERED TRACTOR TRUCKS AND TIERING MACHINES.**—Many statements on p. 23–79, *et seq.*, concerning storage battery equipment, as to methods of handling and storing goods, cost of handling and probable saving, apply also to gasoline equipment. In addition to former statements the following advantages are claimed for gasoline tractors and trucks: 1. They can work 24 hrs. per day and take a fresh supply of fuel in 5 min. or less. 2. The torque of the engine and the use of the clutch exert a strong pull to start heavy loads or get over bad places in floors or paving. 3. They can go almost anywhere, are self-contained and independent of power lines and charging stations. 4. They can travel over streets, public roads, bad floors, rough paving.

The Ready-Power gas-electric unit (Ready-Power Co., Detroit) is a self-contained power plant for use on tractors and trucks designed for storage battery power where time cannot be spared for recharging. It is a 4-cylinder automobile-type gasoline engine, with automatic governor and generator armature instead of a fly-wheel. It will deliver 120 amp. at 36 volts, or 90 amp. at 48 volts, weighs 1000 lb. and fits in the space required by a storage battery.

**PROBABLE SAVINGS BY USE OF TRUCK SYSTEM.**—It is difficult to determine mathematically whether or not the use of any system of mechanical handling will be profitable. The Coes formula, *Trans.* A.S.M.E., vol. xliv, 1923, gives various debit and credit items which should be considered. If proper values are assigned, the justifiable investment and the probable profit based on the assumptions, will be given by the formula. It may happen, however, that indirect savings previously mentioned (p. 23–79) will determine whether or not a materials handling system will pay a profit. The Crockett-Payne paper referred to (p. 23–79) gives explicit data on operation of electric lift-trucks and tractors in twenty plants. All show saving of time and money. In some plants indirect and incidental advantages are most important. For examples of the use of the Coes formula, see Koshkin, Modern Materials Handling, p. 194 (John Wiley & Sons). See also *Matl. Handling and Distrib.*, July, Aug., 1931. The same magazine, in Mar., 1931, describes a case where 11 tractors and 23 trailers costing $27,000 saved $48,000 per year. The plant covered 40 acres, and all haulage was done on a timed schedule. For other cases, see a list of 50 installations in *Matl. Handling and Distrib.*, Jan., 1930.

**ELECTRIC INDUSTRIAL LOCOMOTIVES** are of several types: Overhead-wire, third-rail, storage-battery. For mine work, the first are in most general use. They are low, 4- or 6-wheel structures weighing 5 to 35 tons, for track gages as narrow as 18-in. and with one or two trolley poles. Two are used when rails are not bonded for the return circuit. 250-volt series motors generally are used. Haulage capacity, rated as draw-bar pull, usually is taken as $^1/_5$ of the weight of the locomotive, without excessive heating in the motor. The rating may be reduced to $^1/_2$ the standard rating for steady work on long hauls or steep grades. The pull for mine cars on straight level tracks may be taken at 30 lb. per ton; for grades, at 20 lb. per ton for each 1% of grade. Usual speeds are 6 to 8 mi. per hr. Locomotives of this type are made by the large electric manufacturing companies and by manufacturers [8], [34], [44] (see p. 23–81).

**Advantages of Storage-battery Locomotives** are: It is unnecessary to bond rails or remove rock to make legal clearances for trolley wires. They are safe in gaseous mines, can place cars at the working face, remove loaded cars, and go wherever a track can be laid. If batteries are recharged at night, the day load of the power plant is decreased. For mine service they are made in sizes up to 10 tons; for shifting railroad cars, size may be 50 tons. Shifting locomotives (Atlas Car and Mfg. Co., Cleveland) have draw-bar capacity of 30,000 lb. on sanded track, and have standard couplers in radial pockets to permit passing short-radius curves. They are designed to deliver a draw-bar pull of about $^1/_5$ their own weight, but speed of travel and amount of work available between chargings depend on size and discharge rate of the battery. Performance of different sizes and makes should be compared on this basis.

**STORAGE BATTERIES** are of two kinds: 1. The lead battery (Exide, Electric Storage Battery Co., Philadelphia) is very efficient as a storage of electrical energy, and is used for services requiring heavy rates of discharge. 2. The nickel-iron battery (Edison

Storage Battery Co., Orange, N. J.) is lighter than the lead battery but costs more. From $^1/_3$ to $^1/_2$ the price of a storage-battery locomotive is the cost of the storage battery. In shop tractors and trucks the proportion is about the same. To prolong life of cells and insure steady service it is important to use proper charging equipment. A simple, but wasteful, way is to reduce line current from 110 or 250 volts direct current to battery voltage by rheostat control. A better method is to use a motor-generator set whose current consumption is low, and with which the battery cannot be overcharged.

**COMBINATION LOCOMOTIVES** fitted with both storage batteries and current-collectors also are built. The General Electric Co. has built heavy locomotives with Diesel engine generating plants, storage batteries, and current collectors for railroad switching work. Gasoline locomotives are preferred for some kinds of haulage, as they are independent of a power house, have large radius of travel, and, as compared with steam locomotives, require less attention and use fuel only when working. They are made in sizes up to 30 tons, have 4- or 6-cylinder engines and gear shifts for various speeds. For outdoor work or in mines with good ventilation, the exhaust gases are not objectionable.

Diesel-engine locomotives in 25 and 30 ton sizes, also combination gasoline-electric or Diesel-electric locomotives in 50 and 60 ton sizes, are made. See publications of Fate-Root-Heath Co., Plymouth, O.; Whitcomb Locomotive Co., Rochelle, Ill.

## MANUFACTURERS OF MATERIAL HANDLING EQUIPMENT

In the preceding pages, the manufacturers of the several types of equipment described are referred to by number as follows:

1. Alliance Machine Company, Alliance, Ohio.
2. American Engineering Company, Philadelphia, Pa.
3. American Hoist & Derrick Company, St. Paul, Minn.
4. American Steel & Wire Company, Worcester, Mass.
5. Atlas Car & Manufacturing Company, Cleveland, Ohio.
6. Automatic Transportation Company, Chicago, Ill.
7. Baker-Raulang Company, Cleveland, Ohio
8. Baldwin Locomotive Works, Philadelphia, Pa.
9. Barber-Greene Company, Aurora, Ill.
10. Barrett-Cravens Company, Chicago, Ill.
11. R. H. Beaumont & Company, Philadelphia, Pa.
12. Blaw-Knox Company, Pittsburgh, Pa.
13. Bleichert, Leipzig, Germany.
14. Brown Crane Company, Cleveland, Ohio.
15. Bucyrus-Erie Company, Milwaukee, Wis.
16. Byers Machine Company, Ravenna, Ohio.
17. Chain Belt Company, Milwaukee, Wis.
18. Chisholm & Moore Hoist Corporation, Tonawanda, N. Y.
19. Clark Tructractor Company, Battle Creek, Mich.
20. Cleveland Crane & Engineering Company, Wickliffe, Ohio.
21. Clyde Iron Works, Duluth, Minn.
22. Colson Company, Elyria, Ohio.
23. Curtis Pneumatic Machinery Company, St. Louis, Mo.
24. Easton Car & Construction Company, Easton, Pa.
25. Economy Engineering Company, Chicago, Ill.
26. Electro-Lift, Inc., New York.
27. Elwell-Parker Electric Company, Cleveland, Ohio.
28. Euclid-Armington Corporation, Euclid, Ohio.
29. S. Flory Manufacturing Company, Bangor, Pa.
30. Ford Chain Block Company, Philadelphia, Pa.
31. Fuller-Lehigh Company, Fullerton, Pa.
32. Gillis & Geoghegan, New York.
33. Godfrey Conveyor Company, Elkhart, Indiana.
34. Goodman Manufacturing Company, Chicago, Ill.
35. Grindle Fuel Equipment Company, Harvey, Ill.
36. George Haiss Manufacturing Company, New York.

37. Hanna Engineering Works, Chicago, Ill.
38. Hardinge Company, York, Pa.
39. Harnischfeger Corporation, Milwaukee, Wis.
40. Hayward Company, New York.
41. Industrial-Brownhoist Company, Bay City, Mich.
42. Ingersoll-Rand Company, New York.
43. International Combustion Engineering Corporation, New York.
44. Jeffrey Manufacturing Company, Columbus, Ohio.
45. Kennedy-Van Saun Engineering Corporation, New York.
46. Lamson Company, Syracuse, N. Y.
47. A. Leschen and Sons Company, St. Louis, Mo.
48. Lewis-Shepard Company, Boston, Mass.
49. Lidgerwood Manufacturing Company, New York.
50. Link-Belt Company, Chicago, Ill.
51. Logan Company, Louisville, Ky.
52. Maine Electric Company, Portland, Maine.
53. Marion Steam Shovel Company, Marion, Ohio.
54. Maris Brothers, Philadelphia, Pa.
55. Matthews Conveyor Company, Ellwood City, Pa.
56. Mead-Morrison Manufacturing Company, New York.
57. Mechanical Handling Systems, Inc., Detroit, Mich.
58. Mercury Manufacturing Company, Chicago, Ill.
59. Morgan Engineering Company, Alliance, Ohio.
60. National Equipment Company, Milwaukee, Wis.
61. Northern Engineering Works, Detroit, Mich.
62. Nutting Truck Company, Faribault, Minn.
63. Ohio Locomotive Crane Company, Bucyrus, Ohio.
64. Ohio Power Shovel Company, Lorain, Ohio.
65. Samuel Olson & Company, Chicago, Ill.
66. Orton Crane and Shovel Company, Chicago, Ill.
67. Osgood Company, Marion, Ohio.
68. Page Engineering Company, Chicago, Ill.
69. Palmer-Bee Company, Detroit, Mich.
70. Philadelphia Gear Works, Philadelphia, Pa.
71. Pohlig, Leipzig, Germany.
72. Portable Machinery Company, Clifton, N. J.
73. Raymond Bros. Impact Pulverizer Company, Chicago, Ill.
74. Revolvator Company, Jersey City, N. J.
75. Riley Stoker Company, Worcester, Mass.
76. Robbins and Myers, Inc., Springfield, Ohio.
77. Robins Conveying Belt Company, New York.
78. John A. Roebling's Sons Company, Trenton, N. J.
79. Sauerman Bros., Chicago, Ill.
80. Shaw Electric Crane Company, Muskegon, Mich.
81. Shepard-Niles Crane Company, Montour Falls, N. Y.
82. Specialty Engineering Company, Philadelphia, Pa.
83. Standard Conveyor Company, Ellwood City, Pa.
84. Standard Pressed Steel Company, Jenkintown, Pa.
85. Stephens-Adamson Company, Aurora, Ill.
86. Sullivan Machinery Company, Chicago, Ill.
87. Thew Shovel Company, Lorain, Ohio.
88. Jervis B. Webb Company, Detroit, Mich.
89. Webster Manufacturing Company, Chicago, Ill.
90. Wellman Engineering Company, Cleveland, Ohio.
91. Whiting Corporation, Harvey, Ill.
92. G. H. Williams Company, Erie, Pa.
93. Wright Manufacturing Company, York, Pa.
94. Wright Manufacturing Company, Bridgeport, Conn.
95. Yale and Towne Manufacturing Company, Philadelphia, Pa.
96. J. S. Mundy Company, Newark, N. J.
97. Detroit Hoist and Machine Company, Detroit, Mich.
98. C. O. Bartlett & Snow Company, Cleveland, Ohio.
99. Gifford-Wood Company, Hudson, N. Y.

## Section 24

# MECHANICAL POWER TRANSMISSION

SHAFTING

PULLEYS

BELT DRIVES

CHAIN DRIVES

ROPE DRIVES

COUPLINGS AND CLUTCHES

SPEED CHANGING MECHANISMS

DYNAMOMETERS

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# MECHANICAL POWER TRANSMISSION

## SHAFTING

### 1. SHAFTING CALCULATIONS

**STANDARD DIAMETERS AND LENGTHS OF SHAFTING.**—Standard stock lengths for cold-finished shafting are 16, 20 and 24 ft. Diameters and tolerances of standard cold-finished shafting are given in Table 1.

**Table 1.—American Standard Cold-finished Shafting Diameters**

A.S.A. Standard B17a–1927

| Transmission Shafting Sizes, in. | Machinery Shafting Sizes, in. | Tolerance on Diameter, in. | Transmission Shafting Sizes, in. | Machinery Shafting Sizes, in. | Tolerance on Diameter, in. | Transmission Shafting Sizes, in. | Machinery Shafting Sizes, in. | Tolerance on Diameter, in. | Transmission Shafting Sizes, in. | Machinery Shafting Sizes, in. | Tolerance on Diameter, in. | Transmission Shafting Sizes, in. | Machinery Shafting Sizes, in. | Tolerance on Diameter, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ...... | 1/2 | −0.002 | 1 3/16 | 1 3/16 | −0.003 | ...... | 1 7/8 | −0.003 | ...... | 2 5/8 | −0.004 | 3 15/16 | 4 | −0.004 |
| ...... | 9/16 | −0.002 | ...... | 1 1/4 | −0.003 | 1 15/16 | 1 15/16 | −0.003 | ...... | 2 3/4 | −0.004 | ...... | 4 1/4 | −0.005 |
| ...... | 5/8 | −0.002 | ...... | 1 5/16 | −0.003 | ...... | 2 | −0.003 | ...... | 2 7/8 | −0.004 | 4 7/16 | 4 1/2 | −0.005 |
| ...... | 11/16 | −0.002 | ...... | 1 3/8 | −0.003 | ...... | 2 1/16 | −0.004 | 2 15/16 | 3 | −0.004 | ...... | 4 3/4 | −0.005 |
| ...... | 3/4 | −0.002 | 1 7/16 | 1 7/16 | −0.003 | ...... | 2 1/8 | −0.004 | ...... | 3 1/8 | −0.004 | 4 15/16 | 5 | −0.005 |
| ...... | 13/16 | −0.002 | ...... | 1 1/2 | −0.003 | 2 3/16 | 2 3/16 | −0.004 | ...... | 3 1/4 | −0.004 | ...... | 5 1/4 | −0.005 |
| ...... | 7/8 | −0.002 | ...... | 1 9/16 | −0.003 | ...... | 2 1/4 | −0.004 | ...... | 3 3/8 | −0.004 | 5 7/16 | 5 1/2 | −0.005 |
| 15/16 | 15/16 | −0.002 | ...... | 1 5/8 | −0.003 | ...... | 2 5/16 | −0.004 | 3 7/16 | 3 1/2 | −0.004 | ...... | 5 3/4 | −0.005 |
| ...... | 1 | −0.002 | 1 11/16 | 1 11/16 | −0.003 | ...... | 2 3/8 | −0.004 | ...... | 3 5/8 | −0.004 | 5 15/16 | 6 | −0.005 |
| ...... | 1 1/16 | −0.003 | ...... | 1 3/4 | −0.003 | 2 7/16 | 2 7/16 | −0.004 | ...... | 3 3/4 | −0.004 | ...... | ...... | ...... |
| ...... | 1 1/8 | −0.003 | ...... | 1 13/16 | −0.003 | ...... | 2 1/2 | −0.004 | ...... | 3 7/8 | −0.004 | ...... | ...... | ...... |

**HORSEPOWER TRANSMITTED BY SHAFTING.**—For shafts subjected to torsion only, let $d$ = diameter of shaft, in.; $P$ = force, lb., applied on a lever arm at a distance $r$ in. from the axis; $S$ = shearing resistance at outer fiber, lb. per sq. in. Then

$$Pr = \pi d^3 S/16 = d^3 S/5.1 = 0.1963\, d^3 S \quad [1]$$

$$d = \sqrt[3]{5.1\, Pr/S} = \sqrt[3]{P\, r/K} \quad [2]$$

where $K = S/5.1$.

If $N$ = revolutions per minute, the horsepower transmitted is

$$\text{Hp.} = \frac{P\, r\, 2\pi\, N}{33{,}000 \times 12} = \frac{\pi\, d^3\, S \times 2\pi\, N}{16 \times 33{,}000 \times 12} = \frac{N\, S\, d^3}{321{,}000} \quad [3]$$

$$d = \sqrt[3]{321{,}000\ \text{Hp.}/N\, S} = \sqrt[3]{C \times \text{Hp.}/N} \quad [4]$$

where $C = 321{,}000/S$.

In practice, empirical values are given to $S$ and to the coefficients $K$ and $C$ according to the factor of safety assumed, which depends upon the material, on whether the shaft is subjected to steady, fluctuating, bending or reversed strains, on the distance between bearings, etc. The factors of safety generally used in practice are: For head-shafts, 15; for line-shafts carrying pulleys, 10; for small, short shafts, countershafts, etc., 7. For steel shafting of 60,000 lb. per sq. in. tensile strength, the allowable stress $S$ with the above factors would be 4,000, 6,000, and 8,500 lb. per sq. in. respectively, whence the diameters of shafts would be

Head-shafts $$d = \sqrt[3]{80\ \text{Hp.}/N} \quad [5]$$

Line-shafts $$d = \sqrt[3]{53.5\ \text{Hp.}/N} \quad [6]$$

Short shafts $$d = \sqrt[3]{38\ \text{Hp.}/N} \quad [7]$$

Table 2 gives the horsepower transmitted by cold-rolled steel line-shafting.

**DEFLECTION OF SHAFTING.—Linear Deflection.**—For continuous line shafting, it is considered good practice to limit deflection to a maximum of 0.01 in. per foot of length. The weight of bare shafting, in pounds, is $W = 2.6\, d^2 L$. When the shaft is fully loaded with pulleys as is usual in practice, with an allowance of 40 lb. per inch of width for the vertical pull of belts, the load in pounds will be about $W \doteq 13\, d^2 L$. With a modulus of

### Table 2.—Horsepower Transmitted by Cold-rolled Steel Line-shafting

Shafting well supported by bearings 8 to 10 ft. apart, with pulleys located close to bearings.
For turned shafting, multiply the figures in the table by 0.775.
For turned shafting simply transmitting power, without pulleys, or short countershafts, multiply by 1.75.
For turned head shafting, multiply by 0.54.
For cold-rolled head shafting multiply by 0.70.

| Diam. of Shaft, in. | Revolutions per minute 100 | 150 | 200 | 250 | 300 | 400 | 500 | 600 | Diam. of Shaft, in. | Revolutions per minute 100 | 150 | 200 | 250 | 300 | 400 | 500 | 600 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Horsepower | | | | | | | | | Horsepower | | | | | | | |
| 1 1/2 | 4.8 | 7.2 | 9.6 | 12.0 | 14.4 | 19.2 | 24 | 29 | 2 13/16 | 32 | 47 | 63 | 79 | 95 | 127 | 159 | 190 |
| 1 9/16 | 5.5 | 8.2 | 10.9 | 13.6 | 16.4 | 22 | 27 | 33 | 2 7/8 | 34 | 51 | 68 | 85 | 101 | 135 | 169 | 203 |
| 1 5/8 | 6.1 | 9.2 | 12.2 | 15.3 | 18.4 | 24 | 31 | 37 | 2 15/16 | 36 | 54 | 72 | 90 | 108 | 144 | 181 | 217 |
| 1 11/16 | 6.9 | 10.3 | 13.7 | 17.1 | 21 | 27 | 34 | 41 | 3 | 39 | 58 | 77 | 96 | 116 | 154 | 192 | 231 |
| 1 3/4 | 7.7 | 11.5 | 15.3 | 19.1 | 23 | 31 | 38 | 46 | 3 1/8 | 44 | 65 | 87 | 109 | 131 | 174 | 218 | 261 |
| 1 13/16 | 8.5 | 12.7 | 16.9 | 21 | 25 | 34 | 42 | 51 | 3 1/4 | 49 | 73 | 98 | 122 | 147 | 196 | 245 | 294 |
| 1 7/8 | 9.4 | 14.1 | 18.8 | 23 | 28 | 38 | 47 | 57 | 3 3/8 | 55 | 83 | 110 | 137 | 165 | 220 | 275 | 330 |
| 1 15/16 | 10.4 | 15.6 | 21 | 26 | 31 | 42 | 52 | 62 | 3 1/2 | 61 | 92 | 123 | 153 | 184 | 245 | 307 | 367 |
| 2 | 11.4 | 17.2 | 23 | 29 | 34 | 46 | 57 | 69 | 3 5/8 | 68 | 102 | 136 | 170 | 204 | 272 | 340 | 408 |
| 2 1/16 | 12.6 | 18.9 | 25 | 31 | 38 | 50 | 63 | 76 | 3 3/4 | 75 | 113 | 151 | 189 | 226 | 301 | 377 | 452 |
| 2 1/8 | 13.7 | 21 | 27 | 34 | 41 | 55 | 68 | 82 | 3 7/8 | 83 | 125 | 166 | 207 | 249 | 332 | 415 | 498 |
| 2 3/16 | 15.0 | 22 | 30 | 37 | 45 | 60 | 75 | 90 | 4 | 92 | 137 | 183 | 229 | 274 | 366 | 457 | 549 |
| 2 1/4 | 16.3 | 24 | 33 | 41 | 49 | 65 | 81 | 98 | 4 1/8 | 101 | 150 | 201 | 251 | 300 | 401 | 501 | 601 |
| 2 5/16 | 17.7 | 27 | 35 | 44 | 53 | 71 | 88 | 106 | 4 1/4 | 110 | 164 | 219 | 273 | 328 | 438 | 547 | 657 |
| 2 3/8 | 19.2 | 29 | 38 | 48 | 57 | 76 | 96 | 115 | 4 3/8 | 120 | 180 | 239 | 298 | 358 | 478 | 597 | 717 |
| 2 7/16 | 20 | 30 | 41 | 51 | 61 | 81 | 102 | 122 | 4 1/2 | 130 | 195 | 261 | 326 | 391 | 521 | 651 | 781 |
| 2 1/2 | 22 | 33 | 45 | 56 | 67 | 89 | 112 | 133 | 4 5/8 | 141 | 212 | 283 | 354 | 425 | 566 | 707 | 848 |
| 2 9/16 | 24 | 36 | 48 | 60 | 72 | 96 | 120 | 144 | 4 3/4 | 153 | 230 | 307 | 382 | 459 | 613 | 765 | 919 |
| 2 5/8 | 26 | 39 | 52 | 64 | 77 | 104 | 129 | 155 | 4 7/8 | 166 | 249 | 331 | 413 | 496 | 662 | 827 | 994 |
| 2 11/16 | 28 | 42 | 55 | 69 | 83 | 111 | 138 | 166 | 5 | 179 | 268 | 358 | 447 | 537 | 715 | 895 | 1074 |
| 2 3/4 | 30 | 44 | 59 | 74 | 89 | 119 | 148 | 178 | | | | | | | | | |

### Table 3.—Dimensions of Shaft Collars

(Dodge Mfg. Corp., Mishawaka, Ind.)

| Shaft Size, in. | Plain Solid Collars Diam., in. | Width, in. | Corner Radius, in.* | Safety Collars Diam. Solid Collar, in. | Diam. Split Collar, in. | Width, in. | Shaft Size, in. | Safety Collars Diam. Solid Collar, in. | Diam. Split Collar, in. | Width, in. |
|---|---|---|---|---|---|---|---|---|---|---|
| 3/4 | 1 3/8 | 9/16 | 1/16 | ........ | ........ | ........ | 2 15/16 | 4 15/16 | 5 7/16 | 1 7/8 |
| 7/8 | 1 1/2 | 9/16 | 1/16 | | | | 3 3/16 | 5 3/16 | 5 11/16 | 2 |
| 15/16 | 1 5/8 | 9/16 | 1/16 | 2 7/16 | 3 1/8 | 1 3/8 | 3 7/16 | 5 7/16 | 5 15/16 | 2 |
| 1 | 1 5/8 | 9/16 | 1/16 | | | | 3 11/16 | 5 15/16 | 6 5/8 | 2 1/4 |
| 1 1/16 | 1 3/4 | 9/16 | 1/16 | | | | 3 15/16 | 6 3/16 | 7 | 2 1/4 |
| 1 1/8 | 1 3/4 | 9/16 | 1/16 | | | | 4 3/16 | 7 3/8 | 7 3/4 | 3 1/4 |
| 1 3/16 | 2 | 3/4 | 3/32 | 2 7/16 | 3 1/8 | 1 3/8 | 4 7/16 | 7 5/8 | 8 | 3 1/4 |
| 1 1/4 | 2 | 3/4 | 3/32 | | | | 4 11/16 | 7 7/8 | 8 1/4 | 3 1/4 |
| 1 5/16 | 2 1/8 | 3/4 | 3/32 | | | | 4 15/16 | 8 1/8 | 8 1/2 | 3 1/4 |
| 1 3/8 | 2 1/8 | 3/4 | 3/32 | | | | 5 3/16 | 8 5/8 | 9 | 3 1/4 |
| 1 7/16 | 2 1/4 | 3/4 | 3/32 | 3 | 3 3/8 | 1 5/8 | 5 7/16 | 8 5/8 | 9 | 3 1/4 |
| 1 1/2 | 2 1/4 | 3/4 | 3/32 | | | | 5 11/16 | 9 1/8 | 9 1/2 | 3 1/4 |
| 1 5/8 | 2 1/2 | 3/4 | 3/32 | | | | 5 15/16 | 9 1/8 | 9 1/2 | 3 1/4 |
| 1 11/16 | 2 1/2 | 3/4 | 3/32 | 3 3/16 | 3 5/8 | 1 5/8 | 6 3/16 | 10 3/8 | 10 1/2 | 3 1/2 |
| 1 3/4 | 2 3/4 | 7/8 | 1/8 | | | | 6 7/16 | 10 3/8 | 10 1/2 | 3 1/2 |
| 1 7/8 | 2 3/4 | 7/8 | 1/8 | | | | 6 11/16 | 10 7/8 | 11 | 3 1/2 |
| 1 15/16 | 3 | 7/8 | 1/8 | 3 7/16 | 3 7/8 | 1 5/8 | 6 15/16 | 10 7/8 | 11 | 3 1/2 |
| 2 | 3 | 7/8 | 1/8 | | | | 7 1/2 | 11 3/8 | 11 5/8 | 3 1/2 |
| 2 3/16 | 3 1/4 | 7/8 | 1/8 | 4 1/16 | 4 11/16 | 1 7/8 | 8 | 11 7/8 | 12 | 3 1/2 |
| 2 1/4 | 3 1/4 | 7/8 | 1/8 | | | | 8 1/2 | 13 | 13 | 3 3/4 |
| 2 7/16 | 3 1/2 | 1 | 5/32 | 4 5/16 | 4 15/16 | 1 7/8 | 9 | 13 1/2 | 13 1/2 | 3 3/4 |
| 2 1/2 | 3 1/2 | 1 | 5/32 | | | | 9 1/2 | 14 | 14 | 3 3/4 |
| 2 11/16 | ....... | ....... | ....... | 4 9/16 | 5 3/16 | 1 7/8 | 10 | 14 1/2 | 14 1/2 | 3 3/4 |

* Radius on one corner only.

transverse elasticity of 26,000,000, the following equations determine length between bearings to avoid excessive deflection.

For bare shafting, $L = \sqrt[3]{873\,d^2};\ d = \sqrt{L^3/873}$ . . . . . . . [8]

For shafting carrying pulleys, etc., $L = \sqrt[3]{175\,d^2};\ d = \sqrt{L^3/175}$ . . . . . . . [9]

In the foregoing equations $L$ = maximum distance, ft., between bearings for continuous shafting subjected to bending stress only; $d$ = diameter of shafting, in.

**Torsional Deflection.**—Torsional stress is inversely proportional to the velocity of rotation, but bending stress will not be reduced in the same ratio. It, therefore, is not practicable to write an equation covering the whole problem which is sufficiently simple for practical use. The following rules, however, are correct within the range of velocities that usually occur under shop conditions. For continuous shafting, so proportioned as to deflect not more than 0.01 in. per foot of length, allowance being made for the weakening effect of keyseats,

For bare shafts, $d = \sqrt[3]{50\ \text{Hp.}/N};\ L = \sqrt[3]{720\,d^2}$ . . . . . . [10]

For shafts carrying pulleys, etc., $d = \sqrt[3]{70\ \text{Hp.}/N};\ L = \sqrt[3]{140\,d^2}$ . . . . . . [11]

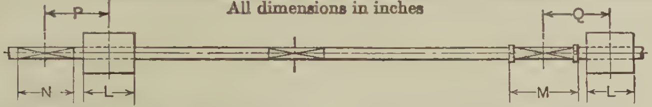
For a more complete discussion of the stresses in shafts, see p. 11–04.

**FLEXIBLE SHAFTING** consists of an inner core of built-up wire coils, brazed or soldered into end fittings, surrounded by a protecting casing. Table 5 gives the horsepower transmitted by flexible shafting at various speeds.

**SHAFT HANGERS** are made of cast iron or pressed steel in a variety of forms by different manufacturers. The frames will accommodate various types of hanger bearings, as ball-and-socket bearings, ball or roller bearings, ring or capillary oiling bearings and plain babbitted bearings. Hangers usually have adjusting screws to adjust the bearing

**Table 4.—Dimensions for Laying Out Shafting with Compression Couplings**

All dimensions in inches



| Shaft Size | Coupling Length, $L$ | Without Collars | | | | | | | | With Collars | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Plain-oiling Babbitted Box | | Ring-oiling Box | | Roller Bearing | | Ball Bearing | | Plain-oiling Babbitted Box | | Ring-oiling Box | | Roller Bearing | |
| | | $N$ | $P$ | $N$ | $P$ | $N$ | $P$ | $N$ | $P$ | $M$ | $Q$ | $M$ | $Q$ | $M$ | $Q$ |
| 15/16 | 3 | 3 3/4 | 4 1/8 | ..... | ....... | ..... | ....... | 3 11/16 | 4 1/8 | 6 5/8 | 5 5/16 | ..... | ....... | ..... | ....... |
| 1 3/16 | 3 3/4 | 4 3/4 | 4 7/8 | 6 | 5 1/2 | ..... | ....... | 3 7/8 | 4 7/16 | 7 5/8 | 6 5/16 | 8 7/8 | 6 15/16 | ..... | ....... |
| 1 7/16 | 4 1/2 | 5 3/4 | 5 7/8 | 7 1/4 | 6 5/8 | 5 3/8 | 5 11/16 | 4 1/2 | 5 1/4 | 9 1/8 | 7 3/16 | 10 5/8 | 8 5/16 | 7 1/4 | 6 5/8 |
| 1 11/16 | 5 1/4 | 6 3/4 | 7 1/2 | 8 1/2 | 8 3/8 | 5 3/4 | 7 | 4 3/4 | 6 1/2 | 10 1/8 | 9 3/16 | 11 7/8 | 10 1/16 | 7 1/2 | 7 7/8 |
| 1 15/16 | 6 | 7 1/2 | 8 11/16 | 9 3/4 | 9 13/16 | 6 1/4 | 8 1/16 | 5 1/8 | 7 1/2 | 10 7/8 | 10 3/8 | 13 1/8 | 11 1/2 | 8 1/4 | 9 1/16 |
| 2 3/16 | 6 3/4 | 8 3/4 | 9 7/8 | 11 | 11 | 6 7/8 | 9 | 5 1/2 | 8 1/4 | 12 5/8 | 11 13/16 | 14 7/8 | 12 15/16 | 8 7/8 | 10 1/16 |
| 2 7/16 | 7 1/2 | 9 5/8 | 10 13/16 | 12 1/4 | 12 1/8 | 7 5/8 | 9 13/16 | 6 1/2 | 9 1/4 | 13 1/2 | 13 3/4 | 16 1/8 | 14 1/16 | 9 5/8 | 10 13/16 |
| 2 15/16 | 9 | 11 7/8 | 12 15/16 | 14 3/8 | 14 3/8 | 8 1/4 | 11 1/8 | 7 1/8 | 10 9/16 | 16 | 15 | 18 1/2 | 16 1/4 | 10 3/8 | 12 3/16 |
| 3 7/16 | 12 5/8 | 13 9/16 | 15 7/16 | 17 1/4 | 17 5/16 | 9 1/4 | 13 9/16 | 7 3/4 | 12 13/16 | 18 | 17 15/16 | 21 5/8 | 20 | 11 1/4 | 14 5/8 |
| 3 15/16 | 13 1/8 | 15 5/8 | 17 1/16 | 20 | 19 1/4 | 13 1/4 | 15 7/8 | 8 3/8 | 13 15/16 | 20 5/8 | 19 9/16 | 25 | 21 3/4 | 16 1/2 | 17 1/2 |
| 4 7/16 | 14 1/8 | 17 3/8 | 18 1/2 | 18 | 18 13/16 | ..... | ....... | 9 3/4 | 14 11/16 | 24 3/8 | 22 | 25 | 22 5/16 | ..... | ....... |
| 4 15/16 | 16 1/8 | 19 1/8 | 20 3/8 | 20 | 20 13/16 | ..... | ....... | 10 1/4 | 15 15/16 | 26 1/8 | 23 7/8 | 27 | 24 5/16 | ..... | ....... |

**Table 5.—Horsepower Transmitted by Flexible Shafting**

(Stow Mfg. Co., Binghamton, N. Y.)

| Core Diameter, in. | Core Length, in. | Maximum Allowable Torque, lb.-ft. | Safe Horsepower Transmitted | | | |
|---|---|---|---|---|---|---|
| | | | 100 r.p.m. | 1000 r.p.m. | 1750 r.p.m. | 3500 r.p.m. |
| 0.130 | ........ | 0.219 | 0.00105 | 0.0105 | 0.01825 | 0.0366 |
| .130 | ........ | .151 | .00072 | .0072 | .0126 | .0252 |
| .150 | ........ | .260 | .00124 | .0124 | .0217 | .0434 |
| .150 | ........ | .188 | .00091 | .0091 | .0159 | .0318 |
| 1/4 | 31 | 2 1/4 | .0107 | .107 | .1873 | .3745 |
| 5/16 | 36 | 3 | .0142 | .142 | .2485 | .497 |
| 3/8 | 41 | 5 | .0238 | .238 | .4165 | .833 |
| 1/2 | 51 1/4 | 11 | .0523 | .523 | .9153 | 1.831 |
| 5/8 | 62 | 18 | .0856 | .856 | 1.498 | 2.996 |
| 7/8 | 74 | 32 | .1523 | 1.523 | 2.665 | 5.330 |
| 1 | 84 5/8 | 46 | .2189 | 2.189 | 3.83 | 7.66 |
| 1 1/4 | 86 1/4 | 60 | .2856 | 2.856 | 4.998 | 9.996 |
| 1 3/8 | 86 1/4 | 96 | .4569 | 4.569 | 7.996 | 15.991 |
| 1 5/8 | 108 | 140 | .6664 | 6.664 | 11.662 | 23.324 |

**Table 6.—Dimensions of Four-point Adjustable Cast-iron Shaft Hangers**

(Dodge Mfg. Corp., Mishawaka, Ind.)

| Shaft Size, in. | Dimensions, in. See Fig. 1 | | | | | | | | Bolts | | Weight, lb.* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | A | B | C | D | E | F | G | H | No. | Size, in. | |
| 15/16 | 6–8 | 13 | 3 3/4 | 9 1/4 | ....... | 7/8 | 4 3/8 | 3 1/4 | 2 | 1/2 | 13 |
| | 10–12 | 15 1/2 | 4 | 12 | ....... | 7/8 | 4 3/8 | 3 1/4 | 2 | 1/2 | 15 |
| | 14–16 | 18 1/8 | 4 7/8 | 14 1/4 | ....... | 7/8 | 4 3/8 | 3 1/4 | 2 | 1/2 | 19 |
| | 18–20 | 20 3/4 | 5 | 17 1/8 | ....... | 7/8 | 4 3/8 | 3 1/4 | 2 | 1/2 | 21 |
| 1 3/16 and 1 7/16 | 6–8 | 13 1/2 | 4 | 10 | ....... | 1 | 4 7/8 | 3 3/4 | 2 | 5/8 | 14 |
| | 10–12 | 16 | 4 | 12 1/2 | ....... | 1 | 4 7/8 | 3 3/4 | 2 | 5/8 | 17 |
| | 14–16 | 18 1/2 | 5 | 15 | ....... | 1 | 4 7/8 | 3 3/4 | 2 | 5/8 | 20 |
| | 18–20 | 20 3/4 | 5 | 17 1/2 | ....... | 1 | 4 7/8 | 3 3/4 | 2 | 5/8 | 24 |
| | 22–24 | 23 3/8 | 5 | 20 | ....... | 1 | 4 7/8 | 3 3/4 | 2 | 5/8 | 27 |
| 1 11/16 and 1 15/16 | 6–8 | 14 3/4 | 5 1/2 | 10 3/4 | ....... | 1 | 5 3/4 | 4 5/8 | 2 | 5/8 | 21 |
| | 10–12 | 16 1/2 | 5 1/2 | 12 3/4 | ....... | 1 | 5 3/4 | 4 5/8 | 2 | 5/8 | 26 |
| | 14–16 | 18 1/2 | 5 1/2 | 14 3/4 | ....... | 1 | 5 3/4 | 4 5/8 | 2 | 5/8 | 30 |
| | 18–20 | 20 1/2 | 6 1/4 | 16 1/2 | ....... | 1 | 5 3/4 | 4 5/8 | 2 | 5/8 | 34 |
| | 22–24 | 22 3/4 | 6 1/4 | 18 3/4 | ....... | 1 | 5 3/4 | 4 5/8 | 2 | 5/8 | 38 |
| | 26–28 | 24 1/4 | 6 1/4 | 20 1/4 | 3 1/4 | 1 | 5 3/4 | 4 5/8 | 4 | 5/8 | 41 |
| | 30–32 | 27 3/4 | 6 1/4 | 23 1/2 | 3 1/4 | 1 | 5 3/4 | 4 5/8 | 4 | 5/8 | 50 |
| 2 3/16 and 2 7/16 | 6–8 | 16 1/2 | 6 1/2 | 12 | 2 3/4 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 28 |
| | 10–12 | 18 1/2 | 6 1/2 | 13 3/4 | 2 3/4 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 32 |
| | 14–16 | 20 1/2 | 6 1/2 | 15 3/4 | 2 3/4 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 36 |
| | 18–20 | 22 3/8 | 8 1/2 | 17 3/4 | 4 5/8 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 42 |
| | 22–24 | 24 1/2 | 8 1/2 | 19 3/4 | 4 5/8 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 49 |
| | 26–28 | 26 1/2 | 8 1/2 | 21 3/4 | 4 5/8 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 52 |
| | 30–32 | 28 | 8 1/2 | 23 3/4 | 4 5/8 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 54 |
| | 34–36 | 30 1/4 | 8 1/2 | 25 3/4 | 4 5/8 | 1 1/8 | 6 1/2 | 5 3/8 | 4 | 3/4 | 60 |
| 2 11/16 and 2 15/16 | 6–8 | 17 | 7 | 13 | 3 3/8 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 39 |
| | 10–12 | 20 | 7 | 15 1/4 | 3 3/8 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 43 |
| | 14–16 | 22 3/8 | 7 | 17 3/4 | 3 3/8 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 47 |
| | 18–20 | 24 5/8 | 7 | 19 3/4 | 3 3/8 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 53 |
| | 22–24 | 27 5/8 | 8 3/4 | 22 3/4 | 4 3/4 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 65 |
| | 26–28 | 29 7/8 | 8 3/4 | 25 1/8 | 4 3/4 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 68 |
| | 30–32 | 32 | 8 3/4 | 27 | 4 3/4 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 77 |
| | 34–36 | 34 3/4 | 8 3/4 | 29 1/2 | 4 3/4 | 1 1/8 | 7 1/4 | 6 1/8 | 4 | 3/4 | 83 |
| 3 3/16 and 3 7/16 | 10–12 | 21 1/4 | 8 1/4 | 15 3/4 | 4 3/8 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 51 |
| | 14–16 | 23 3/8 | 8 1/4 | 18 | 4 3/8 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 60 |
| | 18–20 | 25 1/2 | 8 1/4 | 20 | 4 3/8 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 65 |
| | 22–24 | 27 1/2 | 9 3/4 | 22 1/4 | 6 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 75 |
| | 26–28 | 29 7/8 | 9 3/4 | 25 1/4 | 6 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 82 |
| | 30–32 | 32 | 9 3/4 | 27 | 6 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 86 |
| | 34–36 | 34 1/4 | 9 3/4 | 29 5/8 | 6 | 1 5/16 | 7 3/4 | 6 3/4 | 4 | 3/4 | 93 |

**Table 7.—Dimensions of Four-point Adjustable Pressed Steel Shaft Hangers**

(American Pulley Co., Philadelphia, Pa.)

| Shaft Size, in. | Dimensions, in. See Fig. 1 | | | | | | | Bolts | | Weight, lb.* |
|---|---|---|---|---|---|---|---|---|---|---|
| | A | B | C | D | F | G | H | No. | Size, in. | |
| 0 to 1 1/2 | 6–8 | 14 5/8 | 4 1/4 | 11 1/8 | 7/8 | 3 3/4 | 5 3/8 | 2 | 5/8 | 8.8 |
| | 10–12 | 16 7/16 | 4 1/2 | 12 9/16 | 7/8 | 3 3/4 | 5 3/8 | 2 | 5/8 | 10.0 |
| | 14–16 | 17 9/16 | 4 1/2 | 13 5/8 | 7/8 | 3 3/4 | 5 3/8 | 2 | 5/8 | 11.8 |
| | 18–20 | 19 3/8 | 5 5/16 | 15 1/8 | 7/8 | 3 3/4 | 5 3/8 | 2 | 5/8 | 14.0 |
| 1 9/16 to 2 | 7–9 | 16 3/4 | 4 1/2 | 12 7/8 | 7/8 | 4 1/2 | 5 7/8 | 2 | 5/8 | 11.8 |
| | 10–12 | 17 9/16 | 4 1/2 | 13 11/16 | 7/8 | 4 1/2 | 5 7/8 | 2 | 5/8 | 13.2 |
| | 14–16 | 19 7/16 | 5 5/16 | 15 3/16 | 7/8 | 4 1/2 | 5 7/8 | 2 | 5/8 | 16.0 |
| | 18–20 | 20 1/2 | 5 5/16 | 16 1/4 | 7/8 | 4 1/2 | 5 7/8 | 2 | 5/8 | 18.2 |
| 2 1/16 to 2 1/2 | 8–10 | 20 7/16 | 5 1/2 | 15 13/16 | 1 1/8 | 5 1/4 | 6 7/8 | 2 | 3/4 | 23.0 |
| | 10–12 | 21 1/8 | 5 1/2 | 16 1/2 | 1 1/8 | 5 1/4 | 6 7/8 | 2 | 3/4 | 23.8 |
| | 14–16 | 23 1/8 | 6 | 18 3/8 | 1 3/16 | 5 1/4 | 6 7/8 | 2 | 3/4 | 28.8 |
| | 18–20 | 24 7/16 | 6 | 19 11/16 | 1 3/16 | 5 1/4 | 6 7/8 | 2 | 3/4 | 31.8 |
| | 22–24 | 25 3/4 | 6 | 21 | 1 3/16 | 5 1/4 | 6 7/8 | 2 | 3/4 | 33.8 |
| 2 9/16 to 3 | 8–10 | 21 3/16 | 5 1/2 | 16 9/16 | 1 1/8 | 6 | 7 1/2 | 2 | 1 | 24.5 |
| | 10–12 | 21 7/8 | 5 1/2 | 17 1/4 | 1 1/8 | 6 | 7 1/2 | 2 | 1 | 26.0 |
| | 14–16 | 23 7/8 | 6 | 19 1/8 | 1 3/16 | 6 | 7 1/2 | 2 | 1 | 29.0 |
| | 18–20 | 25 3/16 | 6 | 20 7/16 | 1 3/16 | 6 | 7 1/2 | 2 | 1 | 34.5 |
| | 22–24 | 26 1/2 | 6 | 21 3/4 | 1 3/16 | 6 | 7 1/2 | 2 | 1 | 40.0 |

* Frame only, without bearings.

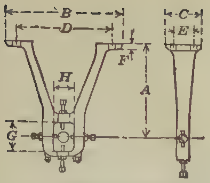


Fig. 1. Shaft Hanger Dimensions. See Tables 6 and 7

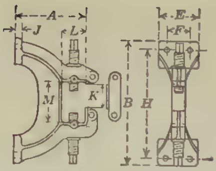
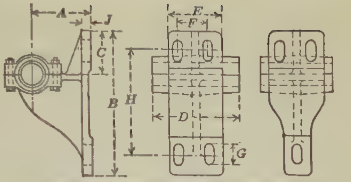


Fig. 2. Adjustable Ball-and-Socket Post Hanger



Ring Oiling Bearings 1 7/16-4 15/16 in. Babbitted Bearings over 3 in.

Babbitted Bearings up to 3-in. shaft

Fig. 3. Rigid Post Bearing

in a vertical plane, to conform to the desired shaft height. Tables 6 and 7 give the dimensions of typical commercial cast-iron and pressed-steel hangers. The length of the bearing varies with the type. Lengths of the various types which may be used with the hangers of Table 6 are given in Table 4.

**Post Hangers** are used when shafting is to be mounted on building columns. Table 8 gives dimensions of one type of adjustable ball-and-socket post hanger. Table 9 gives the dimensions of rigid post bearings.

### Table 8.—Dimensions of Adjustable Ball-and-Socket Post Hangers

(Dodge Mfg. Corp., Mishawaka, Ind.)

| Shaft Size, in. | Dimensions, in. See Fig. 2 | | | | | | | | | Bolts | | Weight, lb.* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | A | B | E | F | H | J | K | L | M | No. | Size, in. | |
| 15/16 | 6 | 12 3/4 | 3 1/2 | ...... | 11 1/4 | 5/8 | 2 1/2 | 2 1/2 | 4 1/4 | 2 | 5/8 | 14 |
| 1 3/16 | 6 | 12 3/4 | 3 1/2 | ...... | 11 1/4 | 5/8 | 2 1/2 | 2 1/2 | 4 1/4 | 2 | 5/8 | 14 |
| 1 7/16 | 6 | 15 | 4 | ...... | 13 1/8 | 5/8 | 3 | 3 | 4 7/8 | 2 | 5/8 | 18 |
| 1 11/16 | 6 | 15 | 4 | ...... | 13 1/8 | 5/8 | 3 | 3 | 4 7/8 | 2 | 5/8 | 18 |
| 1 15/16 | 6 | 17 1/2 | 4 3/8 | ...... | 15 1/2 | 3/4 | 3 1/8 | 3 3/4 | 5 1/8 | 2 | 5/8 | 28 |
| 2 3/16 | 6 1/8 | 19 1/2 | 5 1/4 | 2 3/8 | 15 1/2 | 7/8 | 3 5/8 | 4 1/8 | 6 1/2 | 4 | 5/8 | 38 |
| 2 7/16 | 6 1/8 | 19 1/2 | 5 1/4 | 2 3/8 | 15 1/2 | 7/8 | 3 5/8 | 4 1/8 | 6 1/2 | 4 | 5/8 | 38 |
| 2 11/16 | 6 1/2 | 20 5/8 | 6 | 3 1/8 | 16 5/8 | 3/4 | 4 3/8 | 5 | 7 1/2 | 4 | 3/4 | 45 |
| 2 15/16 | 6 1/2 | 20 5/8 | 6 | 3 1/8 | 16 5/8 | 3/4 | 4 3/8 | 5 | 7 1/2 | 4 | 3/4 | 45 |
| 3 3/16 | 8 | 21 | 7 | 3 3/4 | 17 | 7/8 | 5 1/8 | 5 3/4 | 8 1/4 | 4 | 3/4 | 66 |
| 3 7/16 | 8 | 21 | 7 | 3 3/4 | 17 | 7/8 | 5 1/8 | 5 3/4 | 8 1/4 | 4 | 3/4 | 66 |
| 3 11/16 | 8 1/4 | 21 1/2 | 8 | 4 | 17 1/8 | 1 | 5 1/4 | 6 3/8 | 9 | 4 | 7/8 | 68 |
| 3 15/16 | 8 1/4 | 21 1/2 | 8 | 4 | 17 1/8 | 1 | 5 1/4 | 6 3/8 | 9 | 4 | 7/8 | 68 |
| 4 7/16 | 8 1/4 | 25 1/2 | 9 | 5 | 20 3/8 | 1 | 5 1/2 | 7 | 9 3/8 | 4 | 7/8 | 104 |
| 4 15/16 | 8 3/4 | 27 | 10 | 5 3/4 | 21 3/8 | 1 1/8 | 6 | 8 | 10 | 4 | 1 | 140 |

* Frame only, without bearings.

### Table 9.—Dimensions of Rigid Post Bearings

(Dodge Mfg. Corp., Mishawaka, Ind.)

| Shaft Size, in. | Dimensions, in. See Fig. 3 | | | | | | | | | Bolts | | Weight, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | A | B | C | D | E | F | G | H | J | No. | Size, in. | |
| | Plain Babbitted Bearings | | | | | | | | | | | |
| 1 3/16 | 6 | 9 7/8 | 3 | 3 3/4 | 3 1/2 | 1 3/4 | 1 1/4 | 7 | 9/16 | 3 | 1/2 | 10 |
| 1 7/16 | 6 | 11 1/8 | 3 1/4 | 4 1/2 | 4 | 2 | 1 1/4 | 8 1/8 | 5/8 | 3 | 1/2 | 13 |
| 1 11/16 | 6 | 12 1/2 | 3 3/4 | 5 1/4 | 4 3/8 | 2 3/8 | 1 1/2 | 9 1/8 | 11/16 | 3 | 1/2 | 16 |
| 1 15/16 | 6 | 13 3/4 | 4 | 6 | 4 7/8 | 2 3/4 | 1 3/4 | 10 1/4 | 3/4 | 3 | 5/8 | 24 |
| 2 3/16 | 6 | 15 1/8 | 4 1/2 | 6 3/4 | 5 3/8 | 3 1/8 | 2 | 11 3/8 | 13/16 | 3 | 5/8 | 28 |
| 2 7/16 | 6 | 16 1/4 | 4 5/8 | 7 1/2 | 5 3/4 | 3 1/2 | 2 1/4 | 12 1/8 | 7/8 | 3 | 3/4 | 35 |
| 2 11/16 | 6 | 17 5/8 | 5 1/8 | 8 1/4 | 6 1/4 | 3 3/4 | 2 1/4 | 13 1/2 | 15/16 | 3 | 3/4 | 46 |
| 2 15/16 | 6 | 18 7/8 | 5 3/8 | 9 | 6 3/4 | 4 1/8 | 2 1/2 | 14 1/2 | 1 | 3 | 3/4 | 56 |
| 3 7/16 | 6 | 22 | 6 | 10 1/2 | 8 1/2 | 4 1/2 | 2 1/4 | 17 1/4 | 7/8 | 4 | 3/4 | 100 |
| 3 15/16 | 6 | 24 | 7 | 12 | 10 1/8 | 5 1/2 | 2 3/4 | 18 1/4 | 1 | 4 | 1 | 139 |
| 4 7/16 | 6 | 24 | 6 | 13 1/2 | 11 | 6 | 2 3/4 | 18 1/4 | 1 | 4 | 1 | 142 |
| 4 15/16 | 6 | 25 | 6 3/4 | 15 | 11 | 6 | 2 7/8 | 19 1/2 | 1 3/8 | 4 | 1 1/8 | 179 |
| | Ring and Capillary Oiling Bearings | | | | | | | | | | | |
| 1 7/16 | 6 | 11 | 2 1/2 | 6 | 3 7/8 | 2 1/2 | 1 3/8 | 8 5/8 | 3/4 | 4 | 1/2 | 20 |
| 1 11/16 | 6 | 12 | 3 | 6 3/4 | 4 1/2 | 2 3/4 | 1 1/2 | 9 1/4 | 3/4 | 4 | 1/2 | 25 |
| 1 15/16 | 6 | 12 | 3 5/8 | 7 1/2 | 5 1/8 | 3 | 1 3/4 | 9 | 1 | 4 | 5/8 | 32 |
| 2 3/16 | 6 | 13 1/2 | 4 | 8 1/4 | 5 1/4 | 3 1/8 | 1 7/8 | 10 1/8 | 1 | 4 | 5/8 | 42 |
| 2 7/16 | 6 | 15 | 4 5/8 | 9 | 5 1/2 | 3 1/4 | 1 7/8 | 10 7/8 | 1 | 4 | 5/8 | 55 |
| 2 11/16 | 6 | 16 1/4 | 4 3/4 | 9 7/8 | 6 | 3 1/2 | 2 1/8 | 12 1/4 | 1 1/8 | 4 | 3/4 | 75 |
| 2 15/16 | 6 | 17 1/2 | 5 | 10 5/8 | 6 1/4 | 3 3/4 | 2 1/4 | 13 1/2 | 1 1/8 | 4 | 3/4 | 90 |
| 3 7/16 | 6 | 20 | 6 | 12 | 7 | 4 3/8 | 2 1/4 | 16 | 1 1/4 | 4 | 3/4 | 113 |
| 3 15/16 | 6 | 23 3/4 | 6 3/4 | 13 1/2 | 8 1/2 | 5 1/2 | 2 7/8 | 18 3/4 | 1 1/4 | 4 | 1 | 150 |
| 4 7/16 | 6 | 24 | 6 | 16 1/2 | 9 1/4 | 6 | 2 7/8 | 18 1/4 | 1 3/8 | 4 | 1 | 185 |
| 4 15/16 | 6 | 25 | 6 3/4 | 15 | 11 | 6 | 2 7/8 | 19 1/2 | 1 3/8 | 4 | 1 1/8 | 219 |

**DISTANCE BETWEEN BEARINGS.**—The spacing of hangers for line-shafting depends on the size of shaft and the points at which load is applied. The following figures represent average practice with pulleys placed near the bearings:

| Shaft size, in.... | $^{15}/_{16}$–1 $^{7}/_{16}$ | 1 $^{11}/_{16}$–2 $^{3}/_{16}$ | 2 $^{7}/_{16}$–2 $^{15}/_{16}$ | 3 $^{7}/_{16}$–3 $^{15}/_{16}$ | 4 $^{7}/_{16}$–4 $^{15}/_{16}$ |
|---|---|---|---|---|---|
| Distance between bearings, ft.... | 6 | 8 | 10 | 12 | 14 |

# PULLEYS

## 1. PROPORTIONS OF PULLEYS

**PROPORTIONS OF CAST-IRON PULLEYS** (See also Fly-wheels, p. 11–15).—Pulleys for the transmission of power have been standardized by manufacturers and stock pulleys are available for practically all conditions. The design of pulleys, except for special conditions is not advisable. It usually will prove more economical to consult manufacturers of pulleys for the adaptation of a stock pulley to meet the special conditions.

Text-books on machine design give a number of equations for the proportions of cast-iron pulleys, which give slightly different dimensions. The dimensions of stock pulleys vary but little from the values given by the different equations. The equations most frequently used are those of Unwin, Releaux, and J. W. See. These equations follow.

Let $n$ = number of arms of pulley; $D$ = diameter of pulley; $S$ = thickness of belt; $t$ = thickness of rim at edge; $T$ = thickness of rim at middle; $F$ = width of rim; $B$ = width of belt; $w_h$ = width of arm at hub; $w_r$ = width of arm at rim; $b_h$ = thickness of arm at hub; $b_r$ = thickness of arm at rim; $c$ = amount of crowning; $L$ = length of hub; $M$ = thickness of metal in hub. All dimensions are inches.

Width of rim (Unwin) $F = {}^{9}/_{8}\,(B + 0.4)$ . . . . [1a]

(Releaux) $F = {}^{9}/_{8}\,B$ to ${}^{5}/_{4}\,B$ . . . . [1b]

Thickness of rim at edge, (Unwin) $t = 0.7\,S + 0.005\,D$ . . . [2a]

(Releaux) $t = {}^{1}/_{5}\,w_h$ to ${}^{1}/_{4}\,w_h$ . . . [2b]

Thickness at middle of rim, (Unwin) $T = 2\,t + c$ . . . . . . [3]

Width of arm at hub, (Unwin, single belts) $w_h = 0.6337\,\sqrt[3]{FD/n}$ . . . [4a]

(Unwin, double belts) $w_h = 0.798\,\sqrt[3]{FD/n}$ . . . [4b]

(Releaux) $w_h = (F/4) + (D/20n) + {}^{1}/_{4}$ . [4c]

(See) $w_h = 0.0625\,D + 0.5$ . . . [4d]

Width of arm at rim, (Unwin) $w_r = {}^{2}/_{3}\,w_h$ . . . . . . . [5a]

(Releaux) $w_r = 0.8\,w_h$ . . . . . . . [5b]

(See) $w_r = 0.04\,D + 0.3125$ . . . [5c]

Thickness of arm at hub, (Unwin) $b_h = 0.4\,w_h$ . . . . . . . [6a]

(Releaux) $b_h = 0.5\,w_h$ . . . . . . . [6b]

(See) $b_h = 0.025\,D + 0.2$ . . . . [6c]

Thickness of arm at rim, (Unwin) $b_r = 0.4\,w_h$ . . . . . . . [7a]

(Releaux) $b_r = 0.5\,w_h$ . . . . . . . [7b]

(See) $b_r = 0.016\,D + 0.125$ . . . [7c]

Number of arms, (Unwin) $n = 3 + (FD/150)$ . . . [8a]

(Releaux) $n = {}^{1}/_{2}\{5 + (D/2F)\}$ . . . [8b]

Length of hub, (Unwin) $L = {}^{2}/_{3}\,F$ or $2.5\,S$, min. . . [9a]

(Releaux) $L = B$ for single arm pulleys

$= 2\,B$ for double arm pulleys [9b]

Thickness of metal in hub, (Releaux) $M = w_h$ to ${}^{3}/_{4}\,w_h$ . . . . [10a]

Crowning, (Unwin) $c = {}^{1}/_{24}\,F$ . . . . . . . [10b]

The number of arms is arbitrary and may be altered if necessary. Commercial stock pulleys usually are made with four arms up to 10 in. diameter, 6 arms from 11 to 74 in. diameter, and 8 arms for larger diameters. Pulleys with two or three sets of arms may be considered as two or three separate pulleys combined in one. Releaux states in such cases that the proportions of the arms should be 0.8 or 0.7 those of single-arm pulleys.

**Width of Pulley in Relation to Belt Width.**—An equation that is widely used for proportioning the face of pulleys in relation to the width of belt was published by Carl G. Barth (*Am. Mach.*, Feb. 11, 1915). If $F$ and $B$ = respectively, the width of pulley and belt, in.,

$$F = 1.1875\,B + 0.375 \qquad [11]$$

If limits of design make these proportions impractical, the following equation may be used:

$$F = 1.09375\,B + 0.1875 \qquad [12]$$

**Crowning of Pulleys** is necessary to hold the belt central on the face of the pulley, since a belt tends to climb to the largest diameter over which it runs. Different authorities give values for the height of crown $C$ ranging all the way from 0.1 $F$ to 0.011 $F$. Barth uses the equation

$$C = 0.03125\,F^{2/3} \qquad [13]$$

Notation is the same as in the preceding equations.

**STEEL SPLIT PULLEYS.**—Split pulleys made of pressed steel with the arms riveted to the circumference and to the hub are widely used for power transmission. They have the advantages of high strength and light weight. They are available in standard sizes from 3 in. diameter by 3 in. face to 144 in. diameter by 36 in. face, with bores ranging from 1 7/16 to 8 7/16 in. diameter. Pulleys up to 96 in. diameter are made with 6 arms. Larger pulleys have 8 arms.

**WOOD-FACE PULLEYS.**—Pulleys made entirely of wood, or cast-iron pulleys with a wood face, are used where lightness and high belt traction are desirable. Pulleys made entirely of wood are available from 3 in. diameter by 12 in. face to 120 in. diameter by 24 in. face, with bores ranging from 1 1/2 to 7 1/2 in. Their weight is approximately 1/3 that of cast-iron pulleys of similar dimensions.

Wood-faced pulleys are used with high surface speeds and heavy loads. The high coefficient of friction of the wood face insures a more uniform speed and greater tractive effort than is possible with a metal face pulley. They are made both solid and split in sizes from 8 in. diameter by 3 in. face to 72 in. diameter by 40 in. face.

**RELATIVE CAPACITY OF DIFFERENT TYPES OF PULLEYS.**—In an investigation to ascertain the value of compressed spruce pulleys, Prof. G. S. Wilson (Bulletin No. 10, Univ. of Wash. Engg. Expt. Station, 1920) found the horsepower capacity of several different types of pulleys to be as given in Table 1. In this investigation the slip of the belt was measured at belt tensions ranging from 37 1/2 to 187 1/2 lb. per sq. in. of belt section, and the horsepower determined by a Prony brake. The pulleys used in the investigation were 18 in. diam. and 6. in. face.

A paper by Lawrence Whitcomb (Nat'l Assoc. of Cotton Mfrs., 1909) gives the results of tests made at the Lowell Textile School on the relative transmitting capacity of wood-rim, wood-rim with cork inserts, wood-rim lagged with leather, and steel pulleys, all 18 in. diam. by 6 in. face. The results of the tests are summarized in Table 2.

**WEIGHT OF CAST-IRON PULLEYS.**—Table 3 gives the weight of solid cast-iron single-arm pulleys. The values given are approximate and will vary with changes in bore. For split pulleys, the figures in Table 3 should be increased by 10%, and for split-hub pulleys by 4%. The weights of multiple arm pulleys are approximately as follows:

**Table 1.—Horsepower Capacities per Inch Width of Belt of Various Pulleys**

| Pulley | Weight, lb. | Slip of Belt | | | Relative Capacity at 2% Slip |
|---|---|---|---|---|---|
| | | 1% | 1 1/2% | 2% | |
| Kind | | Horsepower | | | |
| Cast-iron | 65.0 | 0.58 | 0.69 | 0.74 | 100.0 |
| Steel | 40.0 | .71 | .78 | .81 | 109.4 |
| Wood split | 21.5 | .63 | .77 | .82 | 110.8 |
| Paper | 41.5 | .62 | .76 | .83 | 112.1 |
| Compressed spruce | 22.0 | .76 | .81 | .84 | 113.5 |

**Table 2.—Tests of Lagged and Cork Insert Pulleys**

| Type of Pulley | Hp. at 2% Slip | Maximum Hp. Transmitted | Slip, percent at Maximum Hp. | Relative Value at 2% Slip |
|---|---|---|---|---|
| Wood-rim pulley with cork inserts | 11.8 | 13.2 | 4.6 | 100% |
| Wood-rim pulley, leather lagged | 8.7 | 12.4 | 4.7 | 73% |
| Steel pulley | 8.3 | 12.7 | 6.0 | 70% |
| Wood-rim pulley | 7.8 | 9.6 | 4.3 | 65% |

Double arm pulley, 2.5 $F/2$; triple arm pulley, 3.75 $F/3$; quadruple arm pulley, 5 $F/4$, where $F$ = width of face of pulley.

**PULLEY WINDAGE.**—The loss of power due to windage of pulleys, *i.e.*, fanning of the air by the pulley arms, may be considerable in transmissions involving a large number of pulleys. The windage varies widely with different types of pulleys. Tests made at

### Table 3.—Weight of Solid Single-arm Cast-iron Pulleys

(Dodge Manufacturing Co., Mishawaka, Ind.)

| Diam., in. | Face, in. | | | | | | | | | | | | | | Rim Thickness, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 4 | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 30 | |
| **4 Arms** | | | | | | | | | | | | | | | |
| 6 | 11 | 15 | 19 | 25 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1/4 |
| 7 | 12 | 17 | 23 | 31 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | |
| 8 | 14 | 21 | 29 | 38 | 45 | 56 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5/16 |
| 9 | 15 | 24 | 33 | 44 | 54 | 62 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | |
| 10 | 17 | 25 | 36 | 45 | 61 | 69 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | |
| **6 Arms** | | | | | | | | | | | | | | | |
| 11 | 19 | 30 | 39 | 54 | 67 | 76 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5/16 |
| 12 | 21 | 32 | 43 | 65 | 76 | 86 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | |
| 13 | 23 | 34 | 54 | 69 | 79 | 95 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | |
| 14 | 25 | 37 | 57 | 73 | 87 | 101 | 108 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | |
| 15 | 27 | 45 | 66 | 77 | 96 | 104 | 132 | 159 | ..... | ..... | ..... | ..... | ..... | ..... | 3/8 |
| 16 | 33 | 52 | 69 | 81 | 102 | 117 | 139 | 171 | ..... | ..... | ..... | ..... | ..... | ..... | |
| 17 | 38 | 57 | 74 | 87 | 110 | 132 | 159 | 184 | ..... | ..... | ..... | ..... | ..... | ..... | |
| 18 | 41 | 60 | 76 | 91 | 114 | 138 | 171 | 197 | 215 | ..... | ..... | ..... | ..... | ..... | |
| 19 | 43 | 63 | 80 | 102 | 119 | 157 | 178 | 206 | 235 | ..... | ..... | ..... | ..... | ..... | |
| 20 | 47 | 69 | 84 | 109 | 127 | 167 | 195 | 214 | 250 | ..... | ..... | ..... | ..... | ..... | |
| 21 | 49 | 72 | 88 | 113 | 149 | 174 | 203 | 231 | 266 | ..... | ..... | ..... | ..... | ..... | |
| 22 | 50 | 75 | 93 | 115 | 156 | 187 | 211 | 250 | 276 | ..... | ..... | ..... | ..... | ..... | |
| 23 | 52 | 80 | 104 | 134 | 165 | 199 | 227 | 258 | 286 | ..... | ..... | ..... | ..... | ..... | |
| 24 | 60 | 83 | 108 | 142 | 172 | 206 | 235 | 273 | 302 | ..... | ..... | ..... | ..... | ..... | |
| 26 | 66 | 90 | 130 | 161 | 186 | 222 | 264 | 294 | 363 | ..... | ..... | ..... | ..... | ..... | 7/16 |
| 28 | 70 | 105 | 139 | 173 | 213 | 244 | 285 | 343 | 391 | ..... | ..... | ..... | ..... | ..... | |
| 30 | 84 | 113 | 149 | 184 | 226 | 274 | 341 | 383 | 405 | ..... | ..... | ..... | ..... | ..... | |
| 32 | 90 | 130 | 167 | 212 | 246 | 311 | 365 | 404 | 456 | 497 | ..... | ..... | ..... | ..... | |
| 34 | 99 | 139 | 177 | 224 | 292 | 342 | 396 | 433 | 485 | 521 | ..... | ..... | ..... | ..... | |
| 36 | 108 | 149 | 191 | 256 | 316 | 357 | 413 | 470 | 513 | 617 | ..... | ..... | ..... | ..... | 1/2 |
| 38 | 113 | 162 | 221 | 274 | 329 | 389 | 440 | 497 | 584 | 656 | ..... | ..... | ..... | ..... | |
| 40 | 119 | 178 | 243 | 287 | 349 | 403 | 473 | 555 | 628 | 683 | ..... | ..... | ..... | ..... | |
| 42 | 132 | 196 | 254 | 318 | 375 | 422 | 526 | 586 | 660 | 720 | ..... | ..... | ..... | ..... | |
| 44 | 141 | 206 | 265 | 331 | 390 | 492 | 553 | 628 | 686 | 765 | ..... | ..... | ..... | ..... | |
| 46 | 163 | 221 | 287 | 351 | 445 | 518 | 579 | 659 | 719 | 797 | ..... | ..... | ..... | ..... | |
| 48 | 169 | 230 | 300 | 395 | 465 | 543 | 611 | 695 | 776 | 840 | ..... | ..... | ..... | ..... | |
| 50 | 178 | 239 | 329 | 410 | 486 | 570 | 659 | 723 | 809 | 932 | ..... | ..... | ..... | ..... | 9/16 |
| 52 | 184 | 264 | 361 | 441 | 528 | 587 | 681 | 768 | 896 | 1000 | ..... | ..... | ..... | ..... | |
| 54 | 201 | 276 | 378 | 459 | 545 | 632 | 710 | 800 | 926 | 1033 | ..... | ..... | ..... | ..... | |
| 56 | 245 | 307 | 394 | 477 | 566 | 657 | 784 | 881 | 960 | 1083 | 1199 | 1273 | ..... | ..... | |
| 58 | 278 | 342 | 410 | 495 | 590 | 720 | 816 | 917 | 1030 | 1115 | 1234 | 1400 | ..... | ..... | 5/8 |
| 60 | 290 | 354 | 430 | 514 | 648 | 753 | 863 | 949 | 1064 | 1151 | 1345 | 1443 | ..... | ..... | |
| 62 | 297 | 371 | 443 | 593 | 671 | 779 | 895 | 983 | 1111 | 1304 | 1403 | 1547 | ..... | ..... | |
| 64 | ..... | 401 | 500 | 610 | 717 | 834 | 928 | 1054 | 1209 | 1346 | 1448 | 1589 | ..... | ..... | |
| 66 | ..... | 434 | 517 | 631 | 742 | 869 | 950 | 1149 | 1250 | 1389 | 1533 | 1641 | 1808 | ..... | |
| 68 | ..... | 474 | 563 | 652 | 761 | 893 | 1076 | 1179 | 1332 | 1438 | 1592 | 1699 | 1967 | ..... | 11/16 |
| 70 | ..... | 493 | 581 | 671 | 821 | 966 | 1104 | 1218 | 1366 | 1475 | 1627 | 1833 | 2018 | ..... | |
| 72 | ..... | 511 | 600 | 686 | 856 | 996 | 1144 | 1253 | 1402 | 1556 | 1772 | 1962 | 2089 | ..... | |
| 74 | ..... | ..... | 621 | 780 | 937 | 1075 | 1181 | 1342 | 1462 | 1709 | 1833 | 2014 | 2142 | ..... | |
| **8 Arms** | | | | | | | | | | | | | | | |
| 76 | ..... | ..... | 681 | 810 | 964 | 1107 | 1270 | 1392 | 1643 | 1768 | 1951 | 2081 | 2272 | 2406 | 11/16 |
| 78 | ..... | ..... | 728 | 832 | 996 | 1133 | 1307 | 1493 | 1681 | 1809 | 2017 | 2151 | 2350 | 2607 | 3/4 |
| 80 | ..... | ..... | 744 | 851 | 1016 | 1159 | 1404 | 1587 | 1730 | 1862 | 2058 | 2195 | 2510 | 2660 | |
| 82 | ..... | ..... | 763 | 872 | 1036 | 1261 | 1433 | 1632 | 1769 | 1965 | 2103 | 2415 | 2566 | 2794 | |
| 84 | ..... | ..... | 836 | 950 | 1150 | 1283 | 1477 | 1667 | 1800 | 2009 | 2268 | 2488 | 2643 | 2875 | |

Purdue Univ. to determine the windage loss of cast-iron pulleys and two different types of wood pulleys are reported in *Power*, May 1, 1917. The pulleys were 24 in. diam. by 6 in. face. Following are the horsepower losses due to windage at various speeds:

| R.P.M. | Cast-iron Pulley | Wood Pulley *A* | Wood Pulley *B* |
|---|---|---|---|
| 232 | 0.0042 | 0.0046 | 0.00345 |
| 323 | .0048 | .0120 | .0058 |
| 417 | .0078 | .02585 | .01535 |
| 508 | .0192 | .0443 | .02745 |

Steel pulleys usually have less windage loss than cast-iron pulleys.

**Table 4.—Weight of Standard Steel Split Pulleys**

(American Pulley Co., Philadelphia)

| Diam., in. | Bore, in. | Nominal Face, in. | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 2 | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 30 | 36 |
| | | Approximate Weight, lb., Without Bushings | | | | | | | | | | | | | | | | | |
| 3 | ...... | 1.2 | 1.5 | 1.8 | 2.8 | 3.1 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3 1/2 | ...... | 1.5 | 1.9 | 2.1 | 3.2 | 3.5 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 4 | ...... | 2.1 | 2.7 | 2.9 | 3.2 | 5.4 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 5 | ...... | 2.5 | 3.3 | 3.6 | 4.6 | 5.1 | 7.2 | 7.8 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 6 | ...... | 4.3 | 4.5 | 6.4 | 6.9 | 7.4 | 11.9 | 13.8 | 14.6 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 7 | ...... | 7 | 7 | 8 | 9 | 10 | 15 | 17 | 20 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 8 | ...... | 9 | 11 | 12 | 14 | 15 | 17 | 21 | 30 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 9 | ...... | 11 | 12 | 13 | 14 | 16 | 18 | 22 | 32 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 10 | ...... | 12 | 14 | 17 | 20 | 21 | 22 | 31 | 37 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 11 | ...... | 12 | 17 | 18 | 20 | 22 | 25 | 34 | 39 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 12 | ...... | 14 | 18 | 19 | 22 | 24 | 31 | 37 | 40 | 50 | 52 | 68 | .... | .... | .... | .... | .... | .... | .... |
| 13 | ...... | 15 | 18 | 19 | 24 | 25 | 33 | 40 | 41 | 52 | 54 | 72 | .... | .... | .... | .... | .... | .... | .... |
| 14 | ...... | 15 | 19 | 20 | 24 | 26 | 35 | 41 | 44 | 54 | 56 | 76 | .... | .... | .... | .... | .... | .... | .... |
| 15 | ...... | 16 | 20 | 22 | 26 | 27 | 37 | 43 | 46 | 56 | 58 | 80 | .... | .... | .... | .... | .... | .... | .... |
| 16 | ...... | 18 | 21 | 23 | 27 | 28 | 39 | 45 | 47 | 60 | 64 | 84 | .... | .... | .... | .... | .... | .... | .... |
| 17 | ...... | ... | 21 | 24 | 28 | 28 | 41 | 47 | 49 | 61 | 66 | 88 | .... | .... | .... | .... | .... | .... | .... |
| 18 | ...... | ... | 23 | 25 | 29 | 30 | 43 | 50 | 52 | 65 | 70 | 93 | .... | .... | .... | .... | .... | .... | .... |
| 19 | ...... | ... | 25 | 25 | 30 | 32 | 45 | 51 | 55 | 67 | 74 | 97 | .... | .... | .... | .... | .... | .... | .... |
| 20 | ...... | ... | 25 | 26 | 31 | 33 | 47 | 52 | 56 | 69 | 91 | 100 | .... | .... | .... | .... | .... | .... | .... |
| 21 | ...... | ... | 25 | 26 | 31 | 35 | 51 | 54 | 59 | 71 | 94 | 103 | .... | .... | .... | .... | .... | .... | .... |
| 22 | ...... | ... | 26 | 28 | 32 | 36 | 53 | 57 | 60 | 75 | 99 | 107 | .... | .... | .... | .... | .... | .... | .... |
| 23 | ...... | ... | 31 | 32 | 38 | 39 | 56 | 59 | 64 | 82 | 102 | 111 | .... | .... | .... | .... | .... | .... | .... |
| 24 | ...... | ... | 32 | 34 | 38 | 41 | 59 | 61 | 65 | 89 | 107 | 114 | 118 | .... | .... | .... | .... | .... | .... |
| 25 | 4 7/16 | ... | ... | 75 | 78 | 82 | 92 | 104 | 110 | 124 | 131 | 185 | 201 | .... | .... | .... | .... | .... | .... |
| 26 | 4 7/16 | ... | ... | 77 | 80 | 85 | 96 | 108 | 114 | 129 | 136 | 194 | 212 | .... | .... | .... | .... | .... | .... |
| 28 | 4 7/16 | ... | ... | 80 | 86 | 89 | 103 | 114 | 120 | 137 | 144 | 207 | 224 | .... | .... | .... | .... | .... | .... |
| 30 | 4 7/16 | ... | ... | 85 | 90 | 92 | 108 | 119 | 126 | 144 | 153 | 218 | 234 | 241 | 257 | 273 | 277 | 353 | .... |
| 32 | 4 7/16 | ... | ... | 92 | 95 | 102 | 112 | 125 | 143 | 152 | 160 | 226 | 244 | 253 | 279 | 292 | 296 | 367 | .... |
| 34 | 4 7/16 | ... | ... | 96 | 100 | 106 | 117 | 130 | 150 | 159 | 168 | 237 | 254 | 264 | 290 | 306 | 309 | 384 | .... |
| 36 | 4 7/16 | ... | ... | 100 | 103 | 110 | 121 | 136 | 156 | 166 | 175 | 246 | 266 | 276 | 303 | 319 | 322 | 398 | 438 |
| 38 | 4 7/16 | ... | ... | 104 | 106 | 115 | 126 | 142 | 163 | 173 | 183 | 255 | 278 | 288 | 318 | 332 | 338 | 415 | 457 |
| 40 | 4 7/16 | ... | ... | 108 | 112 | 119 | 131 | 147 | 170 | 181 | 191 | 265 | 289 | 299 | 333 | 345 | 355 | 432 | 476 |
| 42 | 4 7/16 | ... | ... | 112 | 117 | 123 | 136 | 153 | 176 | 188 | 199 | 277 | 299 | 311 | 343 | 359 | 366 | 449 | 495 |
| 44 | 4 7/16 | ... | ... | ... | ... | 128 | 190 | 236 | 270 | 285 | 295 | 380 | 461 | 472 | 528 | 546 | 551 | 704 | 753 |
| 46 | 4 7/16 | ... | ... | ... | ... | 134 | 193 | 240 | 275 | 290 | 300 | 386 | 469 | 480 | 538 | 554 | 562 | 721 | 773 |
| 48 | 4 7/16 | ... | ... | ... | ... | 142 | 198 | 245 | 280 | 295 | 305 | 396 | 478 | 490 | 544 | 558 | 573 | 736 | 790 |
| 50 | 4 7/16 | ... | ... | ... | ... | 220 | 230 | 260 | 289 | 304 | 317 | 454 | 486 | 515 | 547 | 564 | 585 | 753 | 810 |
| 52 | 4 7/16 | ... | ... | ... | ... | 226 | 237 | 267 | 297 | 312 | 327 | 467 | 497 | 539 | 559 | 574 | 602 | 771 | 831 |
| 54 | 4 7/16 | ... | ... | ... | ... | 232 | 244 | 275 | 306 | 321 | 337 | 482 | 514 | 545 | 576 | 590 | 621 | 796 | 858 |
| 56 | 4 7/16 | ... | ... | ... | ... | 237 | 251 | 282 | 316 | 331 | 348 | 495 | 527 | 560 | 592 | 607 | 639 | 818 | 882 |
| 58 | 4 7/16 | ... | ... | ... | ... | 244 | 257 | 291 | 322 | 340 | 358 | 509 | 541 | 575 | 609 | 625 | 656 | 839 | 906 |
| 60 | 4 7/16 | ... | ... | ... | ... | 250 | 263 | 298 | 333 | 349 | 368 | 520 | 555 | 590 | 623 | 641 | 673 | 861 | 930 |
| 62 | 6 | ... | ... | ... | ... | 348 | 362 | 395 | 429 | 449 | 469 | 720 | 753 | 769 | 803 | 821 | 855 | 1150 | 1224 |
| 64 | 6 | ... | ... | ... | ... | 356 | 370 | 404 | 439 | 460 | 480 | 735 | 769 | 782 | 822 | 847 | 876 | 1167 | 1244 |
| 66 | 6 | ... | ... | ... | ... | 363 | 378 | 413 | 449 | 471 | 492 | 750 | 785 | 803 | 840 | 859 | 896 | 1195 | 1275 |
| 68 | 6 | ... | ... | ... | ... | 370 | 385 | 422 | 460 | 481 | 503 | 766 | 802 | 820 | 858 | 878 | 916 | 1218 | 1300 |
| 70 | 6 | ... | ... | ... | ... | 377 | 393 | 431 | 470 | 492 | 514 | 787 | 818 | 837 | 876 | 896 | 935 | 1243 | 1337 |
| 72 | 6 | ... | ... | ... | ... | 385 | 401 | 440 | 480 | 503 | 526 | 796 | 835 | 854 | 894 | 915 | 955 | 1268 | 1355 |
| 78 | 6 | ... | ... | ... | ... | ... | 442 | 466 | 510 | 535 | 560 | 844 | 885 | 906 | 950 | 972 | 1016 | 1353 | 1422 |
| 84 | 8 1/2 | ... | ... | ... | ... | ... | 587 | 632 | 672 | 712 | 753 | 1211 | 1251 | 1291 | 1331 | 1371 | 1391 | 1876 | 1996 |
| 90 | 8 1/2 | ... | ... | ... | ... | ... | .... | 683 | 723 | 763 | 803 | 1304 | 1344 | 1384 | 1424 | 1464 | 1484 | 2011 | 2131 |
| 96 | 8 1/2 | ... | ... | ... | ... | ... | .... | 734 | 774 | 814 | 855 | 1397 | 1437 | 1477 | 1517 | 1557 | 1577 | 2146 | 2266 |
| 102 | 8 1/2 | ... | ... | ... | ... | ... | .... | 855 | 895 | 935 | 975 | 1510 | 1550 | 1590 | 1630 | 1670 | 1690 | 2507 | 2627 |
| 108 | 8 1/2 | ... | ... | ... | ... | ... | .... | 915 | 955 | 995 | 1035 | 1570 | 1610 | 1650 | 1690 | 1730 | 1750 | 2642 | 2762 |
| 114 | 8 1/2 | ... | ... | ... | ... | ... | .... | .... | 1090 | 1130 | 1170 | 1990 | 2030 | 2070 | 2110 | 2150 | 2170 | 2810 | 2930 |
| 120 | 8 1/2 | ... | ... | ... | ... | ... | .... | .... | 1180 | 1220 | 1260 | 2070 | 2110 | 2150 | 2190 | 2230 | 2260 | 2945 | 3065 |

**Table 5.—Actual Width of Standard Steel Split Pulleys**

(American Pulley Co., Philadelphia)

| Pulley Diam., in. | Nominal Face Width, in. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 2 | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 14 |
| | Actual Face Width, in. | | | | | | | | |
| 3 and 3 1/2 | 2 7/16 | 3 7/16 | 4 7/16 | 5 7/8 | 6 7/8 | ........ | ........ | ........ | ........ |
| 4 | 2 7/16 | 3 7/16 | 4 7/16 | 5 7/16 | 6 7/8 | ........ | ........ | ........ | ........ |
| 5 | 2 7/16 | 3 7/16 | 4 7/16 | 5 7/16 | 6 7/16 | 8 7/8 | 10 7/8 | ........ | ........ |
| 6 | 2 7/16 | 3 7/16 | 4 7/16 | 5 7/16 | 6 7/16 | 8 7/16 | 10 7/8 | 12 7/8 | ........ |
| 7 | 2 7/16 | 3 7/16 | 4 7/16 | 5 9/16 | 6 9/16 | 8 7/16 | 10 9/16 | 13 1/8 | ........ |
| 8 and 9 | 2 11/16 | 3 11/16 | 4 11/16 | 5 11/16 | 6 3/4 | 8 13/16 | 10 13/16 | 13 1/2 | ........ |
| 10 and 11 | 2 11/16 | 3 11/16 | 4 11/16 | 5 11/16 | 6 3/4 | 8 13/16 | 10 13/16 | 12 3/4 | ........ |
| 12 | 2 11/16 | 3 11/16 | 4 11/16 | 5 11/16 | 6 3/4 | 8 13/16 | 10 13/16 | 12 13/16 | 14 13/16 |
| 13–16 | 2 11/16 | 3 11/16 | 4 11/16 | 5 3/4 | 6 3/4 | 8 3/4 | 10 13/16 | 12 13/16 | 14 13/16 |
| 17–19 | ........ | 3 11/16 | 4 11/16 | 5 3/4 | 6 3/4 | 8 3/4 | 10 13/16 | 12 13/16 | 14 13/16 |
| 20–23 | ........ | 3 11/16 | 4 11/16 | 5 3/4 | 6 3/4 | 8 3/4 | 10 13/16 | 12 13/16 | 14 13/16 |
| 24 | ........ | 3 11/16 | 4 11/16 | 5 3/4 | 6 3/4 | 8 3/4 | 10 13/16 | 12 13/16 | 14 13/16 |
| 25–28 | ........ | ........ | 4 7/8 | 5 7/8 | 6 15/16 | 8 15/16 | 10 15/16 | 13 | 15 1/16 |
| 30 | ........ | ........ | 4 7/8 | 5 7/8 | 6 15/16 | 8 15/16 | 10 15/16 | 13 | 15 1/16 |
| 32–34 | ........ | ........ | 4 7/8 | 5 7/8 | 6 15/16 | 8 15/16 | 10 15/16 | 13 | 15 1/16 |
| 36–42 | ........ | ........ | 4 7/8 | 5 7/8 | 6 15/16 | 8 15/16 | 10 15/16 | 13 | 15 1/16 |
| 44–48 | ........ | ........ | ........ | ........ | 7 1/8 | 9 1/8 | 11 1/8 | 13 3/16 | 15 1/4 |
| 50–72 | ........ | ........ | ........ | ........ | 7 3/8 | 9 3/8 | 11 3/8 | 13 3/8 | 15 3/8 |
| 74–88 | ........ | ........ | ........ | ........ | ........ | 9 3/8 | 11 3/8 | 13 3/8 | 15 3/8 |
| 90–112 | ........ | ........ | ........ | ........ | ........ | ........ | 11 3/8 | 13 3/8 | 15 3/8 |
| 114–120 | ........ | ........ | ........ | ........ | ........ | ........ | ........ | 13 3/8 | 15 3/8 |

| Pulley Diam., in. | Nominal Face Width, in. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 30 | 36 |
| | Actual Face Width, in. | | | | | | | | |
| 12 | 16 13/16 | 18 13/16 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 13–16 | 16 13/16 | 18 13/16 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 17–19 | 16 13/16 | 18 13/16 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 20–23 | 16 13/16 | 18 13/16 | ........ | ........ | ........ | ........ | ........ | ........ | ........ |
| 24 | 16 13/16 | 18 13/16 | 20 13/16 | ........ | ........ | ........ | ........ | ........ | ........ |
| 25–28 | 17 1/16 | 18 15/16 | 20 15/16 | ........ | ........ | ........ | ........ | ........ | ........ |
| 30 | 17 1/16 | 18 15/16 | 20 15/16 | 22 15/16 | 25 | 27 1/16 | 29 | 31 1/16 | ........ |
| 32–34 | 17 1/16 | 18 15/16 | 20 15/16 | 22 15/16 | 25 | 27 1/16 | 29 | 31 1/16 | ........ |
| 36–42 | 17 1/16 | 18 15/16 | 20 15/16 | 22 15/16 | 25 | 27 1/16 | 29 | 31 1/16 | 37 1/16 |
| 44–48 | 17 1/4 | 19 1/8 | 21 1/8 | 23 1/8 | 25 3/16 | 27 1/4 | 29 3/16 | 31 1/4 | 37 1/4 |
| 50–72 | 17 3/8 | 19 3/8 | 21 3/8 | 23 3/8 | 25 3/8 | 27 3/8 | 29 3/8 | 31 3/8 | 37 3/8 |
| 74–88 | 17 3/8 | 19 3/8 | 21 3/8 | 23 3/8 | 25 3/8 | 27 3/8 | 29 3/8 | 31 3/8 | 37 3/8 |
| 90–112 | 17 3/8 | 19 3/8 | 21 3/8 | 23 3/8 | 25 3/8 | 27 3/8 | 29 3/8 | 31 3/8 | 37 3/8 |
| 112–120 | 17 3/8 | 19 3/8 | 21 3/8 | 23 3/8 | 25 3/8 | 27 3/8 | 29 3/8 | 31 3/8 | 37 3/8 |

**SHAFT SPEEDS WITH VARIOUS DIAMETERS OF PULLEY.**—Let $N$, $n$ = respectively, rev. per min. of driving and driven shafts; $D$, $d$ = respectively, diameter of driving and driven pulleys, in.; $V$ = velocity of belt, ft. per min. Then $DN = dn$; $D = dn/N$; $d = DN/n$; $n = DN/d$; $N = dn/D$; $V = \pi DN/12 = \pi dn/12$.

## 2. CONE PULLEYS

Cone pulleys for driving machine tools should be so designed that the speeds will be in geometrical progression, that is, each speed is the product of the next lower speed and a constant multiplier. According to Carl G. Barth (*Am. Mach.*, Jan. 11, 1912) the ideal ratio of speeds is $\sqrt[4]{2} = 1.189$.

Three general cases exist in the design of cone pulley drives. 1. Crossed belts. 2. Open belts, where the pulleys are so far apart that it is unnecessary to compensate for the tendency of the changing angle to alter the length of the belt. 3. Open belt, with the pulleys so close together that the change in the angle tends to alter the length of the belt, which tendency must be compensated. Inasmuch as in crossed belts the length of the belt is constant, and with open belts with long center distances, nearly so, it is only necessary that the sum of the diameters of the corresponding steps on the mating cones be constant. In case 3, the length of the belt is a function of both the sum and difference of the diameters of the pulleys and a simple solution is not possible. A large number of graphical methods for determining the diameters of the several steps of the pulleys are given in Halsey's Handbook for Machine Designers. A graphical method and a trigonometrical method derived from it are given by C. A. Smith (*Trans.*, A.S.M.E., x, 269).

A slight error, however, in a graphical solution on a small scale may result in a serious error in the size of the pulleys, and a mathematical solution is preferable.

An algebraic solution was derived by William Kent from Burmeister's graphical solution ("*Lehrbuch der Kinematic*"; *Mach'y* Reference Series, No. 14, 1908), which give results far more accurate than are required in practice. In all cases 0.8 of the thickness of the belt should be subtracted from the calculated diameter to obtain the actual diameter of the pulley. This should be done because the belt drawn tight around the pulleys is not the same length as a tape-line measure around them.

The algebraic solution of the problem leads to the following equations:

Let $L$ = distance between the centers; $r_0$ = radius of steps of equal diameter on the two cones; $r_1, r_2$ = radii of any pair of steps; $a = (0.79057\,L - r_0)$. If $r_1$ is given,

$$r_2 = \sqrt{1.25\,L^2 - (0.79057\,L - r_0 + r_1)^2} - 0.79057\,L + r_0. \quad . \quad . \quad . \qquad [14]$$

If the ratio $r_2 \div r_1$ is given, let $r_2/r_1 = c$: $r_2 = cr_1$.

We then have $$a + cr_1 = \sqrt{R^2 - (a + r_1)^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [15]$$

which reduces to $$(1 + c^2)r_1^2 + 2\,a(1 + c)r_1 = 1.25\,L^2 - 2\,a^2, \quad . \quad . \quad . \quad . \quad . \quad . \qquad [16]$$

a quadratic equation, in which $a = 0.79057\,L - r_0$. Substituting the value of $a$ we have

$$(1 + c^2)r_1^2 + (1.58114\,L - 2\,r_0)\,(1 + c)r_1 = 3.16228\,Lr_0 - 2\,r_0^2, \quad . \quad . \qquad [17]$$

in which $L$, $r_0$ and $c$ are given and $r_1$ is to be found.

Let $L = 100$, $c = 4$, $r_0 = 12.858$.

Then $17\,r_1^2 + 10\,ar_1, = 12{,}500 - 8764.62$, from which $r_1 = 5.001$, $r_2 = 20.004$.

If $c = 3$, $r_1 = 6.304$, $r_2 = 18.912$. If $c = 2$, $r_1 = 8.496$, $r_2 = 16.992$.

Checking the results by the approximate formula for length of belt, *viz.*, Length = $2\,L + \pi(r_1 + r_2) + (r_2 - r_1)^2 \div L$, we have

for $c$ = 1, 200 + 80.79 + 0 = 280.79
2, 200 + 80.07 + 0.72 = 280.79
3, 200 + 79.22 + 1.59 = 280.81
4, 200 + 78.56 + 2.25 = 280.81

The maximum difference is only 1 part in 14,000.

J. J. Clark (*Indust. Eng.*, Aug., 1910) gives the following solution: Using the same notation as before,

$$\frac{(c - 1)^2}{L}r_1^2 + \pi(c + 1)r_1 = 2\pi r_0, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [18]$$

$$\pi(c + 1)r_1 + Lx\left(\frac{60 - 13\,x}{60 - 18\,x}\right) = 2\pi r_0, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [19]$$

$$x = (r_2 - r_1)^2 \div L^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [20]$$

The quadratic equation [18] gives the value of $r_1$ with an approximation to accuracy sufficient for all practical purposes. If greater accuracy is for any reason desired it may be obtained by [19] and [20], using in [20] the values of $r_1$ and $r_2 = cr_1$, already found from [18]. Taking $\pi = 3.1415927$, the result will be correct to the seventh figure.

**SPEEDS OF SHAFT WITH CONE PULLEYS.**—If $S$ = speed (revs. per min.) of the driving shaft,

$s_1, s_2, s_3, s_n$ = speeds of the driven shaft,
$D_1, D_1, D_3, D_n$ = diameters of the pulleys on the driving cone,
$d_1, d_2, d_3, d_n$ = diameters of corresponding pulleys on the driven cone,
$SD_1 = s_1\,d_1$; $SD_2 = s_2\,d_2$, etc.
$s_1/S = D_1/d_1 = r_1$; $s_n/S = D_n/d_n, = r_n$.

The speed of the driving shaft being constant, the several speeds of the driven shaft are proportional to the ratio of the diameter of the driving pulley to that of the driven, or to $D/d$.

**SPEEDS IN GEOMETRICAL PROGRESSION.**—If it is desired that the speed ratios shall increase by a constant percentage, or in geometrical progression, then $r_2/r_1 = r_3/r_2 = r_n/r_{n-1} = c$, a constant.

$$r_n \div r_1 = c^{n-1}; \; c = \sqrt[n-1]{r_n \div r_1}.$$

Example.—If the speed ratio of the driven shaft at its lowest speed, to the driving shaft be 0.76923, and at its highest speed 2.197, the speeds being in geometrical progression, what is the constant multiplier if $n = 5$?

Log 2.197 = 0.341830
Log 0.76923 = 1.886056
0.455774

Divide by $n - 1$, = 4, 0.113943 = log of 1.30.
If $D_2/d_2 = 1$, then $D_1/d_1 = 1 \div 1.3 = 0.769$; $D_3/d_3 = 1.30$; $D_4/d_4 = 1.69$; $D_5/d_5 = 2.197$.

# BELT DRIVES

## 1. THEORY OF POWER TRANSMISSION BY BELTING

**BELT TENSIONS.**—A pulley is driven by a belt by means of the friction between the surface of the belt in contact with the pulley. The driving force is the difference between the tension $t_1$ in the tight side of the belt and the tension $t_2$ in the slack side; that is, $p = t_1 - t_2$, where $p$ = effective pull of the belt in pounds. With a given sum of the tensions $t_1 + t_2$, the limit of the tractive force $p$ without serious or objectionable slipping of the belt on the pulley depends upon the length $\alpha$ of the belt in contact with the pulley and upon the coefficient of friction $f$. $\alpha$ is expressed in radians. The value of the coefficient $f$ is discussed below. The relation existing between the quantities $t_1$, $t_2$, $f$ and $\alpha$ is expressed by the exponential formula

$$t_1/t_2 = e^{f\alpha}.$$

Carl G. Barth (*Trans.*, A.S.M.E., xxxi, p. 29, 1909) has developed a theory of belting which is as follows: In considering the formulas which are based primarily on Taylor's investigations on belting (see below), the following assumptions are made: 1. The pulling power of a belt depends on the tension in belt when it is at rest, that is, on the initial tension. 2. The belt in service stretches and thereby decreases the initial tension. The stretch is rapid when the belt is first put up and decreases in amount with the age and service of the belt. 3. A belt whose initial tension is allowed to fall below a predetermined figure will, at a given belt velocity, fail to transmit the desired horsepower. 4. Excessive unit tension in a belt will cause it to stretch rapidly and will increase the maintenance and other charges relating to it. 5. Heavy belts at moderate tensions will give better service than light belts at higher tension. 6. From the standpoint of belt maintenance, the intervals during which belts, running at different velocities and transmitting different horsepowers, will fall to the minimum allowable unit tension and require retightening, should be equalized as far as possible. Belts that are readily accessible can be taken down and retightened more easily than belts in inaccessible locations and, therefore, can be run at higher tensions. An initial unit tension of 200 lb. per sq. in. is about the maximum that should be put upon a belt if it is to be used under the most economical conditions.

**Notation.**—$t_1$ = tension in the tight strand of the running belt, lb. per sq. in.; $t_2$ = tension in the slack strand of the running belt, lb. per sq. in.; $t_0$ = minimum initial tension in the belt at rest on its pulleys, lb. per sq. in. This is the tension to which the belt can be allowed to fall before retightening. $t_m$ = maximum initial tension in the belt at rest upon its pulleys, lb. per sq. in. This is the tension to which the belt is retightened. $t_c$ = centrifugal tension in belt, or loss in effective tension due to centrifugal force, lb. per sq. in.; $p$ = effective pull in belt, lb. per sq. in.; $\alpha$ = arc of contact of the belt on the smalle pulley, in radians = $(\pi/180) \times$ degrees of arc; $f$ = coefficient of friction between belt an pulley; $V$ = velocity of belt, ft. per min.; $e$ = base of Naperian system of logarithms = 2.71828; $A$ = a constant, whose values range from 160 to 320 as explained below.

$p = t_1 - t_2$, . . . . . . . [1]; $t_1 + {}^1/_2\, t_2 = A$ . . . . . . . . [2

For machine belts $A$ = 240 and 320, depending upon whether the values of $t_0$ or $t_m$ respectively, are being sought, as explained later. For countershaft belts, $A$ = 160 an 240, respectively, for values of $t_0$ and $t_m$.

The passage of the belt around the pulley develops a centrifugal force in the belt whic decreases the tension in each strand of the belt, that is, the effective tensions are $(t_1 - t_c$ and $(t_2 - t_c)$ for the tight and slack strands, respectively. The total friction of the bel on the pulley is the ratio of the effective tensions, whence

$(t_1 - t_c) \div (t_2 - t_c) = e^{f\alpha}$ . . . . . . . . . . [3

For the expression $e^{f\alpha}$ may be substituted the expression $10^{0.00758 f\theta}$, where $\theta$ is the arc o contact in degrees. This permits the use of the common system of logarithms instead o the Naperian system. This equation may be interpreted as: The ratio of the effectiv tensions is the number whose common logarithm is $0.00758\, f\theta$.

The centrifugal tension developed at any velocity is

$t_c = 0.000\,003\,454\, V^2$ . . . . . . . . . . . [4 and the coefficient of friction is found from the equation

$$f = 0.54 - \{140/(500 + V)\} \quad . \; . \; . \; . \; . \; . \; . \; . \; [5]$$

Combining the foregoing equations 1, 2, 3, 4 and 5, we obtain the equation expressing the value of $p$ directly in terms of $f$, $\alpha$, $A$ and $V$ as

$$p = \frac{(e^{f\alpha} - 1)\,(2A - 0.000\,010\,36V^2)}{2e^{f\alpha} + 1} \quad . \; . \; . \; . \; . \; . \; [6]$$

which may be expressed as

$$p = \frac{(10^{0.00758f\theta} - 1)\,(2A - 0.000\,010\,36\,V^2)}{2 \times 10^{0.00758f\theta} + 1} \quad . \; . \; . \; . \; . \; . \; [7]$$

The horsepower transmitted then may be determined from the general equation

$$\text{H.P.} = pV/33{,}000.$$

The value of the initial tension in the belt when it is at rest on the pulleys is connected with the formula for $p$ by means of the equation

$$t_{\text{initial}} = {}^1/_{12}\,\{4A - p + \sqrt{(4A - p)^2 - 9p^2}\} \quad . \; . \; . \; . \; . \; . \; [8]$$

which will give values for $t_0$ and $t_m$, respectively, depending on whether 240 or 320 is substituted for machine belts and 160 or 240 for countershaft belts. If, after having determined the values of $p$, $t_0$ and $t_m$, values of $t_1$ and $t_2$ are desired, they may be found by equations [9] and [10].

$$t_1 = {}^1/_3\,(2A + p) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [9]$$

$$t_2 = {}^2/_3\,(A - p) \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [10]$$

These equations are useful if it is desired to determine the initial tension in a horizontal belt which theoretically is slightly different from that which should be used for a vertical belt, to which equation [8] applies. This tension, however, differs so slightly from that of a vertical belt, except in the case of belts of extraordinary length, that there is no practical value in determining it separately. This equation for long horizontal belts is

$$t_{\text{initial}} = {}^1/_4\left\{\sqrt{t_1} + \sqrt{t_2} - \left(\frac{C}{5}\,\frac{t_1 - t_2}{t_1\,t_2}\right)^2\right\}^2 \quad . \; . \; . \; . \; . \; . \; [11]$$

where $C$ = center to center distance of the belts, in., other notation being as above. In using equation [11], $p$ is first determined by equations [6] or [7] and $t_1$ and $t_2$ are then evaluated by equations [9] and [10]. Table 2 has been calculated from these equations.

In earlier discussions of the theory of belting, two important assumptions were made which were found by Mr. Barth to be untrue. These were: (*a*) The sum of the tensions in the two strands of a running belt is constant under any variations in the effective pull. (*b*) The coefficient of friction between the belt and its pulley is constant for any given set of conditions, irrespective of the velocity of the belt. To replace the first of these assumptions, Mr. Barth proved the theorem "Under any variation in the effective pull of the belt, the sum of the square roots of the tensions in the two strands remains constant," and for assumption (*b*) Mr. Barth developed the empirical equation [5] for the coefficient of friction which is discussed below.

**COEFFICIENT OF FRICTION.**—Early writers and experimenters gave constant values for the coefficient of friction between the belt and the surface of the pulley. These varied widely. For instance, Morin gave for oily belts, 0.15; for greasy belts, 0.23; for wet belts, 0.36; for dry belts, 0.56. The experiments of Towne and Briggs in 1868 resulted in a recommendation of $f = 0.42$ as the average of the conditions of belt operation. Releaux gives 0.25, which figure generally has been used in approximate calculations.

Mr. Barth's studies of the experiments of Messrs. Lewis and Bird developed the fact that the coefficient of friction varies with the velocity of the belt and its value may be determined quite closely by equation [5]. Values of $f$ according to this equation, for velocities ranging from 0 to 6500 ft. per min., are given in Table 1.

**Table 1.—Values of the Coefficient of Friction of Leather Belts on Iron Pulleys**

(Barth's Formula)

$V$ = belt velocity, ft. per min.; $f$ = coefficient of friction

| $V$ | $f$ | $V$ | $f$ | $V$ | $f$ | $V$ | $f$ | $V$ | $f$ |
|---|---|---|---|---|---|---|---|---|---|
| 0 | 0.260 | 500 | 0.400 | 1000 | 0.446 | 2000 | 0.484 | 4500 | 0.512 |
| 50 | .285 | 600 | .413 | 1200 | .458 | 2500 | .493 | 5000 | .514 |
| 100 | .307 | 700 | .423 | 1400 | .466 | 3000 | .500 | 5500 | .517 |
| 200 | .340 | 800 | .432 | 1600 | .473 | 3500 | .505 | 6000 | .519 |
| 300 | .365 | 900 | .440 | 1800 | .479 | 4000 | .509 | 6500 | .520 |
| 400 | .384 | | | | | | | | |

**WORKING TENSIONS.**—The formulas developed above, and the tables of horsepower calculated therefrom are not based upon the maximum horsepower that a belt can transmit, or on the smallest size of belt that will transmit a given horsepower. Instead, they are designed to give the size and tension of belt that will give the most economical service, first cost, cost of maintenance and repairs, cost of idle equipment due to belt failures, and loss of power due to excessive tensions all being considered.

The foregoing theory is based on the following rule:

For the driving belt of a machine, the initial tension must be such that when the belt is doing the maximum amount of work intended, the sum of the tension on the tight side of the belt and one-half the tension on the slack side will equal 240 lb. per square inch of cross-section for all belt speeds; and for a belt driving a countershaft or one in any other inaccessible location, this sum will equal 160 lb. The maximum initial tension, that is, the tension under which the belt is put upon the pulley when it is placed in service and the tension to which it is to be retightened as often as it falls to the minimum, must be such that the above derived sum is 320 lb. for machine belts and 240 lb. for countershaft belts. These values represent the constant $A$ in the preceding mathematical discussion. Belts in service should be subjected to regular inspection and care. The tension in them should be weighed at intervals and when it falls to the value $t_0$ as given in Table 2, they should be taken down, and a sufficient amount removed from them to restore them to the original initial tension $t_1$.

## 2. HORSEPOWER OF LEATHER BELTS

**ARC OF CONTACT.**—Equation [3] shows that the pull of a belt, and consequently the horsepower transmitted, is a variable depending on the arc of contact of the belt on the pulley. If $\theta$ = arc of contact, $R$ and $r$ = the radii of the larger and smaller pulleys, respectively, $L$ = the center to center distance of the pulleys, all in inches, and $\alpha = \sin^{-1}(R - r)/L$, then, on the large pulley

$$\theta = (180 + 2\alpha) \text{ deg.} \qquad [12]$$

On the small pulley

$$\theta = (180 - 2\alpha) \text{ deg.} \qquad [13]$$

For crossed belts the value of $\theta$ for both pulleys is found by equation [12].

The length of belt $L_b$ in contact with the pulleys of open belts is

$$\text{Large pulley, } L_b = R(\pi + 0.0349 \sin^{-1}\alpha) \qquad [14]$$

$$\text{Small pulley, } L_b = r(\pi - 0.0349 \sin^{-1}\alpha) \qquad [15]$$

With crossed belts, equation [14] applies for the large pulley. For the small pulley, the equation becomes

$$L_b = r(\pi + 0.0349 \sin^{-1}\alpha) \qquad [16]$$

**HORSEPOWER OF LEATHER BELTS.**—Table 2 has been calculated from the Barth equations for both machine and countershaft belts. The arc of contact used is the arc on the smaller pulley, which determines the amount of power that can be delivered to or by the belt. The tables not only show the power that can be transmitted per square inch of cross-sectional area of belt, but also the limits of tension between which the belt should be used in order to transmit the desired power. The values of initial tension $t_m$ are the tensions to which the belt should be tightened when it is put up, and the values of minimum tension $t_0$ are the values below which the tension should not be allowed to fall without retightening.

To use the table, ascertain the arc of contact of the belt on the smaller pulley, by means of equations [12] to [16], and in the section of Table 2 most nearly corresponding to this arc read the horsepower opposite the belt velocity. Multiply the figure so found by the cross-sectional area of the belt in square inches. For velocities other than those in the table, horsepower may be taken as proportional to the velocity. Similarly, the values of the initial and minimum tensions may be taken as proportional to the velocities, care being used to observe whether the consecutive values of the tension as given in the table on either side of the intermediate velocity are increasing or decreasing due to the effect of centrifugal force.

To ascertain the size of belt that will be required to transmit a given horsepower, find from the table the horsepower that a belt of 1 sq. in. cross-sectional area will transmit under the given conditions of speed and arc of contact. Divide the figure so found into the required horsepower, to find the cross-sectional area of the belt. From Table 5 find the maximum width of belt that can be used on the pulleys which are to be used, and divide this width into the cross-sectional area to ascertain the thickness.

**Manufacturer's Ratings.**—Tables of horsepower transmitted by leather belting now (1937) issued by belt manufacturers show, as a rule, greater capacity than the values given

(*Continued on page* 24–19)

**Table 2.—Horsepower and Tension of Belts per Sq. In. of Cross-section**

| | Velocity of Belt, ft. per min. | Machine Belts | | | | | | Countershaft Belts | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Horsepower | Initial Tension, lb. per sq. in., $t_m$ | Minimum Tension, lb. per sq. in., $t_0$ | Tension, Tight Side, lb. per sq. in., $t_1$ | Tension, Slack Side, lb. per sq. in., $t_2$ | Effective Pull, lb. per sq. in., $t_1 - t_2$ | Horsepower | Initial Tension, lb. per sq. in., $t_m$ | Minimum Tension, lb. per sq. in., $t_0$ | Tension, Tight Side, lb. per sq. in., $t_1$ | Tension, Slack Side, lb. per sq. in., $t_2$ | Effective Pull, lb. per sq. in., $t_1 - t_2$ |
| 180 Degrees Arc of Contact | 250 | 1.05 | 183.75 | 127.5 | 206.0 | 67.9 | 138.1 | 0.70 | 141.0 | 85.0 | 137.3 | 45.3 | 92.0 |
| | 500 | 2.27 | 180.25 | 124.0 | 209.8 | 60.3 | 149.5 | 1.51 | 139.0 | 82.5 | 139.8 | 40.4 | 99.4 |
| | 750 | 3.53 | 179.0 | 121.75 | 211.8 | 56.5 | 155.3 | 2.34 | 138.0 | 81.25 | 141.1 | 37.9 | 103.2 |
| | 1000 | 4.78 | 178.5 | 121.0 | 212.5 | 54.9 | 157.6 | 3.15 | 137.75 | 81.0 | 141.3 | 37.4 | 103.9 |
| | 1200 | 5.77 | 178.0 | 120.5 | 212.9 | 54.3 | 158.6 | 3.78 | 137.75 | 81.0 | 141.3 | 37.3 | 104.0 |
| | 1400 | 6.72 | 178.0 | 120.5 | 212.8 | 54.4 | 158.4 | 4.38 | 138.0 | 81.25 | 141.1 | 37.8 | 103.3 |
| | 1600 | 7.64 | 178.25 | 121.0 | 212.5 | 54.9 | 157.6 | 4.96 | 138.25 | 81.75 | 140.7 | 38.7 | 102.0 |
| | 1800 | 8.53 | 178.5 | 121.5 | 212.1 | 55.8 | 156.3 | 5.47 | 138.75 | 82.25 | 140.1 | 39.8 | 100.3 |
| | 2000 | 9.36 | 179.25 | 122.0 | 211.5 | 57.0 | 154.5 | 5.95 | 139.0 | 83.0 | 139.4 | 41.3 | 98.1 |
| | 2500 | 11.20 | 181.0 | 124.5 | 209.3 | 61.5 | 147.8 | 6.88 | 141.25 | 85.5 | 137.0 | 46.1 | 90.9 |
| | 3000 | 12.62 | 183.5 | 127.5 | 206.3 | 67.5 | 138.8 | 7.40 | 143.5 | 88.5 | 133.8 | 52.4 | 81.4 |
| | 3500 | 13.51 | 186.75 | 131.0 | 202.5 | 75.1 | 127.4 | 7.39 | 146.25 | 91.75 | 129.9 | 60.2 | 69.7 |
| | 4000 | 13.80 | 190.0 | 135.0 | 197.9 | 84.1 | 113.8 | 6.77 | 149.5 | 95.25 | 125.3 | 69.4 | 55.9 |
| | 4500 | 13.40 | 193.75 | 139.25 | 192.7 | 94.5 | 98.2 | 5.46 | 152.5 | 98.75 | 120.0 | 79.9 | 40.1 |
| | 5000 | 12.20 | 197.75 | 143.75 | 186.8 | 106.3 | 80.5 | 3.37 | 156.0 | 103.0 | 114.1 | 91.9 | 22.2 |
| 170 Degrees Arc of Contact | 250 | 1.00 | 185.25 | 129.5 | 204.2 | 71.6 | 132.6 | 0.67 | 142.0 | 86.25 | 136.1 | 47.8 | 88.3 |
| | 500 | 2.18 | 182.0 | 125.75 | 208.0 | 64.1 | 143.9 | 1.45 | 140.0 | 84.0 | 138.6 | 42.9 | 95.7 |
| | 750 | 3.40 | 180.75 | 123.5 | 209.9 | 60.2 | 149.7 | 2.26 | 139.0 | 82.5 | 139.8 | 40.5 | 99.3 |
| | 1000 | 4.61 | 180.0 | 123.0 | 210.7 | 58.7 | 152.0 | 3.04 | 139.0 | 82.25 | 140.1 | 39.8 | 100.3 |
| | 1200 | 5.57 | 179.5 | 122.5 | 211.1 | 57.8 | 153.3 | 3.65 | 138.75 | 82.25 | 140.2 | 39.6 | 100.6 |
| | 1400 | 6.49 | 179.5 | 122.5 | 211.0 | 58.0 | 153.0 | 4.23 | 139.0 | 82.50 | 139.9 | 40.2 | 99.7 |
| | 1600 | 7.38 | 180.0 | 123.0 | 210.8 | 58.5 | 152.3 | 4.78 | 139.0 | 83.0 | 139.5 | 41.0 | 98.5 |
| | 1800 | 8.24 | 180.25 | 123.25 | 210.4 | 59.3 | 151.1 | 5.29 | 139.50 | 83.5 | 139.0 | 42.0 | 97.0 |
| | 2000 | 9.05 | 180.75 | 123.75 | 209.8 | 60.4 | 149.4 | 5.75 | 140.0 | 84.25 | 138.3 | 43.4 | 94.9 |
| | 2500 | 10.83 | 182.5 | 126.0 | 207.7 | 64.7 | 143.0 | 6.66 | 142.0 | 86.5 | 136.0 | 48.1 | 87.9 |
| | 3000 | 12.21 | 184.75 | 129.0 | 204.8 | 70.5 | 134.3 | 7.16 | 144.25 | 89.25 | 132.9 | 54.2 | 78.7 |
| | 3500 | 13.08 | 187.75 | 132.25 | 201.1 | 77.8 | 123.3 | 7.15 | 146.75 | 92.25 | 129.1 | 61.8 | 67.4 |
| | 4000 | 13.37 | 191.0 | 136.0 | 196.8 | 86.5 | 110.3 | 6.56 | 149.75 | 96.75 | 124.7 | 70.6 | 54.1 |
| | 4500 | 12.97 | 194.5 | 140.25 | 191.7 | 96.6 | 95.1 | 5.29 | 152.75 | 99.0 | 119.6 | 80.8 | 38.8 |
| | 5000 | 11.81 | 198.5 | 144.5 | 186.0 | 108.0 | 78.0 | 3.26 | 156.25 | 103.25 | 113.8 | 92.3 | 21.9 |
| 160 Degrees Arc of Contact | 250 | 0.96 | 186.75 | 131.0 | 202.3 | 75.4 | 126.9 | 0.64 | 143.0 | 87.5 | 134.8 | 50.3 | 84.5 |
| | 500 | 2.09 | 183.75 | 127.75 | 206.1 | 67.8 | 138.3 | 1.39 | 141.0 | 85.25 | 137.3 | 45.5 | 91.5 |
| | 750 | 3.27 | 182.25 | 125.75 | 207.9 | 64.1 | 143.8 | 2.17 | 140.0 | 84.0 | 138.6 | 42.8 | 95.8 |
| | 1000 | 4.43 | 181.75 | 125.0 | 208.7 | 62.5 | 146.2 | 2.92 | 140.0 | 83.5 | 138.8 | 42.4 | 96.8 |
| | 1200 | 5.36 | 181.25 | 124.5 | 209.1 | 61.8 | 147.3 | 3.51 | 139.75 | 83.5 | 138.9 | 42.2 | 96.4 |
| | 1400 | 6.25 | 181.25 | 124.5 | 209.1 | 61.9 | 147.2 | 4.07 | 140.0 | 83.75 | 138.7 | 42.7 | 96.7 |
| | 1600 | 7.11 | 181.25 | 125.0 | 208.9 | 62.3 | 146.6 | 4.60 | 140.25 | 84.25 | 138.3 | 43.4 | 94.0 |
| | 1800 | 7.94 | 181.75 | 125.25 | 208.5 | 63.0 | 145.5 | 5.09 | 140.50 | 84.75 | 137.8 | 44.4 | 93.9 |
| | 2000 | 8.72 | 182.25 | 126.75 | 208.0 | 64.1 | 143.9 | 5.54 | 141.0 | 85.25 | 137.1 | 45.7 | 91.4 |
| | 2500 | 10.44 | 184.0 | 127.75 | 205.9 | 68.1 | 137.8 | 6.42 | 142.75 | 87.50 | 134.9 | 50.2 | 84.4 |
| | 3000 | 11.77 | 186.0 | 130.5 | 203.2 | 73.7 | 129.5 | 6.90 | 144.75 | 90.0 | 132.0 | 56.1 | 75.7 |
| | 3500 | 12.61 | 189.0 | 133.5 | 199.7 | 80.7 | 119.0 | 6.90 | 147.25 | 93.75 | 128.3 | 63.3 | 65.0 |
| | 4000 | 12.89 | 192.0 | 137.0 | 195.5 | 89.1 | 106.4 | 6.33 | 150.25 | 96.25 | 124.1 | 71.9 | 52.2 |
| | 4500 | 12.52 | 195.25 | 141.0 | 190.6 | 98.8 | 91.8 | 5.11 | 153.25 | 99.5 | 119.2 | 81.7 | 37.5 |
| | 5000 | 11.40 | 199.0 | 145.0 | 185.1 | 109.8 | 75.2 | 3.15 | 156.5 | 103.5 | 113.6 | 92.8 | 20.8 |
| 150 Degrees Arc of Contact | 250 | 0.92 | 188.25 | 132.75 | 200.3 | 79.4 | 120.9 | 0.61 | 143.75 | 88.75 | 133.5 | 53.0 | 80.5 |
| | 500 | 2.00 | 185.5 | 129.5 | 203.9 | 72.1 | 131.8 | 1.33 | 142.0 | 86.5 | 135.9 | 48.2 | 87.7 |
| | 750 | 3.12 | 184.0 | 128.0 | 205.8 | 68.4 | 137.4 | 2.07 | 141.0 | 85.5 | 137.0 | 45.9 | 91.1 |
| | 1000 | 4.24 | 183.25 | 127.0 | 206.7 | 66.7 | 140.0 | 2.80 | 141.0 | 85.0 | 137.4 | 45.1 | 92.3 |
| | 1200 | 5.13 | 183.0 | 126.5 | 207.1 | 65.9 | 141.2 | 3.37 | 140.75 | 85.0 | 137.5 | 44.9 | 92.6 |
| | 1400 | 5.99 | 183.0 | 126.5 | 207.1 | 65.9 | 141.2 | 3.90 | 141.0 | 85.0 | 137.3 | 45.3 | 92.0 |
| | 1600 | 6.82 | 183.0 | 127.0 | 206.9 | 66.3 | 140.6 | 4.41 | 141.25 | 85.5 | 137.0 | 46.0 | 91.0 |
| | 1800 | 7.62 | 183.5 | 127.0 | 206.5 | 66.9 | 139.6 | 4.89 | 141.5 | 86.0 | 136.5 | 46.9 | 89.6 |
| | 2000 | 8.37 | 183.75 | 127.5 | 206.0 | 68.0 | 138.0 | 5.32 | 142.0 | 86.5 | 135.9 | 48.2 | 87.7 |
| | 2500 | 10.02 | 185.5 | 129.5 | 204.1 | 71.8 | 132.3 | 6.16 | 143.5 | 88.5 | 133.8 | 52.5 | 81.3 |
| | 3000 | 11.31 | 187.5 | 132.0 | 201.5 | 77.1 | 124.4 | 6.63 | 145.5 | 90.75 | 131.0 | 58.1 | 72.9 |
| | 3500 | 12.12 | 190.0 | 135.0 | 198.1 | 83.8 | 114.3 | 6.63 | 148.0 | 93.5 | 127.5 | 65.0 | 62.5 |
| | 4000 | 12.39 | 193.0 | 138.25 | 194.1 | 91.9 | 102.2 | 6.08 | 150.5 | 96.75 | 123.4 | 73.2 | 50.2 |
| | 4500 | 12.03 | 196.0 | 142.0 | 189.4 | 101.2 | 88.2 | 4.91 | 153.5 | 99.75 | 118.7 | 82.7 | 36.0 |
| | 5000 | 10.96 | 199.75 | 145.75 | 184.1 | 111.8 | 72.3 | 3.02 | 156.5 | 103.75 | 113.3 | 93.3 | 20.0 |

*(Table continued on following page)*

**Table 2.—Horsepower and Tension of Belts**—*Continued*

| | Velocity of Belt, ft. per min. | Machine Belts | | | | | | Countershaft Belts | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Horsepower | Initial Tension, lb. per sq. in., $t_m$ | Minimum Tension, lb. per sq. in., $t_0$ | Tension, Tight Side, lb. per sq. in., $t_1$ | Tension, Slack Side, lb. per sq. in., $t_2$ | Effective Pull, lb. per sq. in., $t_1 - t_2$ | Horsepower | Initial Tension, lb. per sq. in., $t_m$ | Minimum Tension, lb. per sq. in., $t_0$ | Tension, Tight Side, lb. per sq. in., $t_1$ | Tension, Slack Side, lb. per sq. in., $t_2$ | Effective Pull, lb. per sq. in., $t_1 - t_2$ |
| 140 Degrees Arc of Contact | 250 | 0.87 | 190.0 | 134.75 | 198.2 | 83.6 | 114.6 | 0.58 | 144.75 | 90.0 | 132.1 | 55.8 | 76.3 |
| | 500 | 1.90 | 187.0 | 131.5 | 201.8 | 76.5 | 125.3 | 1.26 | 143.25 | 87.75 | 134.4 | 51.1 | 83.3 |
| | 750 | 2.98 | 185.75 | 130.0 | 203.7 | 72.7 | 131.0 | 1.97 | 142.25 | 86.75 | 135.6 | 48.8 | 86.8 |
| | 1000 | 4.04 | 185.0 | 129.0 | 204.5 | 71.1 | 133.4 | 2.67 | 142.0 | 86.50 | 136.0 | 48.0 | 88.0 |
| | 1200 | 4.90 | 184.75 | 128.5 | 204.9 | 70.3 | 134.6 | 3.21 | 141.75 | 86.50 | 136.1 | 47.8 | 88.3 |
| | 1400 | 5.71 | 184.75 | 128.5 | 204.9 | 70.2 | 134.7 | 3.73 | 142.0 | 86.50 | 135.9 | 48.1 | 87.8 |
| | 1600 | 6.51 | 185.0 | 129.0 | 204.7 | 70.5 | 134.2 | 4.21 | 142.25 | 86.75 | 135.6 | 48.7 | 86.9 |
| | 1800 | 7.27 | 185.0 | 129.25 | 204.4 | 71.1 | 133.3 | 4.67 | 142.5 | 87.25 | 135.2 | 49.7 | 85.5 |
| | 2000 | 7.99 | 185.5 | 129.5 | 204.0 | 72.1 | 131.9 | 5.08 | 143.0 | 87.75 | 134.6 | 50.8 | 83.8 |
| | 2500 | 9.57 | 187.0 | 131.5 | 202.1 | 75.7 | 126.4 | 5.89 | 144.5 | 89.50 | 132.6 | 54.9 | 77.7 |
| | 3000 | 10.81 | 189.0 | 133.5 | 199.6 | 80.8 | 118.8 | 6.34 | 146.25 | 91.75 | 129.9 | 60.2 | 69.7 |
| | 3500 | 11.59 | 191.25 | 136.5 | 196.4 | 87.2 | 109.2 | 6.34 | 148.5 | 94.25 | 126.6 | 66.8 | 59.8 |
| | 4000 | 11.85 | 194.0 | 139.25 | 193.2 | 93.5 | 97.7 | 5.82 | 151.0 | 97.50 | 122.7 | 74.7 | 48.0 |
| | 4500 | 11.50 | 197.0 | 143.0 | 188.1 | 103.8 | 84.4 | 4.69 | 153.75 | 100.25 | 118.1 | 83.7 | 34.4 |
| | 5000 | 10.48 | 200.25 | 146.5 | 183.1 | 113.9 | 69.2 | 2.89 | 156.75 | 104.0 | 113.0 | 93.9 | 19.1 |
| 130 Degrees Arc of Contact | 250 | 0.82 | 191.5 | 136.75 | 196.0 | 87.9 | 108.1 | 0.55 | 145.75 | 91.25 | 130.7 | 58.7 | 72.0 |
| | 500 | 1.79 | 189.0 | 133.5 | 199.5 | 81.0 | 118.5 | 1.20 | 144.25 | 89.25 | 132.9 | 54.2 | 78.8 |
| | 750 | 2.82 | 187.5 | 132.0 | 201.5 | 77.1 | 124.4 | 1.87 | 143.25 | 88.25 | 134.1 | 51.8 | 82.3 |
| | 1000 | 3.83 | 187.0 | 131.25 | 202.2 | 75.7 | 126.5 | 2.53 | 143.0 | 87.75 | 134.5 | 51.1 | 83.4 |
| | 1200 | 4.64 | 186.5 | 130.75 | 202.6 | 74.9 | 127.7 | 3.05 | 143.0 | 87.75 | 134.6 | 50.8 | 83.8 |
| | 1400 | 5.42 | 186.5 | 130.75 | 202.6 | 74.8 | 127.8 | 3.54 | 143.0 | 87.75 | 134.4 | 51.1 | 83.3 |
| | 1600 | 6.18 | 186.75 | 131.0 | 202.5 | 75.1 | 127.4 | 4.00 | 143.5 | 88.25 | 134.2 | 51.7 | 82.5 |
| | 1800 | 6.91 | 187.0 | 131.25 | 202.2 | 75.6 | 126.6 | 4.43 | 143.5 | 88.50 | 133.8 | 52.5 | 81.3 |
| | 2000 | 7.59 | 187.25 | 131.75 | 201.8 | 76.5 | 125.3 | 4.82 | 144.0 | 89.0 | 133.2 | 53.6 | 79.6 |
| | 2500 | 9.10 | 188.5 | 133.25 | 200.1 | 79.9 | 120.2 | 5.60 | 145.5 | 90.5 | 131.3 | 57.5 | 73.8 |
| | 3000 | 10.27 | 190.5 | 135.25 | 197.7 | 84.7 | 113.0 | 6.02 | 147.0 | 92.75 | 128.8 | 62.5 | 66.3 |
| | 3500 | 11.02 | 192.5 | 137.75 | 194.6 | 90.7 | 103.9 | 6.03 | 149.25 | 95.0 | 125.6 | 68.8 | 56.8 |
| | 4000 | 11.26 | 195.0 | 140.5 | 191.0 | 98.0 | 93.0 | 5.53 | 151.5 | 98.25 | 121.9 | 76.2 | 45.6 |
| | 4500 | 10.94 | 198.0 | 144.0 | 186.8 | 106.5 | 80.3 | 4.46 | 154.25 | 100.50 | 117.6 | 84.9 | 32.7 |
| | 5000 | 9.97 | 201.0 | 147.25 | 182.0 | 116.1 | 65.8 | 2.75 | 156.75 | 104.00 | 112.7 | 94.6 | 18.2 |
| 120 Degrees Arc of Contact | 250 | 0.77 | 193.0 | 138.5 | 193.8 | 92.5 | 101.3 | 0.51 | 146.75 | 92.5 | 129.2 | 61.7 | 67.5 |
| | 500 | 1.68 | 190.75 | 135.75 | 197.1 | 85.8 | 111.3 | 1.12 | 145.5 | 90.5 | 131.2 | 57.5 | 73.7 |
| | 750 | 2.65 | 189.5 | 134.25 | 198.9 | 82.2 | 116.7 | 1.76 | 144.5 | 89.5 | 132.5 | 55.1 | 77.4 |
| | 1000 | 3.61 | 189.5 | 133.0 | 199.7 | 80.6 | 119.1 | 2.38 | 144.25 | 89.25 | 132.9 | 54.3 | 78.6 |
| | 1200 | 4.37 | 188.5 | 132.75 | 200.1 | 79.9 | 120.2 | 2.87 | 144.0 | 89.25 | 133.0 | 54.1 | 78.9 |
| | 1400 | 5.11 | 188.5 | 132.75 | 200.2 | 79.7 | 120.5 | 3.34 | 144.0 | 89.25 | 132.9 | 54.2 | 78.7 |
| | 1600 | 5.83 | 188.5 | 133.25 | 200.1 | 79.9 | 120.2 | 3.77 | 144.5 | 89.5 | 132.6 | 54.8 | 77.8 |
| | 1800 | 6.53 | 188.5 | 133.25 | 199.9 | 80.2 | 119.8 | 4.18 | 144.75 | 89.75 | 132.2 | 55.5 | 76.7 |
| | 2000 | 7.17 | 189.0 | 133.5 | 199.4 | 81.2 | 118.2 | 4.55 | 145.0 | 90.25 | 131.7 | 56.6 | 75.1 |
| | 2500 | 8.60 | 190.25 | 135.0 | 197.8 | 84.4 | 113.5 | 5.29 | 146.25 | 91.75 | 129.9 | 60.2 | 69.7 |
| | 3000 | 9.71 | 191.75 | 137.0 | 195.6 | 88.8 | 106.8 | 5.69 | 148.0 | 93.5 | 127.5 | 64.9 | 62.6 |
| | 3500 | 10.41 | 193.75 | 139.25 | 192.7 | 94.5 | 98.2 | 5.70 | 149.75 | 95.75 | 124.6 | 70.9 | 53.7 |
| | 4000 | 10.65 | 196.25 | 142.0 | 189.3 | 101.4 | 87.9 | 5.23 | 152.0 | 99.0 | 121.0 | 77.9 | 43.1 |
| | 4500 | 10.35 | 199.0 | 145.0 | 185.3 | 109.4 | 75.9 | 4.22 | 154.5 | 101.0 | 117.0 | 86.0 | 31.0 |
| | 5000 | 9.43 | 201.75 | 148.0 | 180.7 | 118.5 | 62.2 | 2.60 | 157.0 | 104.25 | 112.4 | 95.2 | 17.2 |

**Table 3.—Service Factors for Belt Installations**

To be used in connection with Table 4

| Type of Equipment | Service Factor | Type of Equipment | Service Factor |
|---|---|---|---|
| Brick and clay machinery | 1.2–1.3 | Pulp and paper | 1.2–1.4 |
| Coal mining machines | 1.0–1.4 | Pumps and compressors | 1.2–1.3 |
| Cotton ginning | 1.0–1.2 | Rock crushing | 1.2–1.4 |
| Fans and blowers | 1.2–1.4 | Textile | 1.2–1.5 |
| Flour and feed mill | 1.2–1.3 | Woodworking | 1.2–1.3 |
| Laundry | 1.1–1.2 | Additional factors for type of motor* | |
| Machine tools | 1.0–1.3 | Reversing or line-start | 1.25 |
| Metal mining | 1.0–1.4 | High-torque A.C. or Compound-wound D.C. | 1.50 |
| Oil field | 1.2–1.5 | | |
| Printing | 1.0–1.3 | | |

* To be applied in addition to the factors for type of equipment.

(*Continued from page* 24–16)

in Table 2. Two reasons exist for this discrepancy. 1. Improvements in tanning and in methods of manufacture have produced more flexible belts with a higher initial coefficient of friction than that of belts made prior to 1930. 2. As noted above, the values of Table 2 are based, not on the maximum horsepower that can be transmitted by a belt, but on the conditions that will give maximum life of belt and minimum cost of service. In the use of the more modern belts, the same considerations should apply.

Table 4 is condensed from one issued by the Graton & Knight Co., and is typical of 1937 belting practice. It is used in connection with Table 3 and the correction factor chart, Fig. 1. A belt must be proportioned to pull loads in excess of the normal rated horsepower of the drive. The increase in horsepower depends on the type of load and the characteristics of the driving motor. Table 3 gives correction factors for various classes of service. Fig. 1 gives correction factors to be applied to the values of Table 4 to compensate for variation of the arc of contact from 180 deg.

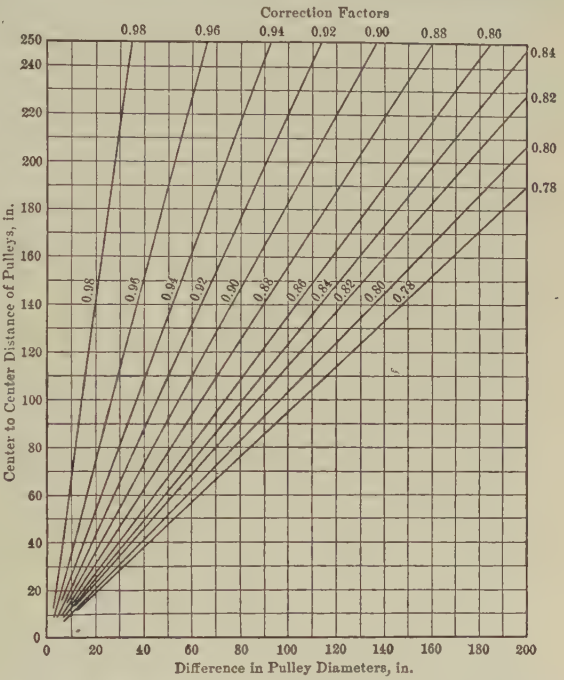


FIG. 1. Factors for Arc of Contact. To be used with Table 4

**Table 4.—Horsepower Transmitted by Leather Belting**

(180 deg. arc of contact)

NOTE.—This table should be used in connection with Table 3 and Fig. 1

| Belt Velocity, ft. per min. | Single Belt | | Double Belt | | | Triple Belt | | Belt Velocity, ft. per min. | Single Belt | | Double Belt | | | Triple Belt | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Medium | Heavy | Light | Medium | Heavy | Medium | Heavy | | Medium | Heavy | Light | Medium | Heavy | Medium | Heavy |
| | Thickness, in. | | | | | | | | Thickness, in. | | | | | | |
| | 11/64 | 13/64 | 18/64 | 20/64 | 23/64 | 30/64 | 34/64 | | 11/64 | 13/64 | 18/64 | 20/64 | 23/64 | 30/64 | 34/64 |
| | Horsepower per Inch of Width | | | | | | | | Horsepower per Inch of Width | | | | | | |
| 500 | 1.0 | 1.1 | 1.4 | 1.6 | 1.8 | 2.0 | 2.2 | 3400 | 5.8 | 6.5 | 8.3 | 9.3 | 10.4 | 11.7 | 12.9 |
| 600 | 1.2 | 1.3 | 1.7 | 1.9 | 2.1 | 2.3 | 2.6 | 3600 | 6.0 | 6.7 | 8.6 | 9.6 | 10.8 | 12.1 | 13.4 |
| 700 | 1.3 | 1.5 | 1.9 | 2.2 | 2.4 | 2.7 | 3.0 | 3800 | 6.3 | 7.0 | 9.0 | 10.0 | 11.2 | 12.6 | 13.9 |
| 800 | 1.5 | 1.7 | 2.2 | 2.4 | 2.7 | 3.1 | 3.4 | 4000 | 6.5 | 7.2 | 9.2 | 10.4 | 11.5 | 13.0 | 14.4 |
| 900 | 1.7 | 1.9 | 2.5 | 2.8 | 3.0 | 3.5 | 3.9 | 4200 | 6.7 | 7.4 | 9.5 | 10.6 | 11.8 | 13.3 | 14.8 |
| 1000 | 2.0 | 2.2 | 2.8 | 3.2 | 3.4 | 3.9 | 4.4 | 4400 | 6.8 | 7.6 | 9.7 | 10.9 | 12.2 | 13.7 | 15.2 |
| 1200 | 2.3 | 2.6 | 3.3 | 3.7 | 4.2 | 4.7 | 5.2 | 4600 | 7.0 | 7.8 | 10.0 | 11.2 | 12.5 | 14.0 | 15.6 |
| 1400 | 2.7 | 3.0 | 3.8 | 4.3 | 4.8 | 5.4 | 6.0 | 4800 | 7.1 | 7.9 | 10.1 | 11.4 | 12.6 | 14.1 | 15.8 |
| 1600 | 3.1 | 3.4 | 4.4 | 4.9 | 5.4 | 6.1 | 6.8 | 5000 | 7.2 | 8.0 | 10.2 | 11.5 | 12.8 | 14.4 | 16.0 |
| 1800 | 3.4 | 3.8 | 4.9 | 5.5 | 6.1 | 6.8 | 7.6 | 5200 | 7.2 | 8.0 | 10.2 | 11.5 | 12.8 | 14.4 | 16.0 |
| 2000 | 3.8 | 4.2 | 5.4 | 6.0 | 6.7 | 7.5 | 8.4 | 5400 | 7.3 | 8.1 | 10.4 | 11.6 | 13.0 | 14.6 | 16.2 |
| 2200 | 4.1 | 4.6 | 5.9 | 6.6 | 7.3 | 8.2 | 9.2 | 5600 | 7.3 | 8.1 | 10.4 | 11.6 | 13.0 | 14.6 | 16.2 |
| 2400 | 4.4 | 4.9 | 6.3 | 7.1 | 7.8 | 8.8 | 9.8 | 5800 | 7.3 | 8.1 | 10.4 | 11.6 | 13.0 | 14.6 | 16.2 |
| 2600 | 4.8 | 5.3 | 6.7 | 7.6 | 8.5 | 9.5 | 10.6 | 6000 | 7.3 | 8.1 | 10.4 | 11.6 | 13.0 | 14.6 | 16.2 |
| 2800 | 5.0 | 5.6 | 7.2 | 8.1 | 9.0 | 10.1 | 11.2 | 6200 | 7.2 | 8.0 | 10.2 | 11.5 | 12.8 | 14.4 | 16.0 |
| 3000 | 5.3 | 5.9 | 7.5 | 8.5 | 9.4 | 10.6 | 11.8 | 6400 | 7.0 | 7.8 | 10.0 | 11.2 | 12.5 | 14.0 | 15.6 |
| 3200 | 5.6 | 6.2 | 7.9 | 8.9 | 9.9 | 11.2 | 12.3 | 6600 | 6.8 | 7.6 | 9.7 | 10.9 | 12.2 | 13.7 | 15.2 |
| Minimum Pulley Diameter, Belts under 8 in. wide | | | | | | | | | | | | | | | |
| | 3 | 5 | 6 | 8 | 12 | 20 | 24 | | 3 | 5 | 6 | 8 | 12 | 20 | 24 |
| Minimum Pulley Diameter, Belts 8 in. wide and over | | | | | | | | | | | | | | | |
| | 5 | 7 | 8 | 10 | 14 | 24 | 30 | | 5 | 7 | 8 | 10 | 14 | 24 | 30 |

**Example.**—A compressor is to be driven by a 5-Hp. high-torque, line-start motor. Motor pulley, 5 1/4 in. diam., 1160 r.p.m. Compressor pulley, 20 in. diam. Center to center distance, 40 in. Required the width and type of belt necessary.

The belt speed is 1594 ft. per min., or say 1600 ft. From Table 4, the heaviest belt that can be used on a 5 1/4-in. pulley is a single belt, 13/64 in. thick. At 1600 ft. per min., the normal capacity of this belt is 3.4 Hp. per inch of width. Difference in the pulley diameters is (20 − 5 1/4) = 14 3/4 in. From Fig. 1, the correction factor for arc of contact is 0.94. The horsepower transmitted by the belt then is 3.4 × 0.94 = 3.2. From Table 3, the service factor for compressors is taken as 1.3. Additional correction factors of 1.25 for line start, and of 1.50 for a high-torque motor, therefore, must be applied. The rated horsepower of the motor is multiplied by these factors to find the maximum horsepower for which the belt must be selected. This figure, divided by the rated capacity of the belt, gives the width of belt necessary, as follows:

$$5 \times 1.3 \times 1.25 \times 1.50/3.2 = 3.8 \text{ in.},$$

or, say, a 4-in. single belt, 13/64 in. thick.

**Table 5.—Width of Pulley to be Used with a Given Width of Belt**

(Table calculated from Barth's formulas. See page 24–09.)

| Width of Belt, $B$ in. | Width of Pulley, in. | | Width of Belt, $B$ in. | Width of Pulley, in. | | Width of Belt, $B$ in. | Width of Pulley, in. | |
|---|---|---|---|---|---|---|---|---|
| | Formula, $F = 1\,3/32\,B + 3/16$ | Formula * $F = 1\,3/16\,B + 3/8$ | | Formula, $F = 1\,3/32\,B + 3/16$ | Formula * $F = 1\,3/16\,B + 3/8$ | | Formula, $F = 1\,3/32\,B + 3/16$ | Formula, * $F = 1\,3/16\,B + 3/8$ |
| 1 | 1 9/32 | 1 9/16 | 7 | 7 27/32 | 8 11/16 | 24 | 26 7/16 | 28 7/8 |
| 1 1/2 | 1 53/64 | 2 5/32 | 8 | 8 15/16 | 9 7/8 | 26 | 28 5/8 | 31 1/4 |
| 2 | 2 3/8 | 2 3/4 | 9 | 10 1/32 | 11 1/16 | 28 | 30 13/16 | 33 5/8 |
| 2 1/2 | 2 59/64 | 3 11/32 | 10 | 11 1/8 | 12 1/4 | 30 | 33 | 36 |
| 3 | 3 15/32 | 3 15/16 | 11 | 12 7/32 | 13 7/16 | 36 | 36 9/16 | 43 1/8 |
| 3 1/2 | 4 1/64 | 4 17/32 | 12 | 13 5/16 | 14 5/8 | 42 | 46 1/8 | 50 1/4 |
| 4 | 4 9/16 | 5 1/8 | 14 | 15 1/2 | 17 | 48 | 52 11/16 | 57 3/8 |
| 4 1/2 | 5 7/64 | 5 23/32 | 16 | 17 11/16 | 19 3/8 | 54 | 59 1/4 | 64 1/2 |
| 5 | 5 21/32 | 6 5/16 | 18 | 19 7/8 | 21 3/4 | 60 | 65 13/16 | 71 5/8 |
| 5 1/2 | 6 13/64 | 6 29/32 | 20 | 22 1/16 | 24 1/8 | 66 | 72 3/8 | 78 3/4 |
| 6 | 6 3/4 | 7 1/2 | 22 | 24 1/4 | 26 1/2 | 72 | 78 15/16 | 85 7/8 |

* Preferred formula

## 3. BELT MAINTENANCE

Correct maintenance of leather belts requires that tensions be held within predetermined limits, and that they be cleaned and greased at regular intervals. Further, regular inspection of belts and pulleys will indicate when belts should be shortened, and also reveal conditions that tend to destroy or damage belts, and permit them to be corrected before harm is done.

**BELT BENCH.**—A convenient method of obtaining the correct tension in belts is the use of a belt bench. This consists of a work bench, of length exceeding the longest center-to-center distance of any belt drive in the shop. On the face of the bench are mounted two non-rotatable pulleys about 18 in. diameter. One of these is fixed. The other is so mounted that it can be moved to any point in the length of the bench, and there locked in position. A scale on the bench reads from zero at the center of the fixed pulley. The reading on the scale opposite the movable pulley is the length of the belt that then can be put around the two pulleys.

In using the bench, the movable pulley is placed at a point on the scale that corresponds to the length of the belt. This length is determined by measurement with a steel tape around the actual pulleys on which the belt is to be used. The belt is placed over the belt bench pulleys, and drawn up to the proper tension by belt clamps. See Fig. 2. The belt clamps are connected by spring balances, one end of each balance being attached to the clamp on the short end of the belt. The other end of each balance is attached to a long draw bolt which passes through the clamp on the long end of the belt. As

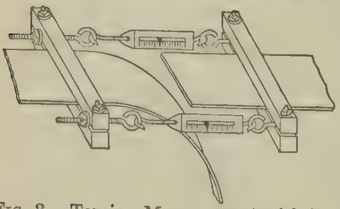


Fig. 2. Tension Measurement with Belt Clamps and Spring Balances

the draw bolts are screwed up by the nuts, the tension in the belt increases and is read on the balances. When the desired tension has been put in the belt, the point where the end of the belt overlaps the main part is marked. The proper addition for lap (see p. 24–23), or deduction for hinge joint is made, and the belt cut off. When the joint has been made, the belt can be placed on the pulleys of the drive, and the tension will be the same as was registered on the spring balances of the belt bench.

Belts which have stretched in service, can be shortened and restored to the proper tension at the belt bench in the same manner.

**TIGHTENING OF BELTS.**—To ascertain the amount that is to be taken out of a belt to restore it to its maximum tension, subtract from the maximum tension the tension in the belt before tightening. Find the figure corresponding to this difference at the bottom of the chart, Fig. 3, and run vertically upward on the chart to the line corresponding to the cross-section of the belt. At the left of the chart, opposite the intersection of the tension and the area so found, read the stretch per 18 in. of length of belt. Multiply this figure by the quotient of the length of the belt in inches divided by 18, or of its length in feet divided by 1.5. The result will be the amount that must be taken out of the belt to restore it to its maximum initial tension.

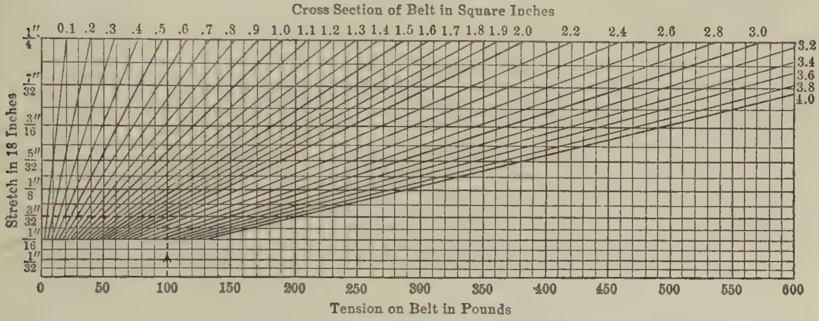


Fig. 3. Tension and Stretch of Belts

Example.—A belt 36 ft. long of 2 in. cross-sectional area shows a total tension of 260 lb. Required the amount by which it must be shortened to restore it to a tension of 360 lb. Increase in tension = 360 − 260 = 100 lb. Enter chart on ordinate of 100 lb. tension, and follow it until it intersects the 2-in. cross-sectional area diagonal. On the scale at the left, opposite this intersection read $^3/_{32}$ in. Then

$$^3/_{32} \times {}^{36}/_{1.5} = 2\,{}^1/_4 \text{ in.} = \text{amount to be taken from belt.}$$

**CLEANING OF BELTS.**—Every precaution should be taken to prevent belts in service becoming saturated with lubricating oil, or, in fact, from coming in contact at all with any kind of oil. If, however, a belt does become so saturated it may be cleaned by first scraping it and then packing it in sawdust or other absorbent for several days. A quicker method is to pass the belt between rolls, under pressure, thereby squeezing out the oil which then may be absorbed by powdered chalk. The surface dirt may be removed from belts by wiping them with a cloth moistened with kerosene. This should be done in every case before applying belt grease.

**BELT DRESSING.**—The use of a belt dressing is to increase the coefficient of friction of a belt and thus enable it to pull a heavier load with a given tension. The dressing should be applied only when the belt slips during working hours. Belts which are regularly inspected and whose tensions are maintained in accordance with the figures of Table 2 will seldom slip, and need no dressing except the regular cleaning and greasing every five or six months. If, however, the belt should slip during working hours as a result of neglect or of being subjected to a heavier load than that for which it was intended, a dressing should be applied sparingly to enable it to continue to pull its load until the machine can be shut down and the belt tightened.

The Tabor Manufacturing Co., Philadelphia, recommends the cleaning of belts at intervals of six months and the application of a grease that will give to the surface a clean sticky pulling surface, similar to that which it has when new, due to the oils and grease that it contains. The grease in the new belt becomes exhausted after the belt has been in service for a certain length of time, as the result of heat, friction and atmospheric action, and the application of additional grease is necessary if the belt is to be maintained in a soft and pliable condition. The grease recommended is the same as that put in the new belt by the manufacturers and consists of 2 lb. edible beef tallow and 1 lb. cod liver oil. The tallow is melted and allowed to cool slightly, after which the cod liver oil is added and the mixture stirred in one direction until cold. It should be applied in light coats by means of a camel's hair brush, and only in such quantities as the belt will easily absorb.

**The Use of Rosin** on slipping belts should be absolutely prohibited. While it may temporarily stop the slipping it will in a very short time ruin the belt.

**TAYLOR'S RULES FOR BELTING.**—F. W. Taylor (*Trans.* A.S.M.E., xv, 204) describes a nine-year experiment on belting in a machine shop, giving results of tests of 42 belts running night and day. Some of the belts ran on cone pulleys and others on shifting or tight-and-loose pulleys. The average net working load on the shifting belts was only 0.4 of that on the cone belts. The experiments are summarized below.

Belt tensions were weighed on spring balances between belt clamps when the belts were first put on, and when they were retightened. Belts were so proportioned that the initial tension with the belt at rest was 71 lb. per inch of width of double belt, equivalent to the following stresses, lb. per sq. in.: Oak tanned and fulled belts, 192; oak tanned, not fulled, belts, 229; semi-rawhide belts, 253; rawhide belts, 284. Conclusions drawn from the experiments were:

Belt speeds for maximum economy should be from 4000 to 4500 ft. per min. The best distance from center to center of shafts is 20 to 25 ft. Double belts should be used on pulleys 12 in. diameter or larger; triple belts on pulleys 20 in. diameter or larger; quadruple belts on pulleys 30 in. diameter or larger. Belts are more durable and work more satisfactorily when made narrow and thick, rather than wide and thin. As belts increase in width they should be made thicker. The ends of the belt should be fastened together by splicing and cementing instead of by lacing, wiring or hooks or clamps of any kind. A V-splice should be used on triple and quadruple belts and when idlers are used. A stepped splice, coated with rubber and vulcanized in place is best for rubber belts. Splices in double belts up to 10 in. wide should be 10 in. long. For belts 10 to 18 in. wide, length of splice should be equal to the width of the belt. For belts over 18 in. wide, splice should be 18 in. long. Belts should be cleaned and greased every five to six months. Double leather belts will last well when repeatedly tightened under a stress, when at rest, of 71 lb. per inch of width, or 240 lb. per sq.in. of cross-section, but they will not maintain this tension for any length of time. Idler pulleys work most satisfactorily when located on the back side of the belt about one-quarter of the distance from the driving pulley.

The comparative performance of belts proportioned according to ordinary practice, giving a total load of 111 lb. per inch of width, and belts proportioned to give a total load of 54 lb. per inch of width, are given in Table 6.

Belts of any width can be shifted backward and forward on tight-and-loose pulleys. The best form of belt shifter for wide belts is a pair of rollers twice the width of the belt, either of which can be pressed against the flat surface of the belt on its slack side close to the driven pulley. The axis of the roller should make an angle of 75 deg. with the center line of the belt.

**Conditions that Injure Belts.**—Belts that are running under improper conditions will give certain indications which may be used to identify and correct such conditions. If the tension on the belt is higher than it should be, the outside surface of the belt will tend to crack. If the laps also open up, the belt probably is too thick for the diameter of the pulleys. Ply separation also indicates too thick a belt. A belt cracked and checked on the pulley side indicates excessive slip of the belt. If the belt runs against the flange of a pulley or runs in contact with improperly placed belt shippers or guides, it will curl and the plies will open up on the edge. A belt running in the wrong direction will be indicated by the outside points of the laps opening. Dry rot of the belt indicates lack of dressing, and oil rot indicates exposure of the belt to lubricating oil. A belt too small for the power to be transmitted is indicated by short life both of the leather and of the cement.

## 4. BELT JOINTS

Four methods of joining the ends of belts are in general use, *i.e.*, cementing or gluing (endless belts), rawhide lacing, metal lacing and wire hinge joints. The endless belt is recommended wherever it can be used, but it is difficult to readjust the tension without taking the joint apart, unless the belt runs on pulleys which are adjustable and can be moved apart to compensate for belt stretch. Rawhide lacing, formerly the common method of making belt joints, has largely been replaced by the wire hinge joint. Metallic lacing is comparatively little used, and is not especially suited to belts running on small diameter pulleys.

**ENDLESS BELTS.**—The length of splice depends on the width and thickness of the belt. Table 7 gives the length of splices as recommended by the Graton & Knight Co.

For making the joints, belt manufacturers offer special cements, or the following glue, recommended by the Tabor Mfg. Co., may be used: Two parts liquid fish glue, one part liquid isinglass, by measure. The glue should be applied hot.

Before the joint is made the belt should be placed over the pulleys and drawn up to the proper tension by means of belt clamps, or it should be measured on a belt bench (see p. 24–20). Tension should be measured by spring balances between the clamps. When the belt has been drawn up to the proper tension, its length under tension should be noted and the proper allowance added for lap. The ends of the belt then are scarfed as shown in Fig. 4. The scarfed surfaces should be thoroughly sized with cement or glue and the two ends squeezed together between boards, pressure being applied by means of screw clamps. In using cement, after sizing, the belt joints are cemented a few inches at a time, each application being allowed to set for one half hour before the next section is cemented.

Endless belts should be run so that the feather edge of the splice on the side next to the pulley points away from the pulley as the splice approaches it.

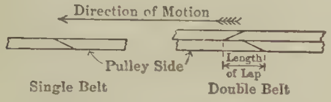


Fig. 4. Belt Laps

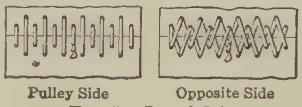


Fig. 5. Laced Joint

**LACED JOINTS.**—The ordinary belt lacing is a strip of rawhide whose width depends on the size of the belt to be laced. The ends of the belt are cut square with the edges and butted together, after which holes are punched in the two ends with an oval punch, the large diameter of the oval being parallel with the sides of the belt. The lacing is started at the center, worked over to one side of the belt, back to the opposite side, and is finally brought to the center at the point at which it started. The lacing is carried out as shown in Fig. 5. The lacing is parallel to the sides of the belt on the face next to the pulley and crossed on the opposite side. Table 8 (A. D. Porter, *Railway Age-Gazette*, Nov. 4, 1910) shows the size of lacing and the punching of the holes for lacing various sizes of belts.

**Table 6.—Performance of Belts under Different Conditions**

| | Total load per inch of width | |
|---|---|---|
| | 111 lb. | 54 lb. |
| | Effective pull per in. of width | |
| | 65 lb. | 26 lb. |
| Shorten double belt to restore tension, in. per 10 ft. of belt | 1/2 | 1 |
| Life of double belt when carefully handled, years | 7 | 18 |
| Cost of maintenance, % of original cost | 37 | 14 |
| Total stretch of belting, % of original length | 6 | 6 |
| Stretch during first 6 months, % of total stretch | 36 | 15 |
| Stretch allowable before retightening, % of original length | 0.47 | 0.81 |
| Interruptions to manufacture per year | 5 | 0.75 |

**Table 7.—Length of Belt Splices**

| Single Belts | | Double Belts | | Triple Belts | |
|---|---|---|---|---|---|
| Width of Belt, *W*, in. | Length of End-lap, in. See Fig. 4 | Width of Belt, *W*, in. | Length of End-lap, in. See Fig. 4 | Width of Belt, *W*, in. | Length of End-lap, in. See Fig. 4 |
| Under 4 | 4 | Under 5 | 8 | Under 5 | 8 |
| 4–10 | 6 | 5–12 | 12 | 5–12 | 12 |
| Over 10 | 8 | 12–24 | *W* | 12–24 | *W* |
| | | Over 24 | 24 | Over 24 | 24 |

**Table 8.—Data for Lacing Belts**

| Width of Belt, in. | Punching of Lace Holes, Distance from Ends of Belt, in. | | Distance of Center Line of First Hole in First Row from Edge of Belt, in. | Size of Belt, Lace, in. |
|---|---|---|---|---|
| | First Row | Second Row | | |
| 2 to 4 | 3/8 | 3/4 | 3/8 | 3/16 |
| 6 " 8 | 1/2 | 1 | 1/2 | 1/4 |
| 10 " 12 | 5/8 | 1 1/4 | 5/8 | 5/16 |
| 14 " 16 | 3/4 | 1 1/2 | 3/4 | 3/8 |
| 18 " 20 | 7/8 | 1 3/4 | 7/8 | 7/16 |
| 22 " 24 | 1 | 2 | 1 | 1/2 |

The strength of a laced joint is about one-half that of the belt itself, or about 1000 to 1500 lb. per sq. in.

The objections to the laced joint are that it is seldom well made, and that it is difficult to lace a belt so as to maintain the same tension in it on the two edges of the belt. It also virtually increases the diameter of the pulley at the point at which the joint is in contact with it, since the lacing cannot be made flush with the surface of the belt. The lacing causes the belt on either side of the joint to be raised from the pulley, and destroys the intimate contact of belt and pulley. The joint also is less flexible than the remainder of the belt, and this has a more or less detrimental effect on the belt and the service rendered by it. Belts as a rule fail at or near the lacing. Added to the other objections, the making of a laced joint is slow and expensive.

**METALLIC BELT LACING** consists of a strip of steel cut into a series of Vs. The points of the Vs on each side are bent at right angles to the body of the lacing. The ends of the belt are butted together and a strip of the metallic lacing whose length is equal to the width of the belt is laid across the joint. The points are then driven through the belt, clinched on the pulley side, and driven into the belt so that there will be no projections to come in contact with the pulley as the joint passes over it. The metallic lacing provides a rapid and cheap means of making a joint, but it is less flexible than the rawhide lacing. It is also difficult to break the joint for purposes of repair without seriously damaging the belt at the joint.

**WIRE-LACED HINGE JOINTS.**—The most satisfactory method of making a belt joint, all things considered, is by the use of wire which is looped through each end of the belt in a series of eyelets which project beyond the end of the belt. The eyelets in the two ends are brought together and a pin of rawhide is run through them, thus joining the ends of the belt and forming a true hinge. The finished joint does not project above the surface of the pulley, and as the joint can be bent through an arc of about 270 deg. there is no diminution in the flexibility of the belt as a whole. Machines for making joints of this character are made by the Peerless Belt Lacing Machine Co., Philadelphia, and by the Birdsboro Foundry & Machine Co., Birdsboro, Pa.

The pin should be cut off about $^1/_8$ in. short on each side of the belt. The strength of a joint made in the above fashion is stated to be about 700 lb. per lineal inch of joint. For belts that are to be used in damp places brass, instead of steel, wire is recommended.

## 5. ARRANGEMENT OF BELT DRIVES

**ARRANGEMENT OF PULLEYS.**—If possible, machinery driven from overhead shafting should be so arranged that the belt will make an angle from the vertical of not less than 45 deg. Never, unless it is absolutely impossible to avoid it, should driving and driven pulleys be in the same vertical line. Care should be taken to arrange the pulleys in line with each other, the tops of the crowns being aligned. Otherwise the belt will tend to stretch unevenly and to run crooked. Machinery should be so placed that the tight strand of the belt will be the bottom strand. If, however, machines are so placed that the pull of the belts at alternate line shaft pulleys is in opposite directions, the wear and strain on bearings will be reduced. This arrangement of machinery will make one-half of the belts drive with the top strand tight.

The distance between shafts has considerable influence on the life of belts. Distances should, as a rule, be made as great as possible up to reasonable limits. For narrow belts on small pulleys, a good average figure is 15 ft.; for larger belts the center line distance should be from 20 to 25 ft., while for main belt drives using wide and heavy belts, pulleys should be from 25 to 30 ft. apart. If the distance is made too great the belt will have a tendency to flap and injure itself.

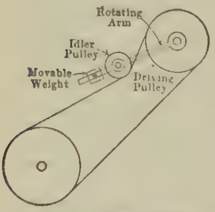


Fig. 6. Idler Pulley

**IDLER PULLEYS.**—Belts which are run on pulleys located close together are liable to stretch rapidly and thus lose the tension necessary to the transmission of the desired amount of power. For such cases, and also for belts which are required to run for long periods without stopping, an idler pulley, which can be moved against the belt to take up the slack and thus increase the tension, will be found desirable. The idler should be located about one-quarter of the distance from the driving pulley and should bear against the slack side of the belt. Its effect is not only to increase the tension, but also the arc of contact. The idler may be so constructed that it can be moved against the belt at will, or it may act automatically by means of a spring or gravity. Such an arrangement is shown in Fig. 6.

**GUIDE PULLEYS.**—Where two shafts that are not parallel and that do not intersect are to be connected by belting, and the conditions are such as to make a quarter-twist belt inadvisable, guide pulleys are used. The function of the guide pulley is to alter the direction of the belt and cause it to run squarely on to the receiving pulley. The location of the guide pulley with reference to the main pulleys is determined by the intersection of the planes passing through the center of the face of the main pulleys and perpendicular to their axes. Thus in Fig. 7, let $A$ and $B$ represent the main pulleys, and $xx$ and $yy$ the planes passed through the middle points of their respective faces. These intersect in the line $CD$. Tangents from any two points as $E$ and $F$ on this line to the points $H$ and $J$, and $G$ and $K$, respectively, on the pulleys $A$ and $B$ will determine proper paths for the belt. The guide pulleys should be located at the intersection of the tangents $HE$ and $EJ$ and $FG$ and $FK$, respectively, and the plane through the center of the face of each pulley should coincide with the planes formed by these tangents. Figs. 8 to 11 show various arrangements of guide pulleys. Others will suggest themselves according to the needs of each particular case. It is only necessary to observe the precaution that the guide pulleys be so placed that they will receive the belt squarely from the main pulley and deliver it squarely to the next one. If this be done, the pulleys may be placed in practically any location and at practically any angle.

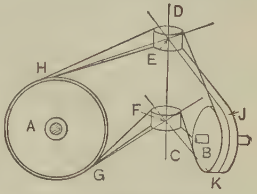


FIG. 7. Guide Pulleys

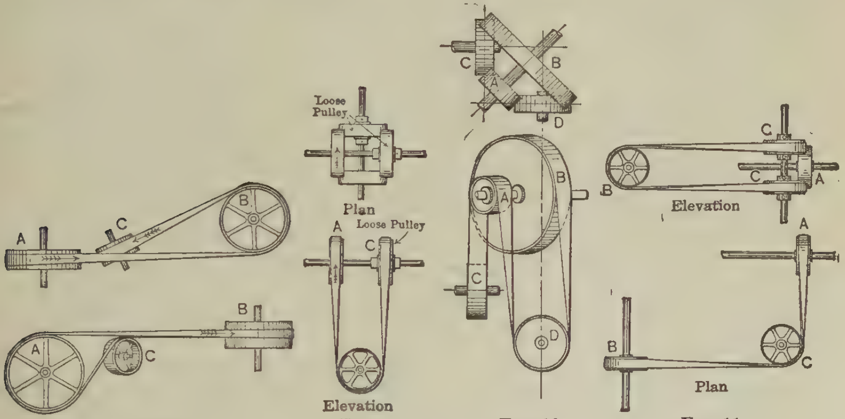


FIG. 8 FIG. 9 FIG. 10 FIG. 11

Drives with Guide Pulleys

**Quarter-Twist Belts.**—One of two shafts at right angles to each other and in different planes may be driven from the other by means of a belt with a quarter twist in it, providing that the distance between the shafts is not too short. The belt must run squarely on to each pulley and the point at which the belt leaves each pulley must lie in the straight line formed by the intersection of the planes passing through the center line of the face of each pulley and perpendicular to their axes, as shown in Fig. 12. The driving pulley is $AD$ and the belt runs in the direction shown by the arrows. The point $C$ at the end of the horizontal diameter of the driven pulley at which the belt leaves it is set so that it is vertically above the point $A$ at the end of the horizontal diameter of the driving pulley where the belt leaves that pulley. Should the shafts in question be respectively a horizontal and a vertical one the relative arrangement of the pulleys would be the same as shown in the illustration. It makes no difference what position the shafts take so long as the leaving points of the belts on the pulleys are in the straight line determined by the intersection of the two planes as described. The quarter-twist has been used with belts up to as much as 30 in. wide. There should be a considerable distance between the shafts, however. For heavy belts and short center line distances, guide pulleys or mule pulley stands should be used. A quarter-twist belt will drive in one direction only.

Where the quarter-twist belt drives through a floor the location of the holes in the floor can be determined by the following method (M. H. Ball, *Machy.*, Sept., 1912) as shown in Fig. 13. From the end of the horizontal diameter at which the belt leaves the upper pulley a plumb line is dropped to the floor, and point *L* on the floor is marked. Through *L* are drawn on the floor the lines *XX* and *YY* respectively perpendicular to the lines of shafting on which the upper and lower pulleys are carried. The outlines of the pulleys in plan are then traced on the floor, these being *aXbdLc* and *efYghL* respectively for the upper and lower pulleys, each being symmetrical about the center lines *XX* and *YY*. These center lines, which are the projections of the horizontal diameters of the pulleys, are next divided into eight equal parts, the divisions being numbered as shown. The numbers should start at the end of the diameters at which the belt runs on the pulley. Next divide the distance in inches between the center lines of the shafts by the number of divisions laid out on the pulley, and the distance from the center line of the upper shaft to the floor by the quotient so obtained. The latter figure represents the number of spaces on the pulley that the belt at the floor will be away from the end of the diameter at which it runs on to the pulleys, that is, the points *J* and *K* in Fig. 13. From *J* as a center with the line *JM*, perpendicular to *XX*, as a radius strike arc *MN*. The arc is divided into eight equal parts and numbered from *M* to *N*. The same number of spaces are taken on the arc as were taken on the lines *XX* and *YY*, and *OP* is drawn through the point *J* and the point on the arc so found, *PJ* being made equal to *JO*. This line is the position of the center line of one strand of the belt at the floor, while a line parallel to *OP* through *K*, and of the same length is the position of the center line of the other strand.

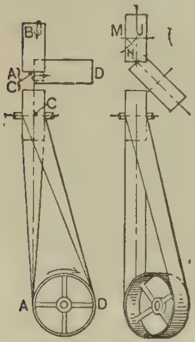


FIG. 12. Quarter-twist Belt

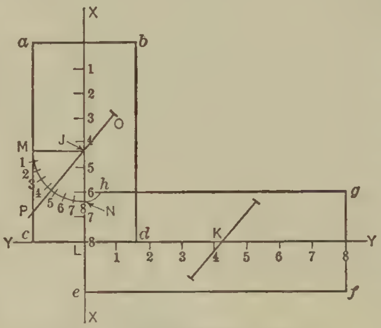


FIG. 13. Laying Out Floor Hole for Quarter-twist Belt

Quarter-twist belts may be used to drive shafts which make angles with one another of other than 90 deg., if the rule is observed that the points at which the belt leaves the pulleys are in the straight line determined as explained previously. The method of laying out holes in the floor is the same as that used for shafts at right angles, except that the arc *MN* on the pulley outline extends only to a line drawn from the point *J* parallel to the face of the lower pulley.

**Mule Pulley Stands.**—A common arrangement of guide pulleys, known as a mule pulley stand, is shown in Figs. 14 and 15. The method followed in locating the pulleys and determining the angles at which they are to be placed is as follows: The pulleys which are to be connected by a belt are *A* and *B*, and they have the relative locations as shown in the various views. The center line *XX* in the elevation is prolonged, and at a distance *L* (see plan) from the line *ZZ* through the middle plane of the face of the pulley *A*, the line *Z′Z′* is drawn. Through the point *p*, the line *YY* is drawn parallel to *XX*, and at a distance below it equal to the vertical distance between the shafts of the two pulleys. With *p* as a center the circle *A′* is drawn with a radius equal to the radius of the pulley *A*. The circles *B* and *A′* are then joined by the tangents *TT′* and *SS′*, which represent the inclination of the belt with respect to the pulleys at all points along its length. At the point *a* where these tangents intersect the line *ZZ* the lines *ab* are drawn parallel to the line *XX*. The diameter of the mule pulleys having been decided, the point *d* is set off on *ab* at a distance from *ZZ* equal to half the diameter of the mule pulley. Through *d* is drawn the line *ef* parallel to *ZZ*. This represents the center of the mule stand. The lines *gh* through the point *d*, perpendicular to the tangent *TT′* or *SS′* will be the axis of the mule pulley. The points *c* and *j* are laid off on either side of the point *d* on the line *gh*, *dc* and *dj*, being made equal to half the width of the mule pulley. The bearings of the mule pulley will lie on the line *gh* outside of the points *c* and *j*, and their position with reference to the center lines of the main pulleys can be ascertained by scaling the drawing or by calculation, as preferred. If the drawing is laid out to a large enough scale, scaling will be accurate enough for all practical purposes. The axes of the mule pulleys will necessarily be inclined from the vertical, not only as regards the plane *ZZ* through the middle of the face of the pulley *A*, as shown in the elevation Fig. 14, but also as regards the plane *WW* through the middle of the face of pulley *B* as shown in Fig. 15, and this angle will be the same in the two views. Lines through the points *c* and *j* parallel to the tangents *TT′* and *SS′* will be the major axes of the ellipses representing the upper and lower sides of the mule pulleys in the two views in Figs. 14 and 15. The position of the center of the mule stand and of the mule pulleys in the end view of the drive, Fig. 15, is determined in exactly the same manner as the similar points in the elevation in Fig. 14, and no further explanation is necessary. The various points in Fig. 15 are designated by the same letters as the corresponding points in Fig. 14, and the same explanation will apply.

If it be borne in mind that the governing principle of mule pulley drives, and in fact of all drives where the belt joins two pulleys that are not in the same plane, is that the belt must be delivered squarely on the pulleys, but that it can leave them at any angle within reasonable limits, it is evident that the mule pulleys can be placed in any location desired. If local conditions make it advisable, the mule pulleys can be so placed as to require the belt to leave either or both of the main driving pulleys in a horizontal position. The method of laying out the drive is the same in any event. The only variation will be in the relative length of the dimensions $L$ and $L'$. The sum of these is taken and the position of the circle $A'$ is then determined as before. The angle of the pulleys will be the same, as will be the length of the belts.

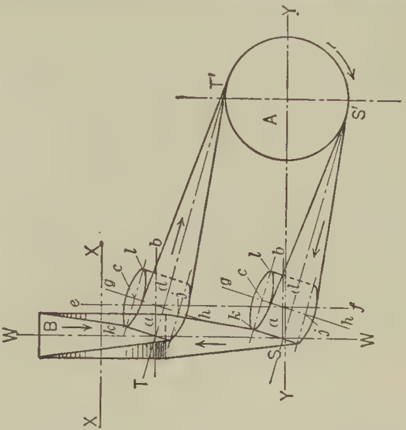


FIG. 15. Mule Pulley Drive

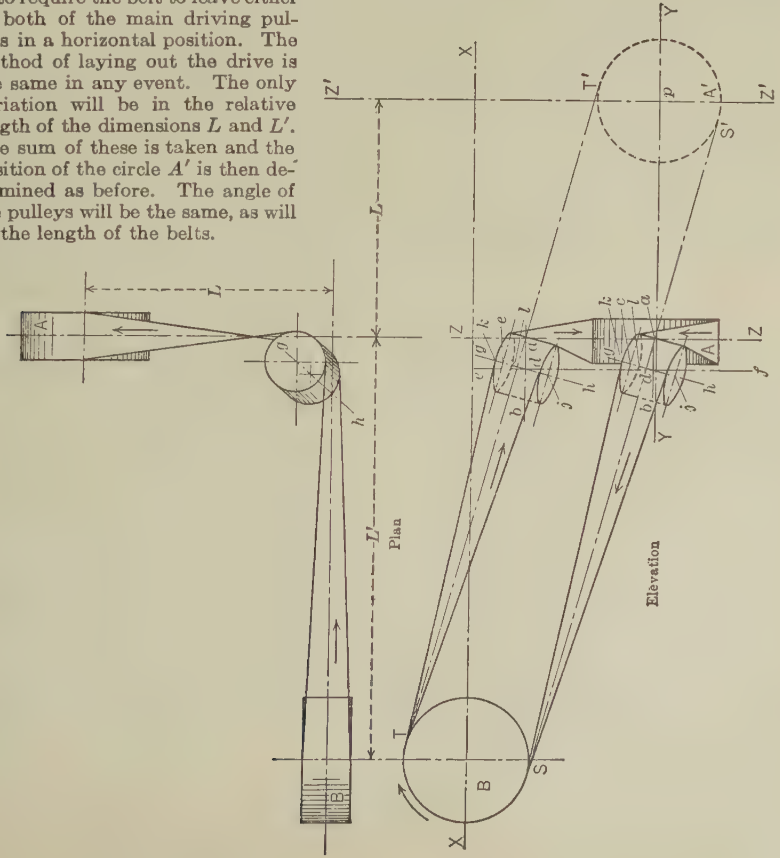


FIG. 14. Mule Pulley Drive

The fact that it is possible to vary the position of the mule pulleys without affecting the working of the drive makes it possible to place both mule pulleys on the same shaft. Referring to Fig. 14, if the belt travels in the direction of the arrows, it is evident that if the lower mule pulley be moved in the direction that the belt travels, the belt will still be delivered squarely to pulley *B*. It will, of course, leave pulley *A* at an angle, but this will have no effect on it if the mule pulley is maintained at the same angle as before, since the belt will still be received squarely by the mule pulley. If the distance between the axes of the mule pulley shafts is not excessive, the lower mule pulley may be moved until the two shafts are in line, as regards Fig. 14. They will, however, still be out of line as regards Fig. 15. By the same process of reasoning as before, however, it is evident that the upper mule pulley can be similarly moved in the direction of the belt travel until its axis is in line with that of the lower pulley in Fig. 15. The axes of the two mule pulleys then will coincide, and in their new locations they can be mounted on the same shaft.

The above considerations apply to drives where the main pulleys are of the same diameter. If they are unequal, the shafts of the mule pulleys will lie at different angles and the pulleys must be mounted on separate shafts. The location of the shafts is determined by the same method as described above, however, the only difference being that the axis of the upper mule pulley is determined by the perpendicular to the tangent $TT'$, Fig. 14, while that of the lower one is determined by the perpendicular to the tangent $SS'$. These tangents in this case will not be parallel. Otherwise the construction is the same as for drives with main pulleys of equal diameter.

## 6. SHORT CENTER DRIVES

Belt drives with flat belts, between shafts whose center lines are but a short distance apart usually prove unsatisfactory unless provision is made to increase the tension in the belt or to give a greater arc of contact on the driving pulley. Increased tension and greater arc of contact may be obtained at the same time by a weighted floating idler (see p. 24–24). Increased tension which automatically adjusts itself to the load is obtained by means of the Rockwood drive (see below). Since 1930 the drive known as the V-belt or Texrope drive has come into wide use for short center drives. Both the V-belt drive and the Rockwood drive may be operated with the driving and driven shaft located in the same vertical plane.

**V-BELT DRIVES.**—The V-belt drive consists of one or more endless belts constructed of cotton cords and fabric impregnated with and imbedded in rubber, and of trapezoidal cross-section. These run on sheaves with V-shaped grooves on the periphery. The angle of the groove depends on the service in which the belt is to be used, and manufacturers of V-belts should be consulted concerning details of a particular drive. The drives are so designed that the belt is approximately flush with the top of the pulley. A clearance is provided at the bottom of the grooves in the sheaves to insure contact with the belt only on the sides of the grooves. These drives will run in either direction, with the slack either on top or bottom strands. They are especially adapted to reversing service. The V-belt was first developed by the Allis-Chalmers Mfg. Co., whose product is sold under the name of Texrope drives. Other manufacturers are the Dayton Rubber Mfg. Co., Dayton, O., Manhattan Rubber Co., Passaic, N. J., Goodyear Rubber Co., Akron, O.

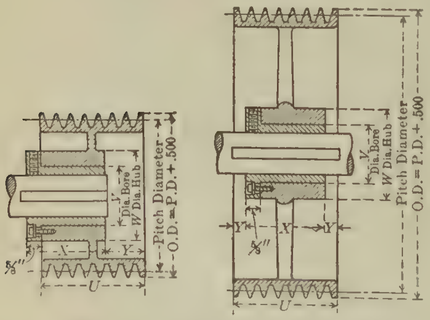


Fig. 16. Texrope Drive Sheaves

Table 9 and Fig. 16 give the dimensions for the sheaves used in connection with the drives listed in Table 12.

The following data were supplied by the Allis-Chalmers Mfg. Co. In the design of Texrope drives provision should be made for peak or shock loads in excess of 100% of full load, by the use of additional Texropes over those required for normal full load operation. Overload factors are as follows:

| | | | | | | |
|---|---|---|---|---|---|---|
| Peak load, percent | 100 | 125 | 150 | 175 | 200 | 250 |
| Overload factor | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 |

The horsepower rating of the motor is multiplied by the overload factor to obtain the normal rating of the required drive. Other factors that must be taken into consideration

**Table 9.—Dimensions of Cast-iron Sheaves for Texrope Drives**

2 to 25 Horsepower. 21/32 × 7/16-in. Texrope

(Allis-Chalmers Mfg. Co., Milwaukee, Wis.)

| Pitch Diam., in. | 2-Groove. U = 1 3/4-in. Face | | | | | | 3-Groove. U = 2 1/2-in. Face | | | | | | 4-Groove. U = 3 1/4-in. Face | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Class of Bushing† | V | W | X | Y | Net Wt., lb. | Class of Bushing† | V | W | X | Y | Net Wt., lb. | Class of Bushing† | V | W | X | Y | Net Wt., lb. |
| 5.4 | N | 3 1/2 | * | 2 3/8 | ...... | 11 | N | 3 1/2 | * | 2 3/8 | 3/4 | 13 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 14 |
| 5.6 | N | 3 1/2 | * | 2 3/8 | ...... | 12 | N | 3 1/2 | * | 2 3/8 | 3/4 | 14 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 15 |
| 5.8 | N | 3 1/2 | * | 2 3/8 | ...... | 13 | N | 3 1/2 | * | 2 3/8 | 3/4 | 15 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 16 |
| 6.0 | N | 3 1/2 | * | 2 3/8 | ...... | 14 | N | 3 1/2 | * | 2 3/8 | 3/4 | 16 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 17 |
| 6.2 | N | 3 1/2 | * | 2 3/8 | ...... | 15 | N | 3 1/2 | * | 2 3/8 | 3/4 | 17 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 18 |
| 6.4 | N | 3 1/2 | * | 2 3/8 | ...... | 16 | N | 3 1/2 | * | 2 3/8 | 3/4 | 18 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 19 |
| 6.6 | N | 3 1/2 | * | 2 3/8 | ...... | 17 | N | 3 1/2 | * | 2 3/8 | 3/4 | 19 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 20 |
| 6.8 | N | 3 1/2 | * | 2 3/8 | ...... | 18 | N | 3 1/2 | * | 2 3/8 | 3/4 | 20 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 21 |
| 7.0 | N | 3 1/2 | * | 2 3/8 | ...... | 19 | N | 3 1/2 | * | 2 3/8 | 3/4 | 21 | N | 3 1/2 | * | 2 3/8 | 1 1/2 | 22 |
| 8.6 | R | 4 1/8 | 5 7/8 | 2 3/8 | ...... | 22 | R | 4 1/8 | 5 7/8 | 2 3/8 | 3/4 | 25 | R | 4 1/8 | 5 7/8 | 2 3/8 | 1 1/2 | 30 |
| 11.0 | R | 4 1/8 | 5 7/8 | 2 3/8 | ...... | 30 | R | 4 1/8 | 5 7/8 | 2 3/8 | 3/4 | 35 | R | 4 1/8 | 5 7/8 | 2 3/8 | 1 1/2 | 40 |
| 13.6 | R | 4 1/8 | 5 7/8 | 2 3/8 | ...... | 35 | R | 4 1/8 | 5 7/8 | 2 3/8 | 3/4 | 40 | R | 4 1/8 | 5 7/8 | 2 3/8 | 1 1/2 | 50 |
| 16.0 | R | 4 1/8 | 5 7/8 | 2 3/8 | ...... | 45 | R | 4 1/8 | 5 7/8 | 2 3/8 | 3/4 | 50 | R | 4 1/8 | 5 7/8 | 2 3/8 | 1 1/2 | 55 |
| 20.0 | R | 4 1/8 | 5 7/8 | 2 3/8 | ...... | 55 | R | 4 1/8 | 5 7/8 | 2 3/8 | 1/16 | 60 | R | 4 1/8 | 5 7/8 | 2 3/8 | 7/16 | 75 |
| 25.0 | S | 4 3/4 | 6 1/2 | 3 | ...... | 75 | S | 4 3/4 | 6 1/2 | 3 | ...... | 85 | S | 4 3/4 | 6 1/2 | 3 | 1/8 | 100 |
| 30.0 | S | 4 3/4 | 6 1/2 | 3 | ...... | 90 | S | 4 3/4 | 6 1/2 | 3 | ...... | 100 | S | 4 3/4 | 6 1/2 | 3 | 1/8 | 115 |
| 38.0 | S | 4 3/4 | 6 1/2 | 3 | ...... | 120 | S | 4 3/4 | 6 1/2 | 3 | ...... | 140 | S | 4 3/4 | 6 1/2 | 3 | 1/8 | 155 |
| | 5-Groove. U = 4-in. Face | | | | | | 6-Groove. U = 4 3/4-in. Face | | | | | | 7-Groove. U = 5 1/2-in. Face | | | | | |
| 5.4 | P | 3 1/2 | * | 3 | 1 5/8 | 17 | P | 3 1/2 | * | 3 | 2 3/8 | 19 | P | 3 1/2 | * | 3 | 3 1/8 | 21 |
| 5.6 | P | 3 1/2 | * | 3 | 1 5/8 | 18 | P | 3 1/2 | * | 3 | 2 3/8 | 20 | P | 3 1/2 | * | 3 | 3 1/8 | 22 |
| 5.8 | P | 3 1/2 | * | 3 | 1 5/8 | 19 | P | 3 1/2 | * | 3 | 2 3/8 | 22 | P | 3 1/2 | * | 3 | 3 1/8 | 23 |
| 6.0 | P | 3 1/2 | * | 3 | 1 5/8 | 21 | P | 3 1/2 | * | 3 | 2 3/8 | 23 | P | 3 1/2 | * | 3 | 3 1/8 | 24 |
| 6.2 | P | 3 1/2 | * | 3 | 1 5/8 | 22 | P | 3 1/2 | * | 3 | 2 3/8 | 24 | P | 3 1/2 | * | 3 | 3 1/8 | 26 |
| 6.4 | P | 3 1/2 | * | 3 | 1 5/8 | 23 | P | 3 1/2 | * | 3 | 2 3/8 | 25 | P | 3 1/2 | * | 3 | 3 1/8 | 27 |
| 6.6 | P | 3 1/2 | * | 3 | 1 5/8 | 25 | P | 3 1/2 | * | 3 | 2 3/8 | 27 | P | 3 1/2 | * | 3 | 3 1/8 | 28 |
| 6.8 | P | 3 1/2 | * | 3 | 1 5/8 | 27 | P | 3 1/2 | * | 3 | 2 3/8 | 29 | P | 3 1/2 | * | 3 | 3 1/8 | 30 |
| 7.0 | P | 3 1/2 | * | 3 | 1 5/8 | 28 | P | 3 1/2 | * | 3 | 2 3/8 | 30 | P | 3 1/2 | * | 3 | 3 1/8 | 32 |
| 8.6 | S | 4 3/4 | 3 1/2 | 3 | 1 5/8 | 35 | S | 4 3/4 | 5 1/2 | 3 | 2 3/8 | 40 | S | 4 3/4 | 6 1/2 | 3 | 3 1/8 | 45 |
| 11.0 | S | 4 3/4 | 6 1/2 | 3 | 1 5/8 | 45 | S | 4 3/4 | 6 1/2 | 3 | 2 3/8 | 55 | S | 4 3/4 | 6 1/2 | 3 | 3 1/8 | 60 |
| 13.6 | S | 4 3/4 | 6 1/2 | 3 | 1 5/8 | 65 | S | 4 3/4 | 6 1/2 | 3 | 2 3/8 | 75 | S | 4 3/4 | 6 1/2 | 3 | 3 1/8 | 80 |
| 16.0 | S | 4 3/4 | 6 1/2 | 3 | 1 5/8 | 75 | S | 4 3/4 | 6 1/2 | 3 | 2 3/8 | 85 | S | 4 3/4 | 6 1/2 | 3 | 3 1/8 | 100 |
| 20.0 | S | 4 3/4 | 6 1/2 | 3 | 1/2 | 90 | S | 4 3/4 | 6 1/2 | 3 | 7/8 | 100 | S | 4 3/4 | 6 1/2 | 3 | 1 1/4 | 110 |
| 25.0 | T | 5 3/8 | 7 1/8 | 4 | ...... | 135 | T | 5 3/8 | 7 1/8 | 4 | 3/8 | 145 | T | 5 3/8 | 7 1/8 | 4 | 3/4 | 155 |
| 30.0 | T | 5 3/8 | 7 1/8 | 4 | ...... | 155 | T | 5 3/8 | 7 1/8 | 4 | 3/8 | 170 | T | 5 3/8 | 7 1/8 | 4 | 3/4 | 180 |
| 38.0 | T | 5 3/8 | 7 1/8 | 4 | ...... | 210 | T | 5 3/8 | 7 1/8 | 4 | 3/8 | 230 | T | 5 3/8 | 7 1/8 | 4 | 3/4 | 250 |
| | 8-Groove. U = 6 1/4-in. Face | | | | | | 9-Groove. U = 7-in. Face | | | | | | 10-Groove. U = 7 3/4-in. Face | | | | | |
| 5.4 | Q | 3 1/2 | * | 4 | 2 7/8 | 24 | Q | 3 1/2 | * | 4 | 3 5/8 | 26 | Q | 3 1/2 | * | 4 | 4 3/8 | 27 |
| 5.6 | Q | 3 1/2 | * | 4 | 2 7/8 | 25 | Q | 3 1/2 | * | 4 | 3 5/8 | 27 | Q | 3 1/2 | * | 4 | 4 3/8 | 28 |
| 5.8 | Q | 3 1/2 | * | 4 | 2 7/8 | 27 | Q | 3 1/2 | * | 4 | 3 5/8 | 28 | Q | 3 1/2 | * | 4 | 4 3/8 | 29 |
| 6.0 | Q | 3 1/2 | * | 4 | 2 7/8 | 29 | Q | 3 1/2 | * | 4 | 3 5/8 | 30 | Q | 3 1/2 | * | 4 | 4 3/8 | 31 |
| 6.2 | Q | 3 1/2 | * | 4 | 2 7/8 | 30 | Q | 3 1/2 | * | 4 | 3 5/8 | 31 | Q | 3 1/2 | * | 4 | 4 3/8 | 32 |
| 6.4 | Q | 3 1/2 | * | 4 | 2 7/8 | 31 | Q | 3 1/2 | * | 4 | 3 5/8 | 32 | Q | 3 1/2 | * | 4 | 4 3/8 | 33 |
| 6.6 | Q | 3 1/2 | * | 4 | 2 7/8 | 33 | Q | 3 1/2 | * | 4 | 3 5/8 | 34 | Q | 3 1/2 | * | 4 | 4 3/8 | 35 |
| 6.8 | Q | 3 1/2 | * | 4 | 2 7/8 | 34 | Q | 3 1/2 | * | 4 | 3 5/8 | 36 | Q | 3 1/2 | * | 4 | 4 3/8 | 37 |
| 7.0 | Q | 3 1/2 | * | 4 | 2 7/8 | 35 | Q | 3 1/2 | * | 4 | 3 5/8 | 38 | Q | 3 1/2 | * | 4 | 4 3/8 | 39 |
| 8.6 | T | 5 3/8 | 7 1/8 | 4 | 2 7/8 | 50 | T | 5 3/8 | 7 1/8 | 4 | 3 5/8 | 55 | T | 5 3/8 | 7 1/8 | 4 | 4 3/8 | 60 |
| 11.0 | T | 5 3/8 | 7 1/8 | 4 | 2 7/8 | 65 | T | 5 3/8 | 7 1/8 | 4 | 3 5/8 | 70 | T | 5 3/8 | 7 1/8 | 4 | 4 3/8 | 75 |
| 13.6 | T | 5 3/8 | 7 1/8 | 4 | 2 7/8 | 90 | T | 5 3/8 | 7 1/8 | 4 | 3 5/8 | 95 | T | 5 3/8 | 7 1/8 | 4 | 4 3/8 | 100 |
| 16.0 | T | 5 3/8 | 7 1/8 | 4 | 2 7/8 | 110 | T | 5 3/8 | 7 1/8 | 4 | 3 5/8 | 120 | T | 5 3/8 | 7 1/8 | 4 | 4 3/8 | 125 |
| 20.0 | T | 5 3/8 | 7 1/8 | 4 | 1 1/8 | 135 | T | 5 3/8 | 7 1/8 | 4 | 1 1/2 | 145 | T | 5 3/8 | 7 1/8 | 4 | 1 7/8 | 150 |
| 25.0 | T | 5 3/8 | 7 1/8 | 4 | 1 1/8 | 165 | T | 5 3/8 | 7 1/8 | 4 | 1 1/2 | 175 | T | 5 3/8 | 7 1/8 | 4 | 1 7/8 | 185 |
| 30.0 | T | 5 3/8 | 7 1/8 | 4 | 1 1/8 | 195 | T | 5 3/8 | 7 1/8 | 4 | 1 1/2 | 205 | T | 5 3/8 | 7 1/8 | 4 | 1 7/8 | 220 |
| 38.0 | T | 5 3/8 | 7 1/8 | 4 | 1 1/8 | 265 | T | 5 3/8 | 7 1/8 | 4 | 1 1/2 | 280 | T | 5 3/8 | 7 1/8 | 4 | 1 7/8 | 295 |

* No hub.

| † Bushing Class | N | P | Q | R | S | T |
|---|---|---|---|---|---|---|
| Smallest Bore, in. | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 |
| Largest Bore, in. | 2 1/2 | 2 1/2 | 2 1/2 | 3 | 3 1/2 | 4 |

are a speed-up factor of 1.2 and a reversing service and line-start motor factor of 1.2. Both these factors are applied in addition to the overload factor if the conditions of the drive require them. A service overload factor, depending on the application of the drive, may be applied instead of the overload factor given above. Service factors are given in Table 10.

Texropes are made in the following sizes of cross-section, the first figure denoting the maximum width and the second the thickness: $1/2 \times 11/32$ in., $21/32 \times 7/16$ in., $7/8 \times 17/32$ in., $1\ 1/4 \times 3/4$ in., $1\ 1/2 \times 1$ in. Selection of the correct drive is made from voluminous tables issued by the manufacturers of V-belts, Tables 11 and 12 being typical of those issued for drives of from 2 to 25 Hp., using cast-iron sheaves. Table 11 gives center distances that are available at various speed ratios of driving and driven sheaves with Texrope drives of the class noted, and also the serial number of the belt that can be used. At the top of each column of center distances the letters A, B, C, and D, refer the user to corresponding columns in Table 12, in which will be found the horsepower that can be trans-

**Table 10.—Service Factors for Texrope Drives**

| Type of Machinery | Service Overload Factors | Type of Machinery | Service Overload Factors |
|---|---|---|---|
| Brick and clay | 1.2 to 1.3 | Oil field | 1.2 to 1.5 |
| Coal mining | 1.0 to 1.4 | Pulp and paper | 1.2 to 1.4 |
| Cotton ginning | 1.0 to 1.2 | Printing | 1.0 to 1.3 |
| Fans and blowers | 1.2 to 1.4 | Pumps and compressors | 1.2 to 1.3 |
| Flour and feed mill | 1.2 to 1.3 | Rock crushing | 1.2 to 1.4 |
| Laundry | 1.1 to 1.2 | Textile | 1.2 to 1.5 |
| Machine tools | 1.0 to 1.3 | Woodworking | 1.2 to 1.3 |
| Metal mining | 1.0 to 1.4 | | |

**Table 11.—Center Distances and Texrope Serial Numbers**

2 to 25 Horsepower. $21/32 \times 7/16$-in. Texrope. 2 to 10 Grooves

(Allis-Chalmers Mfg. Co., Milwaukee, Wis.)

| Speed Ratio | Center Distance, in. A | Texrope Serial No. | Center Distance, in. B | Texrope Serial No. | Center Distance, in. C | Texrope Serial No. | Center Distance, in. D | Texrope Serial No. | Speed Ratio | Center Distance, in. A | Texrope Serial No. | Center Distance, in. B | Texrope Serial No. | Center Distance, in. C | Texrope Serial No. | Center Distance, in. D | Texrope Serial No. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1.00 | 8.1 | B35 | 11.9 | B42 | 16.1 | B51 | 20.6 | B60 | 2.58 | 12.3 | B60 | 16.3 | B68 | 20.2 | B75 | 27.8 | B90 |
| 1.03 | 8.3 | B35 | 11.8 | B42 | 16.3 | B51 | 20.8 | B60 | 2.67 | 12.4 | B60 | 16.7 | B68 | 20.3 | B75 | 28.0 | B90 |
| 1.07 | 8.4 | B35 | 12.2 | B42 | 16.4 | B51 | 20.9 | B60 | 2.76 | 12.5 | B60 | 16.9 | B68 | 20.5 | B75 | 28.2 | B90 |
| 1.10 | 8.6 | B35 | 12.4 | B42 | 16.6 | B51 | 21.1 | B60 | 2.86 | 12.6 | B60 | 17.0 | B68 | 20.6 | B75 | 28.3 | B90 |
| 1.14 | 8.7 | B35 | 12.5 | B42 | 16.8 | B51 | 21.3 | B60 | 2.96 | 12.8 | B60 | 17.1 | B68 | 20.9 | B75 | 28.5 | B90 |
| 1.16 | 7.9 | B35 | 11.5 | B42 | 16.0 | B51 | 20.4 | B60 | 3.03 | 15.8 | B75 | 23.9 | B90 | 31.6 | B105 | 39.3 | B120 |
| 1.19 | 8.8 | B35 | 12.4 | B42 | 16.9 | B51 | 21.4 | B60 | 3.13 | 16.0 | B75 | 24.0 | B90 | 31.8 | B105 | 39.4 | B120 |
| 1.21 | 8.1 | B35 | 11.6 | B42 | 16.2 | B51 | 20.6 | B60 | 3.23 | 16.1 | B75 | 24.1 | B90 | 32.0 | B105 | 39.5 | B120 |
| 1.23 | 9.2 | B42 | 13.9 | B51 | 18.4 | B60 | 22.4 | B68 | 3.33 | 16.2 | B75 | 24.3 | B90 | 32.1 | B105 | 39.7 | B120 |
| 1.26 | 9.6 | B42 | 14.1 | B51 | 18.6 | B60 | 22.6 | B68 | 3.45 | 16.3 | B75 | 24.4 | B90 | 32.2 | B105 | 39.8 | B120 |
| 1.30 | 9.8 | B42 | 14.2 | B51 | 18.7 | B60 | 22.8 | B68 | 3.58 | 16.4 | B75 | 24.5 | B90 | 32.4 | B105 | 39.9 | B120 |
| 1.34 | 9.9 | B42 | 14.4 | B51 | 18.9 | B60 | 22.9 | B68 | 3.71 | 16.5 | B75 | 24.8 | B90 | 32.6 | B105 | 40.0 | B120 |
| 1.39 | 10.0 | B42 | 14.6 | B51 | 19.0 | B60 | 23.0 | B68 | 3.79 | 18.5 | B90 | 26.8 | B105 | 34.7 | B120 | 46.9 | B144 |
| 1.43 | 10.1 | B42 | 14.7 | B51 | 19.2 | B60 | 23.2 | B68 | 3.91 | 18.7 | B90 | 27.0 | B105 | 34.8 | B120 | 47.2 | B144 |
| 1.48 | 10.3 | B42 | 14.8 | B51 | 19.3 | B60 | 23.4 | B68 | 4.03 | 18.8 | B90 | 27.1 | B105 | 34.9 | B120 | 47.4 | B144 |
| 1.54 | 10.5 | B42 | 14.9 | B51 | 19.5 | B60 | 23.5 | B68 | 4.17 | 18.9 | B90 | 27.2 | B105 | 35.2 | B120 | 47.5 | B144 |
| 1.59 | 10.7 | B42 | 15.2 | B51 | 19.7 | B60 | 23.6 | B68 | 4.32 | 19.0 | B90 | 27.4 | B105 | 35.3 | B120 | 47.6 | B144 |
| 1.62 | 12.0 | B51 | 16.6 | B60 | 20.6 | B68 | 24.1 | B75 | 4.47 | 19.1 | B90 | 27.6 | B105 | 35.4 | B120 | 47.8 | B144 |
| 1.67 | 12.1 | B51 | 16.7 | B60 | 20.7 | B68 | 24.2 | B75 | 4.63 | 19.3 | B90 | 27.8 | B105 | 35.5 | B120 | 47.9 | B144 |
| 1.72 | 12.3 | B51 | 16.9 | B60 | 20.9 | B68 | 24.4 | B75 | 4.69 | 21.5 | B105 | 29.8 | B120 | 42.6 | B144 | 49.8 | B158 |
| 1.77 | 12.5 | B51 | 17.1 | B60 | 21.1 | B68 | 24.6 | B75 | 4.84 | 21.6 | B105 | 29.9 | B120 | 42.7 | B144 | 50.0 | B158 |
| 1.83 | 12.6 | B51 | 17.2 | B60 | 21.2 | B68 | 24.7 | B75 | 5.00 | 21.7 | B105 | 30.2 | B120 | 42.8 | B144 | 50.2 | B158 |
| 1.90 | 12.7 | B51 | 17.3 | B60 | 21.3 | B68 | 24.8 | B75 | 5.17 | 21.9 | B105 | 30.3 | B120 | 43.2 | B144 | 50.3 | B158 |
| 1.96 | 12.8 | B51 | 17.4 | B60 | 21.4 | B68 | 25.0 | B75 | 5.37 | 22.0 | B105 | 30.4 | B120 | 43.3 | B144 | 50.4 | B158 |
| 2.04 | 12.9 | B51 | 17.6 | B60 | 21.6 | B68 | 25.1 | B75 | 5.56 | 22.2 | B105 | 30.5 | B120 | 43.4 | B144 | 50.5 | B158 |
| 2.06 | 14.4 | B60 | 18.5 | B68 | 22.1 | B75 | 29.6 | B90 | 5.76 | 33.6 | B144 | 41.3 | B158 | 49.9 | B173 | 53.1 | B180 |
| 2.13 | 14.5 | B60 | 18.6 | B68 | 22.2 | B75 | 29.8 | B90 | 5.93 | 34.0 | B144 | 41.5 | B158 | 50.0 | B173 | 53.3 | B180 |
| 2.20 | 14.6 | B60 | 18.7 | B68 | 22.3 | B75 | 29.9 | B90 | 6.13 | 34.3 | B144 | 41.7 | B158 | 50.1 | B173 | 53.5 | B180 |
| 2.27 | 14.7 | B60 | 18.9 | B68 | 22.4 | B75 | 30.0 | B90 | 6.33 | 34.5 | B144 | 41.8 | B158 | 50.2 | B173 | 53.6 | B180 |
| 2.35 | 14.8 | B60 | 19.1 | B68 | 22.6 | B75 | 30.1 | B90 | 6.55 | 34.7 | B144 | 42.0 | B158 | 50.3 | B173 | 53.7 | B180 |
| 2.43 | 10.3 | B51 | 15.1 | B60 | 19.1 | B68 | 22.8 | B75 | 6.78 | 34.9 | B144 | 42.2 | B158 | 50.4 | B173 | 53.9 | B180 |
| 2.52 | 10.4 | B51 | 15.2 | B60 | 19.3 | B68 | 22.9 | B75 | 7.04 | 35.0 | B144 | 42.4 | B158 | 50.8 | B173 | 54.0 | B180 |

mitted per single Texrope at the given motor speed and selected center line distance. This figure for horsepower is divided into the horsepower required for the drive, the quotient being the number of Texropes that must be used. Since a fraction of a rope cannot be used the next higher integer must be taken when the quotient is a fraction or a mixed

**Table 12.—Horsepower of Texrope Drives with Cast-iron Sheaves**

2 to 25 Horsepower. 21/32 × 7/16-in. Texrope. 2 to 10 Grooves

(Allis-Chalmers Mfg. Co., Milwaukee, Wis.)

| Sheave Diameters, in. | | Speed Ratio | Motor Speed, 1730 r.p.m. | | | | | Motor Speed, 1140 r.p.m. | | | | | Motor Speed, 850 r.p.m. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. of Driven Shaft | Center Distance See Table 11 | | | | R.p.m. of Driven Shaft | Center Distance See Table 11 | | | | R.p.m. of Driven Shaft | Center Distance See Table 11 | | | |
| DriveR | DriveN | | | A | B | C | D | | A | B | C | D | | A | B | C | D |
| | | | Horsepower per Texrope | | | | | | | | | | | | | | |
| 6.4 | 6.4 | 1.00 | 1730 | 3.13 | 3.13 | 3.13 | 3.13 | 1140 | 2.18 | 2.18 | 2.18 | 2.18 | 850 | 1.62 | 1.62 | 1.62 | 1.62 |
| 6.2 | 6.4 | 1.03 | 1677 | 3.05 | 3.05 | 3.05 | 3.05 | 1104 | 2.11 | 2.11 | 2.11 | 2.11 | 823 | 1.58 | 1.58 | 1.58 | 1.58 |
| 6.0 | 6.4 | 1.07 | 1622 | 2.97 | 2.97 | 2.97 | 2.97 | 1069 | 2.03 | 2.03 | 2.03 | 2.03 | 796 | 1.53 | 1.53 | 1.53 | 1.53 |
| 5.8 | 6.4 | 1.10 | 1568 | 2.86 | 2.86 | 2.86 | 2.86 | 1033 | 1.93 | 1.93 | 1.93 | 1.93 | 770 | 1.48 | 1.48 | 1.48 | 1.48 |
| 5.6 | 6.4 | 1.14 | 1514 | 2.79 | 2.79 | 2.79 | 2.79 | 997 | 1.86 | 1.86 | 1.86 | 1.86 | 744 | 1.43 | 1.43 | 1.43 | 1.43 |
| 6.0 | 7.0 | 1.16 | 1483 | 2.91 | 2.94 | 2.94 | 2.94 | 977 | 1.99 | 2.01 | 2.01 | 2.01 | 729 | 1.50 | 1.52 | 1.52 | 1.52 |
| 5.4 | 6.4 | 1.19 | 1460 | 2.70 | 2.72 | 2.72 | 2.72 | 962 | 1.77 | 1.79 | 1.79 | 1.79 | 717 | 1.37 | 1.39 | 1.39 | 1.39 |
| 5.8 | 7.0 | 1.21 | 1433 | 2.83 | 2.86 | 2.86 | 2.86 | 944 | 1.91 | 1.93 | 1.93 | 1.93 | 704 | 1.46 | 1.48 | 1.48 | 1.48 |
| 7.0 | 8.6 | 1.23 | 1408 | 3.30 | 3.34 | 3.34 | 3.34 | 928 | 2.34 | 2.37 | 2.37 | 2.37 | 692 | 1.73 | 1.74 | 1.74 | 1.74 |
| 6.8 | 8.6 | 1.26 | 1368 | 3.22 | 3.22 | 3.26 | 3.26 | 902 | 2.28 | 2.28 | 2.30 | 2.30 | 672 | 1.68 | 1.68 | 1.69 | 1.69 |
| 6.6 | 8.6 | 1.30 | 1328 | 3.12 | 3.15 | 3.18 | 3.18 | 875 | 2.18 | 2.21 | 2.23 | 2.23 | 652 | 1.62 | 1.64 | 1.65 | 1.65 |
| 6.4 | 8.6 | 1.34 | 1288 | 3.04 | 3.07 | 3.10 | 3.10 | 848 | 2.12 | 2.14 | 2.16 | 2.16 | 633 | 1.57 | 1.59 | 1.60 | 1.60 |
| 6.2 | 8.6 | 1.39 | 1247 | 2.96 | 2.99 | 2.99 | 3.02 | 822 | 2.05 | 2.07 | 2.07 | 2.09 | 613 | 1.53 | 1.55 | 1.55 | 1.57 |
| 6.0 | 8.6 | 1.43 | 1207 | 2.85 | 2.91 | 2.91 | 2.94 | 795 | 1.95 | 1.99 | 1.99 | 2.01 | 593 | 1.47 | 1.50 | 1.50 | 1.52 |
| 5.8 | 8.6 | 1.48 | 1168 | 2.78 | 2.80 | 2.84 | 2.86 | 768 | 1.87 | 1.89 | 1.91 | 1.93 | 573 | 1.43 | 1.45 | 1.46 | 1.48 |
| 5.6 | 8.6 | 1.54 | 1127 | 2.71 | 2.74 | 2.76 | 2.76 | 742 | 1.81 | 1.82 | 1.85 | 1.85 | 553 | 1.38 | 1.40 | 1.41 | 1.41 |
| 5.4 | 8.6 | 1.59 | 1086 | 2.62 | 2.67 | 2.70 | 2.70 | 716 | 1.72 | 1.76 | 1.77 | 1.77 | 533 | 1.33 | 1.36 | 1.37 | 1.37 |
| 6.8 | 11.0 | 1.62 | 1069 | 3.13 | 3.16 | 3.19 | 3.23 | 704 | 2.20 | 2.23 | 2.25 | 2.28 | 525 | 1.63 | 1.64 | 1.66 | 1.68 |
| 6.6 | 11.0 | 1.67 | 1038 | 3.05 | 3.08 | 3.12 | 3.15 | 683 | 2.14 | 2.16 | 2.18 | 2.21 | 510 | 1.59 | 1.60 | 1.62 | 1.64 |
| 6.4 | 11.0 | 1.72 | 1007 | 2.94 | 3.01 | 3.04 | 3.07 | 663 | 2.05 | 2.10 | 2.12 | 2.14 | 494 | 1.52 | 1.56 | 1.57 | 1.59 |
| 6.2 | 11.0 | 1.77 | 976 | 2.87 | 2.93 | 2.96 | 2.96 | 642 | 1.98 | 2.03 | 2.05 | 2.05 | 478 | 1.49 | 1.52 | 1.53 | 1.53 |
| 6.0 | 11.0 | 1.83 | 943 | 2.80 | 2.85 | 2.88 | 2.88 | 621 | 1.91 | 1.95 | 1.97 | 1.97 | 463 | 1.44 | 1.47 | 1.49 | 1.49 |
| 5.8 | 11.0 | 1.90 | 912 | 2.72 | 2.75 | 2.80 | 2.80 | 601 | 1.83 | 1.85 | 1.89 | 1.89 | 448 | 1.40 | 1.42 | 1.45 | 1.49 |
| 5.6 | 11.0 | 1.96 | 881 | 2.62 | 2.68 | 2.71 | 2.74 | 580 | 1.75 | 1.79 | 1.81 | 1.82 | 433 | 1.34 | 1.37 | 1.38 | 1.40 |
| 5.4 | 11.0 | 2.04 | 849 | 2.56 | 2.61 | 2.64 | 2.67 | 560 | 1.68 | 1.72 | 1.74 | 1.76 | 417 | 1.30 | 1.33 | 1.34 | 1.36 |
| 6.6 | 13.8 | 2.06 | 840 | 2.95 | 3.02 | 3.05 | 3.12 | 553 | 2.07 | 2.12 | 2.14 | 2.18 | 412 | 1.54 | 1.57 | 1.59 | 1.62 |
| 6.4 | 13.6 | 2.13 | 814 | 2.88 | 2.94 | 2.98 | 3.04 | 536 | 2.01 | 2.05 | 2.07 | 2.12 | 400 | 1.49 | 1.52 | 1.54 | 1.56 |
| 6.2 | 13.6 | 2.20 | 788 | 2.81 | 2.87 | 2.90 | 2.93 | 519 | 1.94 | 1.98 | 2.01 | 2.03 | 387 | 1.45 | 1.49 | 1.50 | 1.52 |
| 6.0 | 13.6 | 2.27 | 763 | 2.73 | 2.80 | 2.82 | 2.85 | 503 | 1.87 | 1.91 | 1.93 | 1.95 | 375 | 1.41 | 1.44 | 1.45 | 1.47 |
| 5.8 | 13.6 | 2.35 | 738 | 2.66 | 2.72 | 2.75 | 2.78 | 486 | 1.80 | 1.83 | 1.85 | 1.87 | 362 | 1.37 | 1.40 | 1.42 | 1.43 |
| 5.6 | 13.6 | 2.43 | 713 | 2.46 | 2.60 | 2.65 | 2.68 | 469 | 1.64 | 1.73 | 1.77 | 1.79 | 350 | 1.25 | 1.33 | 1.35 | 1.37 |
| 5.4 | 13.6 | 2.52 | 687 | 2.39 | 2.53 | 2.56 | 2.62 | 453 | 1.57 | 1.67 | 1.68 | 1.72 | 337 | 1.22 | 1.29 | 1.30 | 1.33 |
| 6.2 | 16.0 | 2.58 | 671 | 2.66 | 2.75 | 2.81 | 2.90 | 442 | 1.84 | 1.90 | 1.94 | 2.01 | 329 | 1.38 | 1.42 | 1.45 | 1.50 |
| 6.0 | 16.0 | 2.67 | 649 | 2.59 | 2.68 | 2.73 | 2.82 | 427 | 1.77 | 1.83 | 1.87 | 1.93 | 318 | 1.33 | 1.38 | 1.41 | 1.45 |
| 5.8 | 16.0 | 2.76 | 627 | 2.52 | 2.61 | 2.66 | 2.75 | 413 | 1.70 | 1.76 | 1.80 | 1.85 | 308 | 1.30 | 1.34 | 1.37 | 1.42 |
| 5.6 | 16.0 | 2.86 | 606 | 2.43 | 2.54 | 2.60 | 2.68 | 399 | 1.62 | 1.69 | 1.73 | 1.79 | 297 | 1.24 | 1.30 | 1.33 | 1.37 |
| 5.4 | 16.0 | 2.96 | 584 | 2.37 | 2.48 | 2.53 | 2.59 | 384 | 1.56 | 1.63 | 1.67 | 1.70 | 287 | 1.20 | 1.26 | 1.29 | 1.32 |
| 6.6 | 20.0 | 3.03 | 571 | 2.76 | 2.92 | 2.99 | 3.05 | 376 | 1.94 | 2.05 | 2.09 | 2.14 | 280 | 1.44 | 1.52 | 1.55 | 1.59 |
| 6.4 | 20.0 | 3.13 | 554 | 2.69 | 2.85 | 2.91 | 2.98 | 364 | 1.88 | 1.98 | 2.03 | 2.07 | 272 | 1.39 | 1.48 | 1.51 | 1.54 |
| 6.2 | 20.0 | 3.23 | 536 | 2.62 | 2.78 | 2.84 | 2.90 | 353 | 1.82 | 1.92 | 1.96 | 2.01 | 263 | 1.36 | 1.44 | 1.47 | 1.50 |
| 6.0 | 20.0 | 3.33 | 519 | 2.56 | 2.70 | 2.76 | 2.82 | 342 | 1.75 | 1.85 | 1.89 | 1.93 | 255 | 1.32 | 1.39 | 1.42 | 1.45 |
| 5.8 | 20.0 | 3.45 | 502 | 2.49 | 2.63 | 2.69 | 2.75 | 330 | 1.68 | 1.78 | 1.81 | 1.85 | 246 | 1.28 | 1.36 | 1.39 | 1.42 |
| 5.6 | 20.0 | 3.58 | 484 | 2.43 | 2.57 | 2.62 | 2.68 | 319 | 1.62 | 1.71 | 1.75 | 1.79 | 238 | 1.24 | 1.31 | 1.34 | 1.37 |
| 5.4 | 20.0 | 3.71 | 467 | 2.37 | 2.50 | 2.56 | 2.62 | 307 | 1.56 | 1.65 | 1.68 | 1.72 | 229 | 1.20 | 1.28 | 1.30 | 1.33 |
| 6.6 | 25.0 | 3.79 | 457 | 2.70 | 2.86 | 2.95 | 3.02 | 301 | 1.89 | 2.00 | 2.07 | 2.12 | 224 | 1.40 | 1.49 | 1.54 | 1.57 |
| 6.4 | 25.0 | 3.91 | 443 | 2.63 | 2.97 | 2.88 | 2.94 | 292 | 1.83 | 1.94 | 2.00 | 2.05 | 217 | 1.36 | 1.44 | 1.49 | 1.52 |
| 6.2 | 25.0 | 4.03 | 429 | 2.56 | 2.72 | 2.81 | 2.87 | 282 | 1.77 | 1.88 | 1.94 | 1.98 | 210 | 1.33 | 1.41 | 1.45 | 1.49 |
| 6.0 | 25.0 | 4.17 | 415 | 2.47 | 2.65 | 2.73 | 2.80 | 273 | 1.69 | 1.81 | 1.87 | 1.91 | 204 | 1.27 | 1.36 | 1.41 | 1.44 |
| 5.8 | 25.0 | 4.32 | 401 | 2.40 | 2.58 | 2.66 | 2.72 | 264 | 1.62 | 1.74 | 1.80 | 1.84 | 197 | 1.24 | 1.33 | 1.37 | 1.40 |
| 5.6 | 25.0 | 4.47 | 387 | 2.34 | 2.51 | 2.57 | 2.65 | 255 | 1.56 | 1.67 | 1.71 | 1.77 | 190 | 1.20 | 1.28 | 1.31 | 1.36 |
| 5.4 | 25.0 | 4.63 | 373 | 2.28 | 2.45 | 2.50 | 2.59 | 246 | 1.50 | 1.61 | 1.65 | 1.70 | 183 | 1.16 | 1.25 | 1.28 | 1.32 |
| 6.4 | 30.0 | 4.69 | 369 | 2.57 | 2.72 | 2.85 | 2.92 | 243 | 1.79 | 1.90 | 1.99 | 2.03 | 181 | 1.33 | 1.41 | 1.48 | 1.51 |
| 6.2 | 30.0 | 4.84 | 358 | 2.50 | 2.65 | 2.78 | 2.84 | 235 | 1.73 | 1.84 | 1.92 | 1.97 | 175 | 1.30 | 1.38 | 1.44 | 1.47 |
| 6.0 | 30.0 | 5.00 | 346 | 2.44 | 2.58 | 2.70 | 2.76 | 228 | 1.67 | 1.77 | 1.85 | 1.89 | 170 | 1.26 | 1.33 | 1.39 | 1.42 |
| 5.8 | 30.0 | 5.17 | 334 | 2.37 | 2.51 | 2.63 | 2.69 | 220 | 1.60 | 1.70 | 1.78 | 1.82 | 164 | 1.22 | 1.30 | 1.36 | 1.39 |
| 5.6 | 30.0 | 5.37 | 323 | 2.31 | 2.47 | 2.57 | 2.62 | 212 | 1.54 | 1.64 | 1.71 | 1.75 | 158 | 1.18 | 1.25 | 1.31 | 1.34 |
| 5.4 | 30.0 | 5.56 | 311 | 2.26 | 2.39 | 2.50 | 2.56 | 205 | 1.48 | 1.58 | 1.65 | 1.68 | 153 | 1.15 | 1.22 | 1.28 | 1.30 |
| 6.6 | 38.0 | 5.76 | 300 | 2.73 | 2.83 | 2.89 | 2.92 | 198 | 1.91 | 1.98 | 2.03 | 2.05 | 147 | 1.42 | 1.47 | 1.50 | 1.52 |
| 6.4 | 38.0 | 5.93 | 291 | 2.66 | 2.76 | 2.82 | 2.85 | 192 | 1.85 | 1.92 | 1.96 | 1.98 | 143 | 1.38 | 1.43 | 1.46 | 1.48 |
| 6.2 | 38.0 | 6.13 | 282 | 2.60 | 2.69 | 2.75 | 2.78 | 186 | 1.79 | 1.86 | 1.90 | 1.92 | 138 | 1.34 | 1.39 | 1.42 | 1.44 |
| 6.0 | 38.0 | 6.33 | 273 | 2.53 | 2.62 | 2.68 | 2.68 | 180 | 1.73 | 1.79 | 1.83 | 1.83 | 134 | 1.30 | 1.35 | 1.38 | 1.38 |
| 5.8 | 38.0 | 6.55 | 264 | 2.46 | 2.54 | 2.60 | 2.60 | 174 | 1.66 | 1.72 | 1.76 | 1.76 | 130 | 1.27 | 1.31 | 1.34 | 1.34 |
| 5.6 | 38.0 | 6.78 | 255 | 2.40 | 2.48 | 2.54 | 2.54 | 168 | 1.60 | 1.66 | 1.69 | 1.69 | 125 | 1.22 | 1.27 | 1.30 | 1.30 |
| 5.4 | 38.0 | 7.04 | 246 | 2.34 | 2.42 | 2.48 | 2.48 | 162 | 1.54 | 1.59 | 1.63 | 1.63 | 121 | 1.19 | 1.23 | 1.26 | 1.26 |

number. It should be noted that the horsepower required for the drive is the maximum horsepower after all overload factors have been applied.

Example.—A fan operating continuously at 400 r.p.m. is to be driven by a 10-Hp. 1140-r.p.m. motor. Center distance of fan and motor shafts, approximately 28 in. Required the drive for this installation. *Solution.*—From Table 10, the overload factor for fans is 1.2 to 1.4. For continuous duty the higher value would be used. Horsepower of drive, therefore, is 1.4 × 10 = 14 Hp. Speed ratio = 1140/400 = 2.85. In Table 11, opposite speed ratio 2.86, find in column D a center distance of 28.3 in., which may be assumed as satisfactory. In Table 12, column D under motor speed of 1140 r.p.m., and opposite speed ratio of 2.86 find the horsepower per No. B90 Texrope to be 2.68. Number of texropes required = 14/2.68 = 5.22. Hence, the drive will consist of six No. B90 Texropes. From Table 12, sheave pitch diameters will be 5.6 and 16 in. respectively.

**PIVOTED MOTOR (ROCKWOOD) DRIVE.**—An efficient short-center drive, generally known as the Rockwood drive, uses a flat belt. The driving motor is mounted on a pair of overhanging arms that are free to rotate about a shaft in a base which is rigidly attached to the supporting structure. See Fig. 17. Various arrangements are possible to permit of floor, ceiling, wall or machine mounting. See Figs. 18 and 18*a*. The weight of the motor thus is utilized to provide tension in the belt. The arms are adjustable, permitting movement of the motor toward or away from the rocker shaft, and thus enabling any desired initial tension to be obtained in the belt. As the belt stretches in service, the

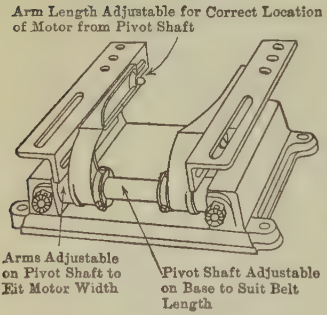

Arm Length Adjustable for Correct Location of Motor from Pivot Shaft
Arms Adjustable on Pivot Shaft to Fit Motor Width
Pivot Shaft Adjustable on Base to Suit Belt Length

Fig. 17. Base for Pivoted Motor Drive

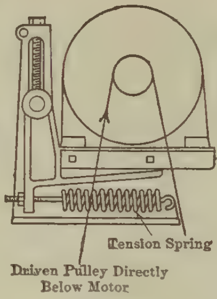


Fig. 18*a*

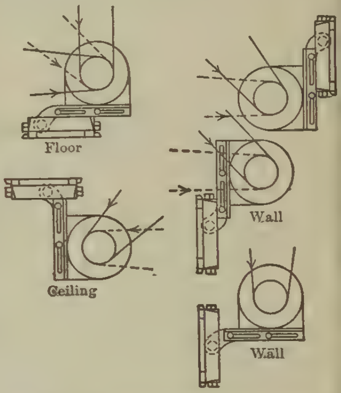
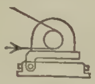
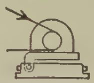


Fig. 18. Arrangements of Pivoted Motor Drives

**Table 13.—Motor Ratings for Rockwood Drives**

(Rockwood Mfg. Co., Indianapolis, Ind.)

| Base No. | Motor Rating, Hp. | | | Direction of Belt | | | |
|---|---|---|---|---|---|---|---|
| | 1800 R.p.m. | 1200 R.p.m. | 900 R.p.m. |  | |  | |
| | | | | Min. Pulley Diam., in. | Width of Belt, in. | Min. Pulley Diam., in. | Width of Belt, in. |
| 0 | 1 | 3/4 | 1/2 | 2 1/2 | 2 1/2 | 3 | 2 |
| | 2 | 1 1/2 | 1 | 3 | 3 | 3 1/2 | 2 1/2 |
| 2 | 3 | 2 | 1 1/2 | 3 1/2 | 3 | 4 | 3 |
| | 5 | 3 | 2 | 4 | 4 | 4 1/2 | 4 |
| 4 | 7 1/2 | 5 | 3 | 4 1/2 | 5 | 5 1/2 | 4 |
| | 10 | 7 1/2 | 5 | 5 | 6 | 6 | 5 |
| 6 | 15 | 10 | 7 1/2 | 6 | 7 | 7 | 6 |
| | 20 | 15 | 10 | 7 | 7 | 8 | 6 |
| 8 | 25 | 20 | 15 | 7 | 8 | 8 | 7 |
| | 30 | 25 | 20 | 8 | 8 | 9 | 7 |
| 10 | 40 | 30 | 25 | 9 | 9 | 10 | 8 |
| | 50 | 40 | 30 | 10 | 10 | 11 | 9 |
| 12 | 60 | 50 | 40 | 11 | 11 | 12 | 10 |
| | 75 | 60 | 50 | 12 | 12 | 13 | 11 |

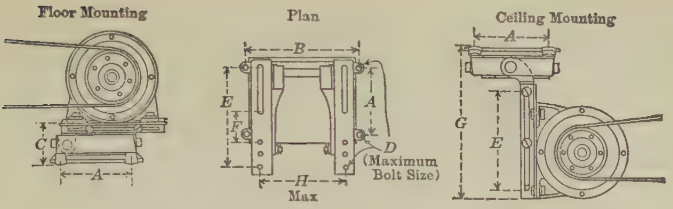


Fig. 19. Dimensions of Standard Rockwood Drives

**Table 14.—Dimensions of Standard Rockwood Drives**
(Rockwood Mfg. Co., Indianapolis, Ind.)

| Base No. | Heaviest Motor Base Should Carry, lb. | | Dimensions, in. | | | | | | | | Shipping Weight, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Floor Mounting | Wall or Ceiling Mounting | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | |
| 0 | 150 | 125 | 7 5/8 | 12 3/8 | 4 1/2 | 1/2 | 11 | 3 1/2 | 16 3/4 | 8 1/2 | 35 |
| 2 | 250 | 200 | 9 3/8 | 15 1/4 | 5 1/2 | 5/8 | 13 | 3 7/8 | 19 1/2 | 10 1/2 | 55 |
| 4 | 450 | 350 | 11 | 17 1/4 | 6 1/4 | 5/8 | 15 | 4 7/8 | 22 5/8 | 13 | 80 |
| 6 | 650 | 500 | 12 3/4 | 19 | 7 | 3/4 | 17 | 5 | 25 3/4 | 14 1/4 | 110 |
| 8 | 900 | 700 | 14 1/4 | 21 | 8 | 3/4 | 19 | 6 1/2 | 28 3/4 | 15 1/4 | 150 |
| 10 | 1500 | 1200 | 16 | 23 | 8 1/2 | 7/8 | 22 | 8 3/8 | 32 1/2 | 18 | 210 |
| 12 | 2500 | 2000 | 19 1/8 | 30 | 10 | 1 | 27 | 12 1/4 | 38 3/4 | 22 | 360 |

weight of the motor automatically takes up the slack developed, and maintains the tension at a constant value. Table 14 and Fig. 19 give the dimensions of standard Rockwood bases and Table 13 gives the size of base that should be used with different sizes of motors under various conditions of driving.

**Theory of the Pivoted Motor Drive.**—R. R. Tatnall (*Mech. Engg.*, May, 1935) outlines the theory of the pivoted motor drives and develops equations to determine the length of lever arm on which the motor is mounted. The tension in the two strands of the belt and the ratio of the tensions is given by the equations

$$T_1 = \{eW/(a + b)\} + bkP/(a + b) \quad \text{[17]}$$

$$T_2 = \{eW/(a + b)\} - akP/(a + b) \quad \text{[18]}$$

$$T_1/T_2 = (eW + bkP)/(eW - akP) \quad \text{[19]}$$

where $T_1$, $T_2$ = respectively, tension in tight and slack strands of the belt; $e$, $a$, $b$ = length of lever arms, in., as shown in Fig. 20; $W$ = weight of motor, lb.; $P$ = load transmitted, Hp.; $k = 126{,}000/Nd$, in which $N$ = r.p.m. of motor, and $d$ = motor pulley diameter, in.

The maximum theoretical load that can be transmitted is $P = eW/ak$. In practice, the maximum load that can be transmitted is limited by slip of the belt, and is about 75% of the theoretical maximum.

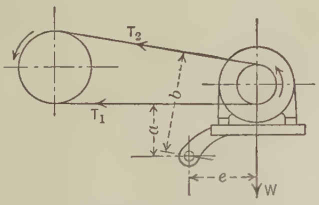


Fig. 20. Lever Arms in Pivoted Motor Drive

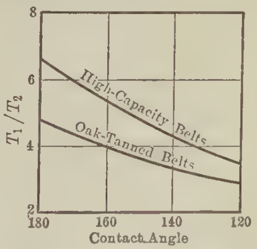


Fig. 21. Tension Ratio in Terms of Contact Angle with Paper Pulleys

In applying equations [17] to [19] to the design of a pivoted motor drive, the ratio of the tensions first is determined. High values of $T_1/T_2$ correspond to low values of $T_2$, with danger of excessive slippage. Low values of $T_1/T_2$ correspond to high total tension, with good frictional grip of the belt on the pulleys. Excessive values of $T_1/T_2$, however, tend to shorten belt life and burn out bearings. Fig. 21 gives values used by the author in connection with paper motor pulleys. Let $R$ = tension ratio selected; $W$ = weight of motor, lb.; $N$ = full speed of motor, r.p.m.; $P$ = maximum load to be transmitted, Hp., = rated Hp. × 1.5 for compensator start = rated Hp. × 2.5 for across line start; $D$, $d$, = respectively, diameter, in., of driven and motor pulleys; $r = d/2$;

$h$ = height of motor shaft above pivot; $k = 126{,}000/Nd$; $m$, $n$ = respectively, angle of slant of tight and slack strands of belt; $x$ = horizontal distance, motor shaft to driven shaft, in.; $y$ = height of driven shaft above motor shaft, in. $= -y$ if driven shaft is below motor shaft; $C$ = center distance between shafts, in. Then

$$R = T_1/T_2 = \{(eW/kP) + b\}/\{(eW/kP) - a\} \quad . \quad . \quad . \quad . \quad [20]$$

and

$$Ra + b = (R - 1)eW/kP \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [21]$$

Lengths of lever arms $a$, $b$ and $e$ are found by

$$a = h \cos m + e \sin m - r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [22]$$

$$b = h \cos n + e \sin n + r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

In the equations [22] and [23], $r$ is positive when the tight strand is on the pivot side of the pulley and negative when the slack strand is on the pivot side.

By substituting equations [22] and [23] in equation [21] and solving for $e$, we have

$$e = \frac{h(R \cos m + \cos n) - (R - 1)r}{\{(R - 1)W/kP\} - R \sin m - \sin n} \quad . \quad . \quad . \quad . \quad . \quad . \quad [24]$$

In using equation [24], $r$ is positive when $T_1$ is on the pivot side of the pulley, and negative when $T_2$ is on the pivot side. $h$ is negative if the pivot is higher than the motor shaft, as in wall or ceiling mountings. Angles $m$ and $n$ are measured upward from a horizontal axis through the motor shaft drawn toward the pivot side. If either strand of the belt slopes downward from the motor, the corresponding angle, therefore, is negative. Angles $m$ and $n$ are computed as follows: Let $A$ = the half-angle between belt strands.

$$A = \sin^{-1} (D - d)/2C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

The angle of slope of the center line of shafts is $S$.

$$S = \tan^{-1} y/x = \sin^{-1} y/C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [26]$$

$S$ is negative when $y$ is negative. Then

$$m = S - A \text{ when } T_1 \text{ is on pivot side of pulley.}$$
$$= S + A \text{ when } T_2 \text{ is on pivot side of pulley.}$$
$$n = S + A \text{ when } T_1 \text{ is on pivot side of pulley.}$$
$$= S - A \text{ when } T_2 \text{ is on pivot side of pulley.}$$

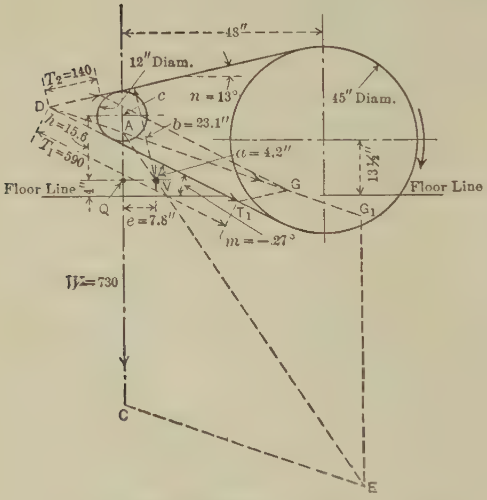


FIG. 22. Graphical Method of Laying Out Pivoted Motor Drive

EXAMPLE.—See Fig. 22. Determine the pivot distance for a drive using a high-capacity belt on a paper motor pulley, under the following conditions: Motor, 25 Hp., compensator started, whence $P = 25 \times 1.5 = 37.5$; r.p.m. $= 865 = N$; weight of motor, 730 lb. $= W$; diameter of motor pulley, 12 in. $= d$; distance of motor shaft above feet = 11 in.; distance of pivot below top surface of pivoted base arms = $4\,^9/_{16}$ in., whence $h = 11 + 4\,^9/_{16}$ in. $= 15.6$; $r = +6$ (positive, because $T_1$ is on pivot side of pulley); diameter of driven pulley, 45 in. $= D$; center line distance $= 48.3 = C$. *Solution.*—Value of $k$ is found from

$$k = 126{,}000/Nd$$
$$= 126{,}000/(865 \times 12) = 12;$$

$kP = 450$, and $WkP = 1.62$. From equation [25] $A = \sin^{-1} (45–12)/96.6 = 20$ deg. From Fig. 21, $R = 4.3$. Angles $m$ and $n$ are found by computation, or by measuring with a protractor to be $m = -27$ deg., $n = 13$ deg., whence $\cos m = 0.891$, $\sin m = -0.454$; $\cos n = 0.974$, $\sin n = 0.225$. From equation [24]

$$e = \frac{15.6\{4.3(0.891) + 0.974\} - (4.3 - 1)6}{(4.3 - 1)1.62 - 4.3(-0.454) - 0.225} = 7.83 \text{ in.}$$

The tension ratio will be 4.3 at a peak load of 37 ½ Hp. if the pivot is 7.83 in. back of the motor shaft.

**Graphical Method.**—See Fig. 22. Draw to scale the motor pulley and the driven pulley in their respective locations. Calculate the two tensions from $T_1 - T_2 = kP$ and

$T_1/T_2 = R$, whence $T_2 = kP/(R-1)$, and $T_1 = RT_2$. Lay off to scale the tensions $T_1$ and $T_2$ from their point of intersection $D$. Construct the parallelogram of forces, and draw the resultant $DG$. Draw the vertical center line of the motor pulley, making $cC$ equal to the weight of the motor, to the same scale as used for the tensions. Prolong $DG$ to $G_1$, making $AG_1 = DG$. On $AG_1$ draw the parallelogram $AG_1EC$, and draw the diagonal $AE$. On $cC$ locate point $Q$, making $cQ$ equal to the distance $h$ of the center line of the motor above the pivot. This distance $h$ is fixed by the dimensions of the motor and the pivoted base. Through $Q$ draw a horizontal line intersecting $AE$ at $V$, which is the location of the pivot, and which can be scaled from the drawing.

## 7. BELT DATA

**U. S. NAVY DEPARTMENT SPECIFICATIONS FOR BELTING.**—Belts shall be made from No. 1 native packer steer hides or their equal. They shall be tanned with white or chestnut oak by the slow process for six or eight months; chemical processes are not permitted. The leather is to be thoroughly curried by hand and must not be stuffed or loaded for artificial weight. The leather must not crack open on the grain side when doubled strongly by hand with the grain side out. All belting shall be cut from the central part of the hide, not more than 15 in. from the backbone or more than 48 in. from the tail toward the shoulder. Belts 8 in. wide and over must be cut to include the backbone. All leather must be stretched 6 in. in the lengthwise direction and should not exceed 54 in. after stretching. Centers and sides are to be stretched 6 in. separately. That is, all sides from which widths of less than 8 in. are cut must be stretched after the backbone section has been removed. Center sections are to be stretched in the same size for which they are to be used.

For single belts up to 6 in. wide the laps must not exceed 6 in. nor be less than 3 1/2 in. long. For single belts wider than 6 in., the laps should not be more than 1 in. longer than the width of the belt. For double belts, laps must not be more than 5 1/2 in. nor less than 3 1/2 in. long. Filling straps are not permitted. All laps must be held securely at every point with the best quality of belt cement, and when pulled apart must not show any resinous, oily, vitreous or watery condition. The belts should be again stretched after manufacture.

**Weight.**—Single belts, all sizes, 16 oz. per square foot; double belts: 1 to 2 in. wide, 26 oz. per sq. ft.; 2 1/2 to 4 in. wide, 28 oz. per sq. ft.; 4 1/2 to 5 1/2 in. wide, 30 oz. per sq. ft.; 6 in. wide and over 32 oz. per sq. ft.

**Rawhide Lacing.**—Only hand-cut green slaughter-house hides are to be used for lacing. They are to be cut in the following sizes: 1/4, 5/16, 3/8, 7/16, 1/2, 5/8 and 3/4 in. They must be cut lengthwise from the hide and have a tensile strength of not less than

| Width, in. | 1/4 | 5/16 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 |
|---|---|---|---|---|---|---|---|
| Tensile strength, lb. per sq. in. | 95 | 125 | 155 | 165 | 180 | 205 | 230 |

**LENGTH OF BELTS.**—Whenever possible, the length of belt required for a pair of pulleys should be determined by actual measurement by means of a steel tape passed around the pulleys. Where this is impracticable, the length of an open belt can be determined by calculation, using one of the three following formulas (Carl G. Barth, *Am. Mach.*, Mar. 12, 1903):

$$L = \frac{(D+d)\pi}{2} + Cx + 2C \qquad [27]$$

$$L = \frac{(D+d)\pi}{2} + Cx\frac{12}{12-x} + 2C \qquad [28]$$

$$L = \frac{(D+d)\pi}{2} + Cx\frac{60-13x}{60-18x} + 2C \qquad [29]$$

where $L$ = length of belt; $D$ = diam. of larger pulley and $d$ = diam. of smaller pulley; $C$ = center to center distance of pulleys, all in the same unit, in. or ft., $\pi = 3.1416$, and $x = \{(D-d)/2C\}^2$. The accuracy of the formulas is as follows, testing them by the limiting case where $d = 0$ and $C = D/2$, giving $L = D\pi$: Formula [27] gives a result somewhat over 2% short; formula [28] gives a result about 1% short; formula [29] gives a result under 0.4% short.

**Crossed Belts.**—Using above notation, length of crossed belts is given exactly by

$$L = \frac{(D+d)\pi}{2} + (D+d)\frac{\pi\beta}{180} + 2L\cos\beta; \quad \beta = \sin^{-1}\frac{D+d}{2L} \qquad [30]$$

**LENGTH AND WEIGHT OF BELTING.**—The length of a belt when closely rolled may be found as follows: Let $D$ = diameter of roll, in., $d$ = diameter of eye, in.,

$N$ = number of coils in roll, $L$ = length of belt, feet. Then $L = 0.1309\ (D + d)N$.

**Approximate Weight of Belts.**—Let $L$ = length of belt, ft., $w$ = width of belt, in., $C$ = a constant = 13 for single, and 8 for double belts. Then, weight of belt, $W$, lb., is $W = L \times w/C$.

**THICKNESS OF LEATHER BELTING.**—Specifications of the American Leather Belting Assoc. give the thickness of first quality leather belting as follows:

| Medium Single | Heavy Single | Light Double | Medium Double | Heavy Double |
|---|---|---|---|---|
| $^{10}/_{64}$–$^{12}/_{64}$ in. | $^{12}/_{64}$–$^{14}/_{64}$ in. | $^{15}/_{64}$–$^{17}/_{64}$ in. | $^{18}/_{64}$–$^{20}/_{64}$ in. | $^{21}/_{64}$–$^{23}/_{64}$ in. |

The thicknesses are average thicknesses in inches, determined by measuring 20 coils and dividing this value by the number of coils measured. In rolls of belting containing less than 20 coils, average thickness should be determined by measuring one-half the total number of coils and dividing this value by the number of coils measured. No point in either single or double belting shall be more than $^2/_{64}$ in. thicker or thinner than the average thickness.

**STRENGTH OF BELTING.**—The ultimate tensile strength of belting does not generally enter as a factor in calculations of power transmission. The strength of the solid leather in belts is from 2000 to 5000 lb. per sq. in.; at the lacings, even if well put together, only about 1000 to 1500. If riveted, the joint should have half the strength of the solid belt. The working strain on the driving side generally is taken at not over one-third of the strength of the lacing, or from one-eighth to one-sixteenth of the strength of the solid belt.

**SAG OF BELTS. DISTANCE BETWEEN PULLEYS.**—Shafts that are to be connected with belts should be at such distance as to allow of a gentle sag to the belt when in motion. A general rule is: Where narrow belts are to be run over small pulleys, 15 ft. is a good average, the belt having a sag of 1 $^1/_2$ to 2 in. For larger belts, working on larger pulleys, the distance should be 20 to 25 ft., with a sag of 2 $^1/_2$ to 4 in. For main belts working on very large pulleys, the distance should be 25 to 30 ft., with a sag of 4 to 5 in. If too great a distance is attempted, the belt will have an unsteady flapping motion, which will destroy both the belt and machinery.

**EFFECT OF HUMIDITY UPON A LEATHER BELT.** (W. W. Bird and F. W. Roys, *Trans.* A.S.M.E., 1915.) Tests with a 4-in. oak-tanned single belt, with constant horsepower transmitted, and with the center distance and humidity varying, showed increase of the sum of the tensions as the humidity decreased. Figures taken from curves of the results are as follows:

| Center distance | 9 ft. 6 in. | 9 ft. 6 $^1/_2$ in. | 9 ft. 7 in. | 9 ft. 7 $^1/_2$ in. |
|---|---|---|---|---|
| Relative Humidity | | Sum of the Tensions, pounds | | |
| 90 | 95 | 210 | 325 | 445 |
| 55 | 125 | 260 | 400 | 550 |
| 20 | 150 | 310 | 465 | 620 |

Increase of temperature as well as increase of humidity tends to lengthen the belt and decrease the tension. The most important conclusions are:

1. If a belt be set up at a medium relative humidity, the tensions will not be excessive at lower relative humidities, nor will there be any great danger of slipping at high relative humidities unless there are excessive temperature changes.

2. If a belt be set up at any relative humidity with a spring or gravity tightener, a load 50% greater than the standard can be transmitted at either high or low humidity without danger of stretching the belt, slipping, or excessive pressure on the bearings.

**COEFFICIENT OF FRICTION.**—The coefficient of friction of a belt on a pulley varies with the character of the belt and with the type of surface and condition of surface of the pulley. It also varies with the velocity of the belt. (See p. 24–15.) Table 15 from

**Table 15.—Coefficient of Friction of Belts on Pulleys**

| Kind of Belt | Friction Surface of Pulley | | | | | | |
|---|---|---|---|---|---|---|---|
| | Iron | Steel | Wood | Paper | Wet Iron | Greasy Iron | Oily Iron |
| Oak-tanned leather | 0.25 | 0.25 | 0.30 | 0.35 | 0.20 | 0.15 | 0.12 |
| Mineral-tanned leather | .40 | .40 | .45 | .50 | .35 | .25 | .20 |
| Canvas stitched | .20 | .20 | .23 | .25 | .15 | .12 | .10 |
| Balata | .32 | .32 | .35 | .40 | .20 | ........ | ........ |
| Cotton woven | .22 | .22 | .25 | .28 | .15 | .12 | .10 |
| Camel-hair | .35 | .35 | .40 | .45 | .25 | .20 | .15 |
| Rubber-friction | .30 | .30 | .32 | .35 | .18 | ........ | ........ |
| Rubber-covered | .32 | .32 | .35 | .38 | .15 | ........ | ........ |
| Rubber on fabric | .35 | .35 | .38 | .40 | .20 | ........ | ........ |

*Machy.*, March 1931, gives the coefficient of friction of various types of belts on various pulley surfaces. These coefficients probably are average values and take no account of the variation due to velocity.

## 8. MISCELLANEOUS TYPES OF BELTS

**RUBBER BELTING.**—The advantages claimed for rubber belting are perfect uniformity in width and thickness; it will endure a great degree of heat and cold without injury; it is also specially adapted for use in damp or wet places, or where exposed to the action of steam; it is very durable, and has great tensile strength, and when adjusted for service it has the most perfect hold on the pulleys, hence is less liable to slip than leather.

*Never use animal oil or grease on rubber belts*, as it will soon destroy them.

Rubber belts will be improved, and their durability increased, by putting on with a painter's brush, and letting it dry, a composition made of equal parts of red lead, black lead, French yellow, and litharge, mixed with boiled linseed-oil and japan enough to make it dry quickly. The effect of this will be to produce a finely polished surface. If, from dust or other cause, the belt should slip, it should be lightly moistened on the pulley side with boiled linseed-oil. Table 16 gives the properties of rubber belting and the maximum tensions under which it should be operated.

**STITCHED CANVAS BELTING.**—The widths of stitched canvas belts vary with the thickness, ranging as follows: 4-ply, 1 1/2 to 18 in.; 3-ply, 3 to 30 in.; 8-ply, 40 to 48 in.; 10-ply, 12 to 60 in. Widths increase by 1/2 in. up to 5 in. wide, and by 1 in. up to 14 in. From 14 in. to 32 in. widths increase by 2 in. Above 32 in., canvas belts are made 36, 40, 48, 54 and 60 in. wide. Table 17 shows the properties of stitched canvas belting.

**TEON BELTING** is a cotton belt impregnated with a special composition to enable the belt to work effectively in heat, steam, water and chemically charged atmospheres under abnormal conditions. It is reinforced with a stitching of copper wire extending in rows lengthwise of the belt, which practically eliminates stretching, the stretch at 190 lb. per sq. in. of belt tension being 0.09 in. The belt is unaffected by temperature variation up to 250° F. Table 18 gives the horsepower capacity of these belts as stated by the maker.

Teon belting is made in the following widths: Light belt, 1, 1 1/4, 1 1/2, 2, 2 1/2, 3, 3 1/2, 4 in.; min. diam. of pulley, 3 in. Medium belt, 1 to 2 in., increasing by 1/4 in.

**Table 16.—Properties of Rubber Belting**

| Kind of Belt | Coefficient of Friction, $f$ | 28 oz. | | | | 32 oz. | | | | 36 oz. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. |
| 3 plies | 0.30 | 0.864 | 72 | 0.150 | 6–9 | 0.972 | 75 | 0.165 | 10 | 1.080 | 78 | 0.180 | 11 |
| 4 plies | .30 | 1.152 | 96 | .200 | 8–12 | 1.296 | 100 | .220 | 13 | 1.440 | 104 | .240 | 14 |
| 5 plies | .30 | 1.440 | 120 | .250 | 10–15 | 1.620 | 125 | .275 | 16 | 1.800 | 130 | .300 | 18 |
| 6 plies | .30 | 1.728 | 144 | .300 | 15–18 | 1.944 | 150 | .330 | 20 | 2.160 | 156 | .360 | 20 |
| 7 plies | .30 | 2.016 | 168 | .350 | 21 | 2.268 | 175 | .385 | 24 | 2.520 | 182 | .420 | 26 |
| 8 plies | .30 | 2.304 | 192 | .400 | 24 | 2.592 | 200 | .440 | 26 | 2.880 | 208 | .480 | 28 |
| 10 plies | .30 | 2.880 | 240 | .500 | 30 | 3.240 | 250 | .550 | 32 | 3.600 | 260 | .600 | 36 |
| 12 plies | .30 | 3.456 | 288 | .600 | 36 | 3.888 | 300 | .660 | 40 | | | | |

**Table 17.—Properties of Stitched Canvas Belting**

| Kind of Belt | Coefficient of Friction, $f$ | 28 oz. | | | | 32 oz. | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. |
| 3 plies | 0.20 | 0.864 | 72 | 0.135 | 8 | 0.972 | 75 | 0.150 | 9 |
| 4 plies | .20 | 1.152 | 96 | .180 | 12 | 1.296 | 100 | .200 | 14 |
| 5 plies | .20 | 1.440 | 120 | .225 | 16 | 1.620 | 125 | .250 | 18 |
| 6 plies | .20 | 1.728 | 144 | .270 | 20 | 1.944 | 150 | .300 | 24 |
| 7 plies | .20 | 2.016 | 168 | .315 | 26 | 2.268 | 175 | .350 | 30 |
| 8 plies | .20 | 2.304 | 192 | .360 | 32 | 2.592 | 200 | .400 | 36 |
| 10 plies | .20 | 2.880 | 240 | .450 | 40 | 3.240 | 250 | .500 | 48 |

increments; 2 $^1/_2$ to 4 $^1/_2$ in., increasing by $^1/_2$ in. increments; 5 to 10 in., increasing by 1 in. increments; 12 in.; 14 to 20 in., increasing by 1 in. increments; min. diam. of pulley, 4 in. Medium heavy belt, 3, 4, 5, 6 in.; min. diam. of pulley 7 in. Heavy belt, 3 to 26 in. increasing by 1 in. increments; min. diam. of pulley, 9 in. Teon belting is made by D. P. Brown & Co., Philadelphia, Pa.

**Table 18.—Horsepower Transmitting Capacity of Teon Belts per Inch of Width ***

| Belt Velocity, ft. per min. | Light | Medium | Heavy | Extra Heavy | Triple | Extra Triple | Belt Velocity, ft. per min. | Light | Medium | Heavy | Extra Heavy | Triple | Extra Triple |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Horsepower | | | | | | | Horsepower | | | | | |
| 500 | 0.70 | 1.04 | 1.70 | 2.40 | 2.70 | 3.00 | 3000 | 4.00 | 5.98 | 10.01 | 13.99 | 15.97 | 17.97 |
| 750 | 1.04 | 1.56 | 2.50 | 3.50 | 4.00 | 4.49 | 3250 | 4.30 | 6.49 | 10.73 | 15.15 | 17.31 | 19.48 |
| 1000 | 1.40 | 2.05 | 3.36 | 4.66 | 5.32 | 5.99 | 3500 | 4.66 | 6.99 | 11.66 | 16.32 | 19.75 | 20.98 |
| 1250 | 1.70 | 2.50 | 4.16 | 5.83 | 6.66 | 7.50 | 3750 | 4.99 | 7.49 | 12.49 | 17.49 | 19.99 | 22.48 |
| 1500 | 2.05 | 3.01 | 5.00 | 6.99 | 7.99 | 8.98 | 4000 | 5.32 | 7.99 | 13.33 | 18.65 | 21.34 | 23.99 |
| 1750 | 2.40 | 3.50 | 5.85 | 8.16 | 9.32 | 10.49 | 4250 | 5.66 | 8.49 | 14.16 | 19.82 | 22.66 | 25.48 |
| 2000 | 2.70 | 4.00 | 6.66 | 9.33 | 10.67 | 12.00 | 4500 | 6.00 | 8.99 | 15.00 | 20.99 | 23.98 | 26.99 |
| 2250 | 3.01 | 4.50 | 7.50 | 10.49 | 11.99 | 13.54 | 4750 | 6.32 | 9.49 | 15.84 | 22.15 | 25.32 | 28.46 |
| 2500 | 3.36 | 4.99 | 8.32 | 11.66 | 13.33 | 15.00 | 5000 | 6.66 | 9.98 | 16.67 | 23.30 | 26.66 | 30.00 |
| 2750 | 3.70 | 5.49 | 9.17 | 12.82 | 14.66 | 16.48 | | | | | | | |

* Based on 180 deg. arc of contact on both pulleys. For other arcs of contact, the tabular values should be increased or decreased as follows: 210 deg., + 11%; 200 deg., + 7%; 190 deg., + 4%; 170 deg., −4%; 160 deg., −8%; 150 deg., −13%.

**BALATA BELTING** is canvas belting impregnated with balata gum, which makes it waterproof and resistant to animal oil. Mineral oils affect it adversely, as do temperatures above 120° F. Table 19 gives the properties of balata belting.

**Table 19.—Properties of Balata Belting**

| Kind of Belt | 28 oz. | | | | | Leather Strips | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. | Coefficient of Friction, $f$ | Weight, lb. per sq. ft. | Maximum Tight Side Tension, lb. per sq. in. | Thickness, in. | Minimum Pulley Diameter, in. | Coefficient of Friction, $f$ |
| 3 plies | 0.864 | 72 | 0.150 | 3 | 0.32 | 1.380 | 72 | $^{17}/_{64}$ | 3–5 | 0.40 |
| 4 plies | 1.152 | 96 | .200 | 5 | .32 | 1.692 | 96 | $^{5}/_{16}$ | 5–8 | .40 |
| 5 plies | 1.440 | 120 | .250 | 8 | .32 | 1.944 | 120 | $^{23}/_{64}$ | 8–12 | .40 |
| 6 plies | 1.728 | 144 | .300 | 12 | .32 | 2.256 | 144 | $^{13}/_{32}$ | 12–15 | .40 |
| 7 plies | 2.016 | 168 | .350 | 15 | .32 | 2.508 | 168 | $^{29}/_{64}$ | 15–18 | .40 |
| 8 plies | 2.304 | 192 | .400 | 18 | .32 | 2.820 | 192 | $^{1}/_{2}$ | 18–21 | .40 |
| 10 plies | 2.880 | 240 | .500 | 30 | .32 | | | | | |

## CHAIN DRIVES

Power transmission chains may be broadly classified as roller chains, block chains, inverted-tooth or silent chains, and malleable chains. The latter are used principally in conveyor applications and for very slow speed drives. For the transmission of power at high chain velocities, the other types are preferred. Block chains comprise 8-shaped or B-shaped blocks with side links riveted to them at both ends. Block chains are used only for light transmissions and at speeds under 800 ft. per min.

### 1. ROLLER CHAINS

Roller chains have been standardized, and all manufacturers of this type of chain build their chains and sprockets in conformity with American Standard A.S.A.–B29a–1930. The following notes are abstracted from that standard. This standard is subject to periodic revision and the latest issue should be consulted.

**TYPES OF ROLLER CHAINS.**—A roller chain comprises a series of pin-links and roller-links assembled together. The roller-link consists of two plates, between which are inserted a pair of bushings, each carrying a steel roller. The pin-link consists of a pair of pins riveted to a side plate, the pins passing through the bushing of the adjacent roller link. They are held in place by a side plate fitting over the projecting ends of the pins.

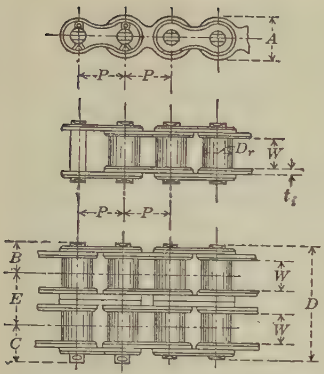


FIG. 1. Dimensions of Commercial Standard Roller Chain

The pins either are riveted over the side plate or are secured by cotters. The bushings are free to turn on the pins, and the rollers are free to turn on the bushings. The distance between the center lines of the pin-links and between the center lines of the bushings of the roller-links is equal to the pitch of the chain. The plates forming the tension members of a roller-link are known as inside plates, and the plates forming the tension members of the pin-link are known as pin-link plates.

Roller chains are made in the standard series, the extra heavy series, and light-weight machinery chain, and also in single, double, triple and quadruple width chain. The center plates in multiple width chains have a thickness equivalent to two thicknesses of inside plates as used on the roller links. Dimensions of commercial standard roller chain are given in Table 1. Light-weight machinery chain, designated as No. 41, is 1/2 in. pitch, 1/4 in. wide, and has side plates 0.050 in. thick. The pin diameter is 0.141 in., and the roller diameter is 0.306 in.

**Table 1.—Dimensions of Roller Chains**
(Morse Chain Co., Ithaca, N. Y.)

| Chain No. | Pitch $P$, in. | Nominal Roller Diam., $D_r$, in. | Roller Width, $W$, in. | Dimensions, in. (See Fig. 1) | | | | | Weight per Foot, lb. | Average Ultimate Strength, lb. |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $A$ | $B$ | $C$ | $D$ | $E$ | | |
| Single Chains | | | | | | | | | | |
| 35 | 3/8 | 0.200 | 3/16 | 0.350 | 0.230 | 0.250 | 0.460 | ....... | 0.21 | 1,400 |
| 40 | 1/2 | 5/16 | 5/16 | .466 | .321 | .391 | .642 | ....... | .38 | 3,200 |
| 41 | 1/2 | 0.306 | 1/4 | .380 | .260 | .280 | .520 | ....... | .25 | 2,000 |
| 50 | 5/8 | 0.4 | 3/8 | .584 | .400 | .439 | .801 | ....... | .69 | 5,200 |
| 60 | 3/4 | 15/32 | 1/2 | .700 | .499 | .538 | .999 | ....... | 1.05 | 7,300 |
| 80 | 1 | 5/8 | 5/8 | .934 | .643 | .699 | 1.287 | ....... | 1.75 | 12,200 |
| 100 | 1 1/4 | 3/4 | 3/4 | 1.166 | .787 | .850 | 1.574 | ....... | 2.60 | 21,000 |
| 120 | 1 1/2 | 7/8 | 1 | 1.400 | .991 | 1.089 | 1.982 | ....... | 3.80 | 30,500 |
| 140 | 1 3/4 | 1 | 1 | 1.634 | 1.071 | 1.154 | 2.142 | ....... | 5.14 | 42,500 |
| 160 | 2 | 1 1/8 | 1 1/4 | 1.866 | 1.274 | 1.368 | 2.548 | ....... | 6.87 | 52,500 |
| Double Chains | | | | | | | | | | |
| 35D | 3/8 | 0.200 | 3/16 | 0.350 | 0.230 | 0.250 | 0.860 | 0.400 | 0.40 | 2,800 |
| 40D | 1/2 | 5/16 | 5/16 | .466 | .321 | .391 | 1.207 | .565 | .81 | 6,400 |
| 50D | 5/8 | 0.4 | 3/8 | .584 | .400 | .439 | 1.510 | .710 | 1.40 | 10,400 |
| 60D | 3/4 | 15/32 | 1/2 | .700 | .499 | .538 | 1.891 | .893 | 2.12 | 14,600 |
| 80D | 1 | 5/8 | 5/8 | .934 | .643 | .699 | 2.433 | 1.147 | 3.83 | 24,400 |
| 100D | 1 1/4 | 3/4 | 3/4 | 1.166 | .787 | .850 | 2.974 | 1.400 | 5.46 | 42,000 |
| 120D | 1 1/2 | 7/8 | 1 | 1.400 | .991 | 1.089 | 3.761 | 1.779 | 8.00 | 61,000 |
| 140D | 1 3/4 | 1 | 1 | 1.634 | 1.071 | 1.154 | 4.054 | 1.912 | 10.78 | 85,000 |
| 160D | 2 | 1 1/8 | 1 1/4 | 1.866 | 1.274 | 1.368 | 4.839 | 2.291 | 14.40 | 105,000 |
| Triple Chains | | | | | | | | | | |
| 35E | 3/8 | 0.200 | 3/16 | 0.350 | 0.230 | 0.250 | 1.260 | 0.400 | 0.66 | 4,200 |
| 40E | 1/2 | 5/16 | 5/16 | .466 | .321 | .391 | 1.772 | .565 | 1.30 | 9,600 |
| 50E | 5/8 | 0.4 | 3/8 | .584 | .400 | .439 | 2.220 | .710 | 2.10 | 15,600 |
| 60E | 3/4 | 15/32 | 1/2 | .700 | .499 | .538 | 2.784 | .893 | 3.30 | 21,900 |
| 80E | 1 | 5/8 | 5/8 | .934 | .643 | .699 | 3.580 | 1.147 | 5.52 | 36,600 |
| 100E | 1 1/4 | 3/4 | 3/4 | 1.166 | .787 | .850 | 4.374 | 1.400 | 8.40 | 63,000 |
| 120E | 1 1/2 | 7/8 | 1 | 1.400 | .991 | 1.089 | 5.540 | 1.779 | 12.00 | 91,500 |
| 140E | 1 3/4 | 1 | 1 | 1.634 | 1.071 | 1.154 | 5.966 | 1.912 | 17.00 | 127,500 |
| 160E | 2 | 1 1/8 | 1 1/4 | 1.866 | 1.274 | 1.368 | 7.130 | 2.291 | 26.67 | 157,500 |
| Quadruple Chains | | | | | | | | | | |
| 35F | 3/8 | 0.200 | 3/16 | 0.350 | 0.230 | 0.250 | 1.660 | 0.400 | 0.88 | 5,600 |
| 40F | 1/2 | 5/16 | 5/16 | .466 | .321 | .391 | 2.337 | .565 | 1.75 | 12,800 |
| 50F | 5/8 | 0.4 | 3/8 | .584 | .400 | .439 | 2.930 | .710 | 2.80 | 20,800 |
| 60F | 3/4 | 15/32 | 1/2 | .700 | .499 | .538 | 3.677 | .893 | 4.26 | 29,200 |
| 80F | 1 | 5/8 | 5/8 | .934 | .643 | .699 | 4.727 | 1.147 | 7.10 | 48,800 |
| 100F | 1 1/4 | 3/4 | 3/4 | 1.166 | .787 | .850 | 5.774 | 1.400 | 11.18 | 84,000 |
| 120F | 1 1/2 | 7/8 | 1 | 1.400 | .991 | 1.089 | 7.319 | 1.779 | 16.34 | 122,000 |
| 140F | 1 3/4 | 1 | 1 | 1.634 | 1.071 | 1.154 | 7.878 | 1.912 | 22.00 | 170,000 |
| 160F | 2 | 1 1/8 | 1 1/4 | 1.866 | 1.274 | 1.368 | 9.421 | 2.291 | 29.50 | 210,000 |

Dimensions of the standard and extra heavy series are given in Table 2. Extra heavy chains are similar in all dimensions to the standard chain, with the exception of the inside plates, which are thicker than the standard plates.

**Table 2.—Dimensions of Standard Roller Chain**

(American Standards Assoc., B29a—1930)

| Pitch $P$, in. | Roller Diam., $D_r$, in. | Standard Series: Width, $W$, in. | Pin Diam., $D_p$, in. | Thickness of Inside Plate $t_p$, in. | Test Load, lb. | Measuring Load, lb. | Standard Chain No. | Extra Heavy Series: Thickness of Inside Plate $t_p$, in. |
|---|---|---|---|---|---|---|---|---|
| 3/8 | 0.200 | 3/16 | 0.141 | 0.050 | 555 | 17 | 35N | ........ |
| 1/2 | 5/16 | 5/16 | .156 | .060 | 740 | 23 | 40 | ........ |
| 5/8 | 0.400 | 3/8 | .200 | .080 | 1,400 | 44 | 50 | ........ |
| 3/4 | 15/32 | 1/2 | .234 | .094 | 2,018 | 63 | 60 | 0.125 |
| 1 | 5/8 | 5/8 | .312 | .125 | 3,820 | 119 | 80 | .156 |
| 1 1/4 | 3/4 | 3/4 | .375 | .156 | 5,626 | 176 | 100 | .187 |
| 1 1/2 | 7/8 | 1 | .437 | .187 | 7,760 | 242 | 120 | .219 |
| 1 3/4 | 1 | 1 | .500 | .219 | 10,220 | 319 | 140 | .250 |
| 2 | 1 1/8 | 1 1/4 | .562 | .250 | 13,008 | 406 | 160 | .281 |
| 2 1/2 | 1 9/16 | 1 1/2 | .781 | .312 | 25,340 | 792 | 200 | .375 |
| 3 | 1.90 | 1 7/8 | .937 | .375 | 36,634 | 1,145 | 240 | .469 |

**Table 3.—Horsepower and Loads of Standard Roller Chains**

Loads in pounds

| Chain No. | Pitch, in. | Maximum, r.p.m. | Chain Velocity, ft. per min. | 50 | 100 | 200 | 300 | 400 | 600 | 800 | 1000 | 1300 | 1600 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 35N | 3/8 | 3620 | Hp. | 0.25 | 0.45 | 0.79 | 1.06 | 1.27 | 1.60 | 1.82 | 1.94 | 2.07 | 2.10 |
| | | | Load | 162 | 150 | 132 | 117 | 105 | 88 | 75 | 64 | 52 | 43 |
| 40 | 1/2 | 2712 | Hp. | 0.41 | 0.76 | 1.33 | 1.78 | 2.12 | 2.65 | 2.98 | 3.25 | 3.41 | 3.44 |
| | | | Load | 269 | 250 | 219 | 194 | 175 | 146 | 123 | 107 | 87 | 71 |
| 50 | 5/8 | 1927 | Hp. | 0.64 | 1.20 | 2.11 | 2.82 | 3.38 | 4.22 | 4.75 | 5.13 | 5.4 | 5.47 |
| | | | Load | 428 | 397 | 348 | 308 | 278 | 232 | 196 | 169 | 137 | 113 |
| 60 | 3/4 | 1454 | Hp. | 0.93 | 1.80 | 2.03 | 4.06 | 4.86 | 6.07 | 7.15 | 7.68 | 8.10 | 8.20 |
| | | | Load | 645 | 597 | 524 | 462 | 419 | 348 | 295 | 253 | 206 | 169 |
| 80 | 1 | 941 | Hp. | 1.66 | 3.07 | 5.58 | 7.20 | 8.64 | 10.8 | 12.1 | 13.0 | 13.75 | 13.85 |
| | | | Load | 1092 | 1014 | 891 | 785 | 712 | 592 | 507 | 431 | 349 | 286 |
| 100 | 1 1/4 | 652 | Hp. | 2.41 | 4.48 | 7.88 | 10.5 | 12.6 | 15.7 | 17.6 | 18.85 | 19.9 | 20.0 |
| | | | Load | 1592 | 1478 | 1294 | 1153 | 1035 | 862 | 727 | 622 | 506 | 412 |
| 120 | 1 1/2 | 515 | Hp. | 3.63 | 6.76 | 11.8 | 15.8 | 18.9 | 23.6 | 26.5 | 28.7 | 30.2 | 30.2 |
| | | | Load | 2415 | 2230 | 1960 | 1730 | 1565 | 1306 | 1098 | 946 | 766 | 623 |
| 140 | 1 3/4 | 375 | Hp. | 4.36 | 8.11 | 14.2 | 19.0 | 22.8 | 28.4 | 31.7 | 34.0 | 36.0 | 36.0 |
| | | | Load | 2880 | 2674 | 2330 | 2070 | 1870 | 1560 | 1308 | 1127 | 914 | 742 |
| 160 | 2 | 316 | Hp. | 5.96 | 11.10 | 19.4 | 25.8 | 31.1 | 38.8 | 43.4 | 46.7 | 49.0 | 49.0 |
| | | | Load | 3940 | 3658 | 3202 | 2840 | 2560 | 2130 | 1792 | 1540 | 1244 | 1013 |
| 200 | 2 1/2 | 224 | Hp. | 10.06 | 18.70 | 32.7 | 43.6 | 52.3 | 65.4 | 73.5 | 79.0 | 84.0 | 84.0 |
| | | | Load | 6636 | 6161 | 5390 | 4790 | 4320 | 3600 | 3030 | 2605 | 2130 | 1742 |

**Table 4.—Horsepower and Loads of Heavy Roller Chains**

Loads in pounds

| Chain No. | Pitch, in. | Maximum r.p.m. | Chain Velocity, ft. per min. | 50 | 100 | 200 | 300 | 400 | 600 | 800 | 1000 | 1300 | 1600 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 60H | 3/4 | 1306 | Hp. | 1.07 | 1.97 | 3.46 | 4.62 | 5.53 | 6.92 | 7.76 | 8.35 | 8.84 | 8.84 |
| | | | Load | 702 | 652 | 570 | 507 | 457 | 382 | 321 | 276 | 224 | 182 |
| 80H | 1 | 871 | Hp. | 1.77 | 3.29 | 5.76 | 7.68 | 9.20 | 11.5 | 12.9 | 13.9 | 14.7 | 14.7 |
| | | | Load | 1170 | 1085 | 950 | 840 | 760 | 634 | 533 | 459 | 372 | 303 |
| 100H | 1 1/4 | 595 | Hp. | 2.55 | 4.73 | 8.30 | 11.1 | 13.3 | 16.6 | 18.6 | 20.0 | 21.0 | 21.0 |
| | | | Load | 1680 | 1565 | 1368 | 1215 | 1093 | 914 | 767 | 659 | 534 | 433 |
| 120H | 1 1/2 | 471 | Hp. | 3.79 | 7.03 | 12.4 | 16.4 | 19.7 | 24.6 | 27.7 | 29.7 | 31.3 | 31.3 |
| | | | Load | 2500 | 2320 | 2032 | 1810 | 1630 | 1360 | 1144 | 982 | 796 | 644 |
| 140H | 1 3/4 | 350 | Hp. | 4.55 | 8.45 | 14.8 | 19.7 | 23.7 | 29.6 | 33.2 | 35.6 | 37.5 | 37.5 |
| | | | Load | 3000 | 2780 | 2440 | 2170 | 1950 | 1630 | 1366 | 1175 | 952 | 771 |
| 160H | 2 | 297 | Hp. | 6.18 | 11.5 | 20.1 | 26.8 | 32.2 | 40.2 | 44.8 | 48.3 | 50.6 | 50.6 |
| | | | Load | 4080 | 3790 | 3320 | 2880 | 2650 | 2210 | 1853 | 1596 | 1287 | 1042 |
| 200H | 2 1/2 | 213 | Hp. | 10.7 | 19.7 | 34.6 | 46.2 | 55.3 | 69.2 | 77.5 | 83.5 | 88.5 | 88.5 |
| | | | Load | 7020 | 6520 | 5700 | 5070 | 4560 | 3810 | 3199 | 2754 | 2248 | 1828 |

**PROPORTIONS OF CHAINS.**—Standard chains are made in the following pitches: 3/8 in., 1/2 in., 5/8 in., 3/4 in., 1 in., 1 1/4 in., 1 1/2 in., 1 3/4 in., 2 in.

If $P$ be the pitch of the chain, in., the proportions of standard chains are determined approximately as follows: Roller diameters, $5/8\ P$; width (minimum distance between inside plates) is the nearest binary fraction to $5/8\ P$; in the narrow series, width is the nearest binary fraction to $0.41\ P$. Pin diameters are approximately $5/16\ P$, or 1/2 of the roller diameter. Thickness of inside plates, standard series, is approximately $1/8\ P$.

Under exceptional conditions, values given by the above equations may be increased by as much as 25%, but only when lubrication and installation, as well as the design of the drive is correct. If lubrication is poor, or the chain is subject to suddenly applied loads or continuous service, or where unusually long life is required, or if the sprockets are subject to misalignment, chains should be selected on a basis of approximately one-half of the loads given by the equations. Tables 3 and 4 show the working loads and horsepowers transmitted by various chains on the basis of equations [2] and [3]. The tables are based on sprockets having not less than 15 teeth.

**CHAIN VELOCITY.**—A chain that is satisfactory in all other respects may prove impracticable due to its velocity being limited by the permissible velocity of the sprocket. The maximum allowable impact between chain rollers and sprocket teeth limits sprocket speed to

$$\text{r.p.m.} = (1920/P) \times \sqrt{A_r/W_f P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [1]$$

where $P$ = pitch, in.; $A_r$ = projected area of roller, sq. in.; $W_f$ = weight of chain, lb. per ft. The maximum revolutions per min. of sprockets as calculated from this equation for standard roller chains are given in Tables 3 and 4. Maximum chain velocities are given in Table 5.

**Maximum Chain Velocity** $= 75\ n \div \sqrt{P}$, where $n$ is the number of teeth in the smaller sprocket. Under favorable conditions, the chain velocity can be very high but with

**Table 5.—Maximum Velocities of Standard Roller Chain**

| No. of Teeth in Sprocket | | Chain No. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 35N | 40 | 41 | 50 | 60 | 80 | 100 | 120 | 140 | 160 | 200 |
| | | Pitch | | | | | | | | | | |
| | | 3/8 | 1/2 | 1/2 | 5/8 | 3/4 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/2 |
| | | Roll Diameter, in. | | | | | | | | | | |
| | | 0.2 | 5/16 | 0.306 | 0.4 | 15/32 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 9/16 |
| 7 | Max. R.p.m. | 3600 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 788 | 758 | 846 | 692 | 657 | 549 | 470 | 455 | 378 | 379 | 350 |
| 9 | Max. R.p.m. | 3600 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1015 | 975 | 1088 | 890 | 844 | 705 | 604 | 585 | 486 | 487 | 450 |
| 11 | Max. R.p.m. | 3600 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1238 | 1191 | 3160 | 1089 | 1032 | 862 | 738 | 715 | 594 | 595 | 550 |
| 13 | Max. R.p.m. | 3600 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1462 | 1410 | 1570 | 1287 | 1220 | 1019 | 872 | 845 | 702 | 704 | 650 |
| 14 | Max. R.p.m. | 3467 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1517 | 1515 | 1694 | 1384 | 1313 | 1097 | 940 | 910 | 755 | 758 | 700 |
| 15 | Max. R.p.m. | 3350 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft per min. | 1570 | 1624 | 1810 | 1480 | 1407 | 1173 | 1010 | 975 | 810 | 813 | 750 |
| 16 | Max. R.p.m. | 3249 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1625 | 1733 | 1933 | 1580 | 1500 | 1254 | 1075 | 1040 | 864 | 867 | 800 |
| 17 | Max. R.p.m. | 3150 | 2600 | 2900 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1675 | 1842 | 2058 | 1680 | 1584 | 1330 | 1145 | 1104 | 917 | 922 | 850 |
| 20 | Max. R.p.m. | 2905 | 2510 | 2725 | 1900 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 1818 | 2095 | 2277 | 1978 | 1875 | 1567 | 1347 | 1300 | 1080 | 1083 | 1000 |
| 24 | Max. R.p.m. | 2655 | 2297 | 2490 | 1824 | 1500 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 2005 | 2300 | 2486 | 2278 | 2250 | 1880 | 1612 | 1560 | 1296 | 1300 | 1200 |
| 28 | Max. R.p.m. | 2455 | 2125 | 2305 | 1686 | 1393 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 2148 | 2482 | 2694 | 2458 | 2440 | 2195 | 1880 | 1820 | 1363 | 1516 | 1400 |
| 32 | Max. R.p.m. | 2292 | 1985 | 2152 | 1578 | 1320 | 940 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 2290 | 2654 | 2871 | 2625 | 2641 | 2500 | 2150 | 2080 | 1728 | 1734 | 1600 |
| 37 | Max. R.p.m. | 2140 | 1850 | 2007 | 1469 | 1212 | 904 | 645 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 2474 | 2857 | 3010 | 2826 | 2805 | 2785 | 2485 | 2410 | 1996 | 2003 | 1850 |
| 48 | Max. R.p.m. | 1877 | 1623 | 1760 | 1289 | 1064 | 794 | 616 | 520 | 370 | 325 | 240 |
| | Ft. per min. | 2812 | 3257 | 3523 | 3218 | 3194 | 3170 | 3057 | 2995 | 2692 | 2600 | 2400 |
| 54 | Max. R.p.m. | 1770 | 1533 | 1659 | 1214 | 1002 | 748 | 581 | 503 | 370 | 325 | 240 |
| | Ft. per min. | 2990 | 3458 | 3740 | 3417 | 3380 | 3358 | 3260 | 3392 | 2915 | 2925 | 2700 |
| 62 | Max. R.p.m. | 1650 | 1428 | 1548 | 1133 | 936 | 698 | 542 | 469 | 370 | 325 | 240 |
| | Ft. per min. | 3198 | 3700 | 4010 | 3658 | 3630 | 3630 | 3398 | 3628 | 3345 | 3357 | 3100 |

unfavorable conditions, 400 ft. per min. may be too great. The critical velocity is reached with the sprocket at its maximum speed.

**STANDARD CHAIN NUMBERS.**—The right-hand digit of the chain number is zero for chains of the usual proportions, 1 for a light-weight chain, and 5 for a rollerless bushing chain. Digits to left of the right-hand digit denote the number of $^1/_8$ inches in the pitch.

The extra-heavy series is designated by the letter H following the chain number. Multiple-width chains are designated by prefixes to the chain number, as follows: D, double-width; E, triple-width; F, quadruple-width, etc.

**Tolerances.**—Chain length may be over-length $^1/_{64}$ in. per foot, but may not be under-length. Chain lengths shall be determined when subjected to a measuring load of $^1/_{32}$ of the test load (see below).

Maximum pin diameters = nominal pin diameter + 0.0005 in.

Minimum hole in bushing = 1.006 nominal pin diameter + 0.005 in.

Minimum clearance between pins and bushings = 0.006 nominal pin diameter.

Maximum width of roller link =

(nominal width of chain) + (2.05 × nominal side plate thickness) + $C$,

where $C$ = 0.005 in. for side plates up to and including 0.156 in. thick, and 0.007 in. for thicker side plates.

Minimum distance between pin plates = maximum width of roller link + 0.002 in.

**LOADS.**—**Test Load** is the load under which a chain should be tested for material defects. For single width chains

$$\text{Test load} = 42{,}000 \times (\text{pin diameter})^2 - 280 \text{ lb.}$$

**Working Load.**—Equation [2] given below is a guide for the selection of chains. Let $F$ = working load, lb.; $A_p$ = projected bearing area of pin, *i.e.* diameter of pin × length of bushing, both in inches; $V$ = chain velocity, ft. per min.; $W_f$ = weight of chain, lb. per ft. Then

$$F = 2{,}600{,}000\, A_p/(V + 600) \qquad [2]$$

When the chain velocity exceeds 800 ft. per min., the value of $F$, as found above, should be diminished by the quantity $W_f V^2/115{,}900$, which represents the centrifugal tension in the chain. The horsepower transmitted by the chain then will be

$$\text{Hp.} = FV/33{,}000 \qquad [3]$$

**LENGTH OF CHAIN.**—Let $P$ = pitch of chain, in.; $C$ = center to center distance of sprockets, pitches; $N$ = number of teeth in large sprocket, in.; $n$ = number of teeth in small sprocket, in.; $L$ = length of chain in pitches. Then

$$L = 2C + {}^1/_2\,(N + n) + \{0.0257\,(N - n)^2/C\} \qquad [4]$$

As a chain cannot contain a fractional part of a pitch, the next whole number above the calculated number of pitches must be used. If it is an odd number, an offset link should be used. The chain length in inches = $L \times P$.

If the center to center distance can be adjusted to suit the length of the chain, an even number of pitches should be chosen to avoid using an offset link. The proper center to center distance then may be found by the equation

$$C' = {}^1/_8\, P\{2L - N - n + \sqrt{(2L - N - n)^2 - 0.824\,(N - n)^2}\}, \qquad [5]$$

where $C'$ = center to center distance of sprockets, in. $C'$ should not be less than 12 $P$ nor less than the pitch diameter of the large sprocket. If possible, adjustment for the center to center distance should be provided to take up excessive slack in the chain due to elongation from wear. A slack in the chain of, say, $^1/_4$ to $^1/_2$ in. drop per foot of span should always be allowed.

**SELECTION OF PROPER CHAIN AND SPROCKETS.**—From the known data, such as horsepower, r.p.m., and center to center distance, the other data can be found from the tables and equations for chain pull, chain velocity, etc., for a trial drive. Select the shortest pitch chain that will carry the load, giving preference to the standard chains. Assume the number of teeth to be used in the sprockets; calculate the chain pull $F$ and the chain velocity $V$, and examine the tables and rules to determine if everything is within safe limits; make the necessary change in chain or sprockets, and again apply the test.

**BEST CONDITIONS OF DRIVING.**—The highest efficiency is reached in a chain drive when the center to center distance can be so adjusted that there will be an integral number of pitches between the points of tangency of the chain on the pitch circles of the two sprockets. Under this condition, the angular velocity ratio between sprockets is most nearly uniform. The true pitch line of a sprocket is a polygon of as many sides as there are sprocket teeth. During the passage of each tooth, the distance of the center line of the chain from the center of the sprocket varies. If one end of the tight strand rises while the other end falls and *vice versa*, there is a tendency to produce rapid fluctuations in the angular velocity ratio between the sprockets. The smoothest action is obtained when the two ends rise and fall in unison. The larger the number of teeth in the sprocket the closer will the polygon approach a circle. This explains why sprockets with few teeth are undesirable unless the motion is so slow that the inertia of the driven wheel and its accessories has but little effect.

Where the center to center distance is not too short, the slack strand may be either above or below. For short centers it is better to have it below. Chains may be run at any angle, but they run best when nearly horizontal. Many successful vertical drives have been installed. In such cases, the slack must not be allowed to develop to a point where the chain may jump the teeth on the lower wheel.

Idler or "jockey" sprockets should be avoided if possible. If, however, it is necessary to use them, they should be as large as possible and as well-made and well-mounted as the main sprocket. A little side float on the idler pin is desirable. Idlers never should be placed against the tight strand, nor should they ever be run above the maximum speed allowed for regular driving sprockets.

When the chain elongates to $3/8$ in. per foot it should be replaced. If any part of a link needs repair the whole link should be replaced. Excessive slack with non-adjustable centers may be corrected by removing two links and inserting an offset link, thus shortening the chain by one pitch.

Sometimes the load may be divided between two single roller chains, the sprockets running side by side. In this case, the teeth of the two parallel sprockets should be opposite and not staggered. Also the two chains should be of equal length with the roller links of one opposite the roller links of the other. If removed, each chain should be replaced over exactly the same teeth as before.

Maximum service and minimum wear of sprockets and chains are obtained when the sprockets are aligned with their mid-planes coincident.

The general conditions affecting the efficiency of a chain drive are as follows: 1. Increasing the number of teeth in the sprocket. 2. Decreasing the velocity ratio represented by r.p.m. of driver ÷ r.p.m. of driven wheel. 3. Increasing the load. 4. Continuous lubrication in an oil bath. 5. Making the center to center distance such that the span of chain on the tight side is an exact multiple of the pitch. 6. Reducing the r.p.m. 7. Decreasing the pin diameter. 8. Decreasing the diameter of the sprocket bearing. 9. Shortening the pitch. 10. Reducing the weight of the chain.

**SPROCKET PROPORTIONS.**—The pitch diameter of a sprocket is given by

$$D_p = P/\sin\,(180\ \text{deg.}/T) \qquad [6]$$

where $D_p$ = pitch diam., in.; $P$ = pitch of chain, in.; $T$ = number of teeth. Dimensions of the section profile (see Fig. 2) are given by the following equations, in which $t$ = thickness of sprocket, in.; $t_p$ = thickness of inside plate, in.; $h$ = depth of chamfer, in.; $g$ = width of chamfer, in.; $r$ = radius of chamfer, in.; $Q$ = depth of shroud, in.; $W$ = width of chain, in.; $K$ = transverse pitch of multiple width sprockets, in.;

$$\text{For single-width chain, } t = 0.93\,W - 0.006 \qquad [7]$$

$$\text{For double- and triple-width chain, } t = 0.90\,W - 0.006 \qquad [8]$$

$$\text{For quadruple-width chain and over, } t = 0.88\,W - 0.006 \qquad [9]$$

$$h = 0.5\,P \qquad [10]; \qquad r = 1.063\,P \qquad [11]$$

$$g = 0.125\,P \qquad [12]; \qquad Q = 0.5\,P \qquad [13]$$

$$K = W + 4.15\,t_p + 0.003 \qquad [14]$$

Minimum outside diameter of sprocket is

$$D_o = P[0.6 + \{\cot\,(180\ \text{deg.}/T)\}] \qquad [15]$$

$$\text{Bottom diameter of sprocket} = D_p - \text{roller diameter.}$$

$$\text{Maximum hub diameter} = D_p - P.$$

Dimensions of commercial sprockets are given in Tables 6 and 7.

**Tolerances.**—The tolerances on the bottom diameter of cut sprockets is $+\ 0.000$ in., $-\ (0.003\text{ in.} + 0.001\,P\sqrt{T})$. Tolerance on sprocket thickness is $\pm\ W/50 + 0.002$ in.

The advantages of standard tooth forms are: (*a*) Distribution of the load among several teeth; (*b*) elimination of jerks when the links come into and pass out of action; (*c*) means to provide for chain elongation by allowing the successive rollers to climb the teeth and adapt themselves to their natural pitch line; (*d*) Quieter action resulting in greater efficiency and longer life; (*l*) Less tendency of the teeth to wear hook-shaped.

The rollers of a new chain of correct pitch will seat themselves at the bottom of the tooth spaces. After the chain has elongated, one roller only at any particular instant will be seated at the bottom of the tooth space. From that point the successive rollers will

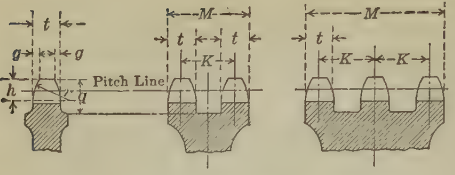


FIG. 2. Transverse Sprocket Tooth Profiles

take higher positions on the working face of the tooth until they have reached a pitch circle large enough to accommodate the increased pitch of the chain. The greatest pressure is on the first tooth, with continually lessened pressure on each successive tooth until the last or "bedding" roller is reached.

If the speed of the smaller wheel is over 60% of the maximum r.p.m. as given in Tables 3 and 4, the sprockets should be widened and all sharp burrs on the edges of the teeth removed. Smaller running sprockets may be made of machine steel, cast iron, semi-steel or bronze.

### Table 6.—Sprocket Tooth Dimensions for Roller Chain

(Morse Chain Co., Ithaca, N. Y.)

| Chain No. | Pitch $P$, in. | Nominal Roller Diam., $D_r$, in. | Sprocket Tooth Dimensions, in. (See Fig. 2) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Thickness, $t$ | | | Width Over Teeth, $M$ | | | Transverse Pitch, $K$ | Chamfer | | | |
| | | | Single | Double, Triple | Quadruple | Double | Triple | Quadruple | | Depth, $h$ | Width, $g$ | Radius, $r$ | Shroud Depth, $q$ |
| 35 | 3/8 | 0.200 | 0.168 | 0.163 | 0.159 | 0.563 | 0.963 | 1.359 | 0.400 | 3/16 | 3/64 | 0.398 | 11/32 |
| 40 | 1/2 | 5/16 | .284 | .275 | .269 | .840 | 1.405 | 1.964 | .565 | 1/4 | 1/16 | .531 | 7/16 |
| 41 | 1/2 | 0.306 | .227 | ...... | ...... | ...... | ...... | ...... | ...... | 1/4 | 1/16 | .531 | 7/16 |
| 50 | 5/8 | 0.4 | .343 | .332 | .324 | 1.042 | 1.752 | 2.454 | .710 | 5/16 | 5/64 | .664 | 9/16 |
| 60 | 3/4 | 15/32 | .459 | .444 | .434 | 1.337 | 2.230 | 3.113 | .893 | 3/8 | 3/32 | .796 | 11/16 |
| 80 | 1 | 5/8 | .575 | .557 | .544 | 1.704 | 2.851 | 3.985 | 1.147 | 1/2 | 1/8 | 1.062 | 7/8 |
| 100 | 1 1/4 | 3/4 | .692 | .669 | .654 | 2.069 | 3.469 | 4.854 | 1.400 | 5/8 | 5/32 | 1.327 | 1 1/8 |
| 120 | 1 1/2 | 7/8 | .924 | .894 | .874 | 2.763 | 4.452 | 6.211 | 1.779 | 3/4 | 3/16 | 1.593 | 1 3/8 |
| 140 | 1 3/4 | 1 | .924 | .894 | .874 | 2.806 | 4.718 | 6.610 | 1.912 | 7/8 | 7/32 | 1.858 | 1 5/8 |
| 160 | 2 | 1 1/8 | 1.156 | 1.119 | 1.094 | 3.409 | 5.699 | 7.964 | 2.290 | 1 | 1/4 | 2.124 | 1 3/4 |

### Table 7.—Standard Sprocket Diameters for 1-in. Pitch Roller Chains

(A.S.A. Standard B29a–1930).

For other pitches than 1 in., multiply the values in the table by the pitch.

To obtain bottom diameter of sprocket, subtract diameter of chain roller from pitch diameter.

| No. of Teeth | Pitch Diam., in. | Outside Diam., in. | No. of Teeth | Pitch Diam., in. | Outside Diam., in. | No. of Teeth | Pitch Diam., in. | Outside Diam., in. | No. of Teeth | Pitch Diam., in. | Outside Diam., in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 2.0000 | 2.332 | 36 | 11.4737 | 12.030 | 66 | 21.0164 | 21.593 | 96 | 30.5628 | 31.146 |
| 7 | 2.3048 | 2.677 | 37 | 11.7917 | 12.349 | 67 | 21.3346 | 21.911 | 97 | 30.8811 | 31.465 |
| 8 | 2.6131 | 3.014 | 38 | 12.1096 | 12.668 | 68 | 21.6528 | 22.230 | 98 | 31.1994 | 31.783 |
| 9 | 2.9238 | 3.347 | 39 | 12.4275 | 12.987 | 69 | 21.9710 | 22.548 | 99 | 31.5177 | 32.102 |
| 10 | 3.2361 | 3.678 | 40 | 12.7455 | 13.306 | 70 | 22.2892 | 22.867 | 100 | 31.8360 | 32.420 |
| 11 | 3.5495 | 4.006 | 41 | 13.0635 | 13.625 | 71 | 22.6074 | 23.185 | 101 | 32.1543 | 32.739 |
| 12 | 3.8637 | 4.332 | 42 | 13.3815 | 13.944 | 72 | 22.9256 | 23.504 | 102 | 32.4726 | 33.057 |
| 13 | 4.1785 | 4.657 | 43 | 13.6995 | 14.263 | 73 | 23.2438 | 23.822 | 103 | 32.7909 | 33.376 |
| 14 | 4.4940 | 4.982 | 44 | 14.0175 | 14.582 | 74 | 23.5620 | 24.141 | 104 | 33.1091 | 33.694 |
| 15 | 4.8097 | 5.305 | 45 | 14.3356 | 14.901 | 75 | 23.8802 | 24.459 | 105 | 33.4274 | 34.012 |
| 16 | 5.1259 | 5.627 | 46 | 14.6536 | 15.219 | 76 | 24.1984 | 24.778 | 106 | 33.7457 | 34.313 |
| 17 | 5.4423 | 5.950 | 47 | 14.9717 | 15.538 | 77 | 24.5166 | 25.096 | 107 | 34.0640 | 34.649 |
| 18 | 5.7588 | 6.271 | 48 | 15.2898 | 15.857 | 78 | 24.8349 | 25.415 | 108 | 34.3823 | 34.968 |
| 19 | 6.0756 | 6.593 | 49 | 15.6079 | 16.176 | 79 | 25.1531 | 25.733 | 109 | 34.7006 | 35.286 |
| 20 | 6.3925 | 6.914 | 50 | 15.9260 | 16.495 | 80 | 25.4713 | 26.052 | 110 | 35.0189 | 35.605 |
| 21 | 6.7095 | 7.235 | 51 | 16.2441 | 16.813 | 81 | 25.7895 | 26.370 | 111 | 35.3371 | 35.923 |
| 22 | 7.0266 | 7.555 | 52 | 16.5619 | 17.132 | 82 | 26.1078 | 26.689 | 112 | 35.655 | 36.241 |
| 23 | 7.3439 | 7.875 | 53 | 16.8803 | 17.451 | 83 | 26.4260 | 27.007 | 113 | 35.974 | 36.560 |
| 24 | 7.6613 | 8.196 | 54 | 17.1984 | 17.769 | 84 | 26.7443 | 27.326 | 114 | 36.292 | 36.878 |
| 25 | 7.9787 | 8.516 | 55 | 17.5166 | 18.088 | 85 | 27.0625 | 27.644 | 115 | 36.610 | 37.197 |
| 26 | 8.2962 | 8.836 | 56 | 17.8347 | 18.406 | 86 | 27.3807 | 27.962 | 116 | 36.929 | 37.515 |
| 27 | 8.6138 | 9.156 | 57 | 18.1529 | 18.725 | 87 | 27.6989 | 28.281 | 117 | 37.247 | 37.833 |
| 28 | 8.9315 | 9.475 | 58 | 18.4710 | 19.044 | 88 | 28.0171 | 28.599 | 118 | 37.565 | 38.152 |
| 29 | 9.2491 | 9.795 | 59 | 18.7892 | 19.363 | 89 | 28.3354 | 28.918 | 119 | 37.883 | 38.470 |
| 30 | 9.5668 | 10.114 | 60 | 19.1073 | 19.681 | 90 | 28.6536 | 29.236 | 120 | 38.201 | 38.788 |
| 31 | 9.8845 | 10.434 | 61 | 19.4255 | 20.000 | 91 | 28.9718 | 29.554 | 121 | 38.519 | 39.106 |
| 32 | 10.2023 | 10.753 | 62 | 19.7437 | 20.318 | 92 | 29.2900 | 29.873 | 122 | 38.837 | 39.425 |
| 33 | 10.5201 | 11.072 | 63 | 20.0618 | 20.637 | 93 | 29.6082 | 30.191 | 123 | 39.156 | 39.743 |
| 34 | 10.8380 | 11.392 | 64 | 20.3800 | 20.955 | 94 | 29.9264 | 30.510 | 124 | 39.475 | 40.062 |
| 35 | 11.1558 | 11.711 | 65 | 20.6982 | 21.274 | 95 | 30.2446 | 30.828 | 125 | 39.794 | 40.381 |

**NUMBER OF TEETH.**—The number of teeth in the sprockets should not be less than 14, unless the speed is very low. However, when a sprocket runs at a low speed, 8, 7 or even 6 teeth may be used if it is not practicable to use a higher number. At moderately high speeds, such low numbers of teeth are inefficient and destructive to the chain.

Whether the number of teeth should be odd or even is a disputed question. If a chain, after cleaning or repairs, is replaced so that the same links mesh with the same teeth as before and further, if there is an even number of pitches in the chain, *i.e.*, no offset links, an even number of teeth is advantageous. In that case one-half of the teeth will so wear as to work best with inside links which do not elongate from wear; the other half will accommodate themselves to the increasing length of the pin links. Otherwise an odd number of teeth is preferable, as then every tooth will mesh alternately with both kinds of links in the chain.

**CUTTERS FOR SPROCKET TEETH.**—Three types of cutters are used for cutting sprocket teeth, *viz.*: Convex cutters, straddle cutters and hobs. Convex cutters will cut the following ranges of teeth: 7–8, 9–11; 12–17; 18–34; 35 and over. These cutters are of the common type, designed to cut the spaces between the teeth. Straddle cutters are designed to straddle the teeth and cut a part of two spaces at the same time. Two straddle cutters will fill the requirements for all numbers of teeth. Hobs are made to cut any required number of teeth of a given pitch and roller diameter. Proportions of cutters are given in American Standard A.S.A. B29a–1930, together with proportions of pattern teeth for cast sprockets.

## 2. SILENT CHAINS

A "silent" or high-speed chain consists of a series of leaves of the shape shown in Fig. 3, which are assembled by means of pins and bushings to form various widths of chain, as shown in Fig. 4. The chain engages with the sprocket as shown in Fig. 5. As the pitch lengthens, due to wear, the links adjust themselves to a larger pitch circle as shown in Fig. 4, thus automatically compensating for wear. These chains can be run in either direction at velocities up to 1500 ft. per min. without noise or shock.

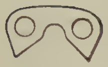


FIG. 3

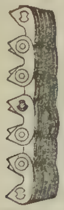


FIG. 4

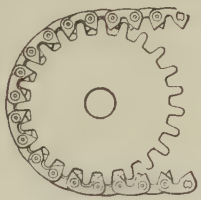


FIG. 5

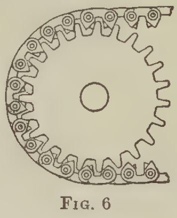


FIG. 6

Silent Chain

**ALLOWABLE TENSION.**—The values of normal tension given in Table 8 may be increased when the speed of the chain is low, and must be decreased with high speeds. The following figures show the percentage variation from normal tension with variation in speed:

| | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Percent of normal r.p.m.. | 10 | 20 | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 | 110 | 120 | 130 | 140 | 150 |
| Percent of normal tension | 300 | 300 | 235 | 200 | 165 | 140 | 125 | 115 | 105 | 100 | 85 | 68 | 54 | 40 | 25 |

EXAMPLE.—Required the safe tension on a 1-in. pitch chain, 3 in. wide, at 1200 r.p.m. and at 600 r.p.m. From Table 2, the normal tension is 3 × 205 = 615 lb. at 1000 r.p.m. At 1200 r.p.m., the speed is 1200/1000 = 120% of normal. Hence the permissible tension is 0.68 × 615 = 417.5 lb. At 600 r.p.m., the speed is 600/1000 = 60% of normal, and the allowable tension is 1.40 × 615 lb. = 861 lb.

Other factors which tend to reduce the allowable tension are continuous operation (24 hr. per day), reversing service, drives subject to shock, and drives in which the drive sprocket has less than 25 teeth. Manufacturers of silent chain should be consulted concerning drives involving unusual conditions.

**Length of Chain.**—If $P$ = pitch of chain, in., $T$ = number of teeth on larger sprocket, $t$ = number of teeth on smaller sprocket, $D$ and $d$ = pitch diameters of larger and smaller sprockets, in., respectively; $C$ = center to center distance between the sprockets, in., and $L$ = length of chain, and $\beta$ = angle whose sine is $\{{}^1/_2(D - d)/C\}$,

$$L = TP\{(180 + 2\,\beta)/360\} + tP\{(180 - 2\,\beta)/360\} + 2\,C\cos\beta\,. \quad . \quad . \quad [16]$$

**DESIGN OF CHAIN DRIVES.**—Pitch of chain first should be selected, on the basis of the values given in Table 8. The higher shaft speed is the limiting factor, and indicates

### Table 8.—Silent Chain Drive Data

(Morse Chain Co., Ithaca, N. Y.)

| Pitch, in. | Pitches per foot | Chain Width, in. | Weight per foot, 1 in. Wide, lb.* | Small Sprocket DriveR | | | | | | Small Sprocket DriveN, Minimum No of Teeth |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Normal r.p.m.† | Normal Tension, 1 in. Wide, lb.† | Minimum No. of Teeth. DriveR | Desirable No. of Teeth, DriveR | Maximum Desirable No. of Teeth, DriveN | Minimum Desirable Center Distance, in. | |
| 3/8 | 32 | 3/8–2 1/2 | 0.75 | 4000 | 75 | 13 | 15–17 | 75 | 6 | 13 |
| 1/2 | 24 | 1/2–4 | 1.00 | 2400 | 100 | 13 | 15–17 | 99 | 8 | 17 |
| 5/8 | 19.2 | 3/4–5 | 1.25 | 1800 | 125 | 13 | 17–21 | 109 | 14 | 17 |
| 3/4 | 16 | 1–7 | 1.50 | 1400 | 150 | 13 | 17–21 | 115 | 18 | 21 |
| 0.9 | 13.33 | 1 1/2–10 | 1.80 | 1100 | 185 | 15 | 17–23 | 125 | 21 | 23 |
| 1 | 12 | 1 1/2–12 | 2.00 | 1000 | 205 | 15 | 17–23 | 125 | 24 | 23 |
| 1.2 | 10 | 2–16 | 2.40 | 800 | 250 | 15 | 17–23 | 129 | 30 | 23 |
| 1.5 | 8 | 2–18 | 3.00 | 600 | 335 | 17 | 17–27 | 129 | 36 | 23 |
| 2 | 6 | 2–20 | 4.00 | 400 | 560 | 17 | 17–31 | 129 | 48 | 25 |
| 3 | 4 | 3–30 | 6.00 | 250 | 935 | 17 | 17–31 | 131 | 66 | 27 |

* Weight per foot = (2 × pitch)/width.

† Under favorable conditions these values may be exceeded, and under unfavorable conditions they must be decreased. See Allowable Tension, p. 24–45.

### Table 9.—Dimensions of Silent Chains 1 in. Wide

(Morse Chain Co., Ithaca, N. Y.)

| Pitch, in. | No. of Links per Inch Wide * | Width Overall, in. | Width Over Washers, in. | Width Over Links, in. | Length of Pins between Shoulders, in. | Length of Rocker, in. | Thickness of Links, in. |
|---|---|---|---|---|---|---|---|
| 3/8 | 24 | 1.099 | 1.039 | 0.959 | 0.959 | 0.954 | 0.040 |
| 1/2 | 16 | 1.130 | 1.030 | .940 | .940 | .925 | .060 |
| 5/8 | 12 | 1.139 | 1.039 | .919 | .919 | .904 | .081 |
| 3/4 | 12 | 1.150 | 1.040 | .920 | .920 | .904 | .081 |
| 0.9 | 8 | 1.118 | 0.998 | .878 | .878 | .863 | .122 |
| 1.0 | 8 | 1.118 | 0.998 | .878 | .878 | .863 | .122 |
| 1.2 | 8 | 1.158 | 1.038 | .878 | .878 | .848 | .122 |
| 1.5 | 8 | 1.184 | 1.038 | .878 | .878 | .848 | .122 |
| 2 | 6 | 1.227 | 1.037 | .837 | .837 | .807 | .163 |
| 3 | 3† | 1.250 | 1.000 | .674 | .674 | .644 | .163 |

* For each inch or half-inch increase in width of chain, add the increased width to dimensions given above. The first inch in width lacks one link. For example, 5/8 in. pitch, 1 in. wide = 11 links, 5 and 6 in alternate sections. For each additional inch or fraction, add as per second column of table. † Double.

the largest pitch chain that can be used. In general, the largest pitch gives the best drive, but limitations of sprocket diameters and necessity for quiet operation often dictate smaller pitches. For the most noiseless drive, the smallest pitch that will carry the load should be chosen. Large pitch chain should be used for slow speeds and large power.

**The Speed Ratio** is the basis of selection of driving and driven sprockets. The smaller sprocket usually is selected first, on the following basis:

| | | | |
|---|---|---|---|
| Speed ratio .......... | 3 : 1 to 5 : 1 | 5 : 1 to 6 : 1 | Over 6 : 1* |
| No. of teeth in pinion..... | 17, 19, 21 | 17 | 15 |

**Chain Speed** is determined by the equation

$$V = T \times \text{r.p.m.}/N \qquad [17]$$

where $V$ = chain velocity, ft. per min.; $T$ = number of teeth in sprocket; $N$ = number of links per foot of chain.

**Chain Width** is determined by first finding the total pull in the chain necessary to transmit the desired horsepower, and then ascertaining the width by means of the allowable tension for the chain used.

$$P = \text{Hp.} \times 33{,}000/V \qquad [18]$$

$$w = P/p \qquad [19]$$

where $P$ = total pull on chain, lb.; $p$ = pull on chain, lb. per inch of width (see Table 8);

---

* For chain pitches 1.2 in. and over, 17 teeth is the minimum. The number of teeth in the large sprocket should not exceed 130, and better results follow from a smaller number of teeth, especially with the smaller pitch chains. An odd number of teeth in both sprockets is advisable. If the speed ratio demands an even number, a hunting link should be used in the chain.

$V$ = chain velocity, ft. per min. The next greater width, in even inches or half-inches, as shown by Table 9, should be used.

**HORSEPOWER TRANSMITTED BY SILENT CHAINS** is given in Table 10.

**SILENT CHAIN SPROCKETS.**—Table 11 gives the weights of solid pinions and armed sprockets for silent chain. Hub lengths are given in Tables 12 and 13. Odd

**Table 10.—Horsepower Rating of Silent Chain Drives**

(Link-Belt Co., Chicago)

| Chain | | Driving Pinion | | Horsepower | |
|---|---|---|---|---|---|
| Pitch, in. | Width, in. | Pitch Diam., in. | No. of Teeth | 10 Hr. per Day | 24 Hr. per Day |
| **1750 r.p.m.** | | | | | |
| 3/8 | 7/16 | 2.04 | 17 | 1.3 | 1.0 |
| 3/8 | 3/4 | 2.04 | 17 | 2.2 | 1.8 |
| 1/2 | 3/4 | 2.72 | 17 | 3.9 | 3.1 |
| 1/2 | 1 | 2.72 | 17 | 5.2 | 4.2 |
| 1/2 | 1 1/2 | 2.72 | 17 | 7.7 | 6.2 |
| 1/2 | 2 | 2.72 | 17 | 10.3 | 8.2 |
| 1/2 | 3 | 3.04 | 19 | 17.7 | 14.2 |
| 1/2 | 3 | 3.67 | 23 | 22.2 | 17.8 |
| **1160 r.p.m.** | | | | | |
| 3/8 | 9/16 | 2.04 | 17 | 1.2 | 1.0 |
| 3/8 | 1 | 2.04 | 17 | 2.1 | 1.7 |
| 1/2 | 1 | 2.72 | 17 | 3.9 | 3.1 |
| 1/2 | 1 1/2 | 2.72 | 17 | 5.9 | 4.7 |
| 1/2 | 2 | 3.04 | 19 | 8.9 | 7.1 |
| 1/2 | 3 | 2.72 | 17 | 11.8 | 9.4 |
| 3/4 | 2 | 4.58 | 19 | 17.6 | 14.1 |
| 3/4 | 2 1/2 | 5.05 | 21 | 24.7 | 19.8 |
| 3/4 | 3 | 5.05 | 21 | 29.7 | 23.8 |
| 3/4 | 4 | 5.05 | 21 | 39.6 | 31.7 |
| 3/4 | 5 | 5.05 | 21 | 49.5 | 39.6 |
| 3/4 | 6 | 5.05 | 21 | 59.4 | 47.5 |
| 3/4 | 7 | 5.05 | 21 | 69.3 | 55.5 |
| 3/4 | 8 | 5.05 | 21 | 79.2 | 63.3 |
| 3/4 | 8 | 5.53 | 23 | 88.0 | 70.4 |
| 3/4 | 9 | 5.53 | 23 | 99.1 | 79.2 |
| **860 r.p.m.** | | | | | |
| 3/8 | 3/4 | 2.04 | 17 | 1.3 | 1.0 |
| 1/2 | 3/4 | 2.72 | 17 | 2.4 | 1.9 |
| 1/2 | 1 | 2.72 | 17 | 3.2 | 2.6 |
| 1/2 | 2 | 2.72 | 17 | 6.3 | 5.0 |
| 1/2 | 2 | 3.36 | 21 | 8.0 | 6.4 |
| 1/2 | 2 1/2 | 3.36 | 21 | 10.0 | 8.0 |
| 3/4 | 2 | 5.05 | 21 | 16.2 | 13.0 |
| 3/4 | 2 1/2 | 5.05 | 21 | 20.3 | 16.2 |
| 3/4 | 4 | 5.05 | 21 | 32.4 | 25.9 |
| 3/4 | 5 | 5.05 | 21 | 40.5 | 32.4 |
| 1 | 5 | 6.10 | 19 | 54.5 | 43.6 |
| 1 | 6 | 6.10 | 19 | 65.4 | 52.3 |
| 1 | 7 | 6.10 | 19 | 76.2 | 61.0 |
| 1 | 7 | 6.74 | 21 | 85.7 | 68.5 |
| 1 | 8 | 6.74 | 21 | 98.0 | 78.4 |
| 1 | 9 | 6.74 | 21 | 110.2 | 88.2 |
| 1 | 12 | 6.74 | 21 | 147.0 | 117.6 |
| 1 1/4 | 10 | 7.62 | 19 | 156.2 | 125.0 |
| 1 1/4 | 12 | 7.62 | 19 | 187.5 | 150.0 |
| **690 r.p.m.** | | | | | |
| 1 | 2 | 6.74 | 21 | 21.3 | 17.0 |
| 1 | 3 | 6.10 | 19 | 28.4 | 22.7 |
| 1 | 3 | 6.74 | 21 | 31.9 | 25.5 |
| 1 | 4 | 6.74 | 21 | 42.6 | 34.1 |
| 1 1/4 | 4 | 8.41 | 21 | 62.6 | 50.1 |
| 1 1/4 | 5 | 8.41 | 21 | 78.2 | 62.5 |
| 1 1/4 | 6 | 7.62 | 19 | 82.0 | 65.6 |
| 1 1/4 | 7 | 7.62 | 19 | 95.7 | 76.5 |
| **690 r.p.m.** (*Continued*) | | | | | |
| 1 1/4 | 7 | 8.41 | 21 | 109.5 | 87.6 |
| 1 1/4 | 8 | 8.41 | 21 | 125.2 | 100.2 |
| 1 1/2 | 9 | 9.05 | 19 | 165.0 | 132.0 |
| 1 1/2 | 10 | 9.05 | 19 | 183.4 | 147.0 |
| 1 1/2 | 14 | 8.41 | 21 | 219.0 | 175.0 |
| 1 1/4 | 16 | 8.41 | 21 | 250.0 | 200.0 |
| 1 1/2 | 16 | 10.01 | 21 | 335.0 | 268.0 |
| 1 1/2 | 18 | 10.01 | 21 | 376.0 | 301.9 |
| **585 r.p.m.** | | | | | |
| 1 1/2 | 12 | 10.01 | 21 | 228.5 | 183.0 |
| 1 1/2 | 14 | 10.01 | 21 | 267.0 | 213.0 |
| 1 1/2 | 16 | 10.01 | 21 | 305.0 | 244.0 |
| 1 1/2 | 20 | 10.01 | 21 | 381.0 | 305.0 |
| 1 1/2 | 22 | 10.01 | 21 | 418.0 | 335.0 |
| 1 1/2 | 24 | 10.01 | 21 | 457.0 | 366.0 |
| 1 1/2 | 24 | 10.96 | 23 | 505.0 | 404.0 |
| 1 1/2 | 24 | 11.92 | 25 | 547.0 | 437.0 |
| **570 r.p.m.** | | | | | |
| 1 1/4 | 5 | 7.62 | 19 | 60.7 | 48.5 |
| 1 1/4 | 6 | 7.62 | 19 | 72.8 | 58.2 |
| 1 1/2 | 5 | 10.01 | 21 | 93.6 | 74.9 |
| 1 1/2 | 7 | 9.05 | 19 | 114.5 | 91.6 |
| 1 1/2 | 8 | 9.05 | 19 | 130.8 | 104.7 |
| 1 1/2 | 9 | 10.01 | 21 | 168.2 | 134.5 |
| 1 1/2 | 10 | 10.01 | 21 | 187.1 | 149.7 |
| **450 r.p.m.** | | | | | |
| 2 | 22 | 14.70 | 23 | 627.0 | 501.0 |
| 2 | 24 | 14.70 | 23 | 683.0 | 546.0 |
| 2 | 26 | 14.70 | 23 | 741.0 | 593.0 |
| 2 | 28 | 14.70 | 23 | 797.0 | 637.0 |
| 2 | 30 | 14.70 | 23 | 855.0 | 684.0 |
| 2 | 32 | 14.70 | 23 | 912.0 | 730.0 |
| **440 r.p.m.** | | | | | |
| 2 | 8 | 13.43 | 21 | 201.5 | 161.0 |
| 2 | 10 | 13.43 | 21 | 252.0 | 201.0 |
| 2 | 16 | 13.43 | 21 | 403.0 | 323.0 |
| 2 | 18 | 13.43 | 21 | 453.0 | 362.0 |
| 2 | 18 | 14.70 | 23 | 505.0 | 404.0 |
| 2 | 20 | 14.70 | 23 | 561.0 | 449.0 |
| **436 r.p.m.** | | | | | |
| 2 | 12 | 13.43 | 21 | 301.0 | 241.0 |
| 2 | 14 | 13.43 | 21 | 351.0 | 281.0 |
| **350 r.p.m.** | | | | | |
| 2 | 20 | 13.43 | 21 | 433.0 | 347.0 |
| 2 | 22 | 13.43 | 21 | 476.0 | 381.0 |
| 2 | 22 | 14.70 | 23 | 528.0 | 423.0 |
| 2 | 24 | 14.70 | 23 | 576.0 | 461.0 |
| 2 | 26 | 14.70 | 23 | 624.0 | 599.0 |
| 2 | 28 | 14.70 | 23 | 672.0 | 537.0 |
| 2 | 30 | 14.70 | 23 | 720.0 | 576.0 |
| 2 1/2 | 20 | 19.95 | 25 | 752.0 | 601.0 |
| 2 1/2 | 24 | 19.95 | 25 | 902.0 | 722.0 |
| 2 1/2 | 28 | 19.95 | 25 | 1053.0 | 843.0 |
| 2 1/2* | 28* | 19.95 | 25 | 2030.0* | 1625.0* |

* 2 Strands.

widths of chain take the hub projection for the next higher width. For cast sprockets the values in Table 11 are the length of hub on each side of the sprocket. Split sprockets should have longer hubs than solid sprockets. Spring sprockets should be used with pulsating loads, as an air compressor, and shearing pin sprockets for drives subject to shock.

**Table 11.—Weight per Inch of Face of Silent Chain Sprockets**

(Compiled from curves issued by Morse Chain Co., Ithaca, N. Y.)

| Pitch, in. | No. of Teeth, Solid Pinion | | | | | No. of Teeth, Sprockets | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 13 | 15 | 20 | 25 | 30 | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 | 110 | 120 | 130 |
| | Weight, lb. | | | | | | | | | | | | | | | |
| 3/8 | .... | .... | 1.2 | 1.9 | 2.65 | 3.0 | 4.0 | 5.0 | 6.0 | 7.0 | 8.0 | 9.0 | ... | ..... | .... | ..... |
| 1/2 | .... | 1.0 | 1.75 | 2.75 | 4.0 | 4.8 | 6.4 | 8.0 | 9.6 | 11.2 | 12.8 | 14.4 | 16 | 17.6 | 19 | 20.4 |
| 5/8 | 1.2 | 1.4 | 2.5 | 3.7 | 5.3 | 7.5 | 10 | 12.5 | 15 | 17.5 | 20 | 22.5 | 25 | 27.5 | 30 | 32.5 |
| 3/4 | 1.5 | 2.0 | 3.5 | 5.2 | 8.0 | 10.5 | 14 | 17.5 | 21 | 24.5 | 28 | 31.5 | 35 | 38.5 | 42 | 45.5 |
| 0.9 | 2.2 | 3.0 | 5.0 | 8.0 | 10.2 | 13.5 | 18 | 22.5 | 27 | 31.5 | 36 | 40.5 | 45 | 49.5 | 54 | 58.5 |
| 1.0 | 2.8 | 3.6 | 6.2 | 10.0 | 15.5 | 18 | 24 | 30 | 36 | 42 | 48 | 54 | 60 | 66 | 72 | 78 |
| 1.2 | 3.8 | 5.0 | 9.0 | 14.0 | ..... | 21 | 28 | 35 | 42 | 49 | 56 | 63 | 70 | 77 | 84 | 91 |
| 1.5 | 5.8 | 7.8 | 14.0 | 21.0 | ..... | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 | 110 | 120 | 130 |
| 2.0 | 9.8 | 13.8 | 22.5 | 35.0 | ..... | 60 | 80 | 100 | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 260 |
| 3.0 | 24.0 | 31.0 | 33.0 | ..... | ..... | 102 | 145 | 188 | 231 | 274 | 317 | 360 | 403 | 446 | 489 | 522 |

**Table 12.—Hub Lengths of Solid Steel Silent Chain Pinions**

(Morse Chain Co., Ithaca, N. Y.)

| Pitch of Chain, in. | Chain Width, in. | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 1 1/2 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
| | Hub Length for Pinions not over 10 in. Diameter, in. | | | | | | | | | | | | | | | | | | |
| 3/8 | 1/2 | 3/4 | 3/4 | 3/4 | 3/4 | .... | .... | .... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 1/2 | 1/2 | 3/4 | 3/4 | 3/4 | 3/4 | .... | .... | .... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 5/8 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | .... | .... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4* | 3/4* | 3/4* | 3/4* | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 0.9 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4* | 3/4* | 3/4* | 3/4* | 1* | 1* | 1* | 1* | 1* | 1* | 1* |
| 1.0 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 | 3/4* | 3/4* | 3/4* | 3/4* | 1* | 1* | 1* | 1* | 1* | 1* | 1* |
| 1.2 | ... | .... | 1 | 1 | 1 | 1 | 1 | 1 | 1* | 1* | 1* | 1* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* |
| 1.5 | ... | .... | 1 | 1 | 1 | 1 | 1 | 1 | 1* | 1* | 1* | 1* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* |
| 2 | ... | .... | 1 | 1 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* |
| 3 | ... | .... | .... | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2 | 1 1/2 | 2 | 2 | 2 | 2 | 2 | 2* | 2* | 2* | 2* | 2* | 2* |
| | Hub Lengths for Pinions over 10 in. Diameter, in. | | | | | | | | | | | | | | | | | | |
| 1.2 | ... | .... | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* |
| 1.5 | ... | .... | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* | 1 1/4* |
| 2 | ... | .... | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/4 | 1 1/2 | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* | 1 1/2* |
| 3 | ... | .... | .... | 1 1/2 | 1 1/2 | 1 1/2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2* | 2* | 2* | 2* | 2* | 2* |

* Hub at each end of pinion.

# ROPE DRIVING

## 1. MANILA AND COTTON ROPE DRIVES

**SYSTEMS OF ROPE DRIVING.**—Two systems of rope driving are in use: 1. The American or continuous system, Fig. 1, in which a single rope passes over the sheaves several times, the slack being taken up by a tension carriage, which also transfers the rope from the outside groove of the driven sheave to the opposite outside groove of the driving sheave. 2. The English multiple system, comprising one or more separate ropes alongside of each other in the pulley sheaves.

**The Continuous System** is adapted to drives where the shafts are not parallel, to vertical drives and to drives where the rope is exposed to the weather. A disadvantage, however, is that the breakage of a single strand puts the entire drive out of service.

The arrangement of the tension carriage is important. The slack preferably should

**Table 13.—Standard Hub Lengths for Cast-iron Silent Chain Sprockets**

Table shows hub length on each side of sprocket

(Morse Chain Co., Ithaca, N. Y.)

| Outside Diam., in. | Chain Width, in. | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 1/4 | 1 1/2 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 12 |
| 13.5 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 16.5 | 3/4 | ....... | ....... | ....... | ...Standard hub length to right of and above heavy | | | | | | | |
| 19.5 | 1 | 3/4 | ....... | ....... | ......line, 1/2 in. | | | | | | | |
| 22.5 | 1 1/4 | 1 | 3/4 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 25.5 | 1 1/2 | 1 1/4 | 1 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 28.5 | 1 3/4 | 1 1/2 | 1 1/4 | 3/4 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 31.5 | 2 | 1 3/4 | 1 1/2 | 1 | ....... | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 34.5 | 2 1/4 | 2 | 1 3/4 | 1 1/4 | 3/4 | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 37.5 | 2 1/2 | 2 1/4 | 2 | 1 1/2 | 1 | ....... | ....... | ....... | ....... | ....... | ....... | ....... |
| 40.5 | 2 3/4 | 2 1/2 | 2 1/4 | 1 3/4 | 1 1/4 | 3/4 | ....... | ....... | ....... | ....... | ....... | ....... |
| 43.5 | 3 | 2 3/4 | 2 1/2 | 2 | 1 1/2 | 1 | ....... | ....... | ....... | ....... | ....... | ....... |
| 46.5 | 3 1/4 | 3 | 2 3/4 | 2 1/4 | 1 3/4 | 1 1/4 | 3/4 | ....... | ....... | ....... | ....... | ....... |
| 49.5 | 3 1/2 | 3 1/4 | 3 | 2 1/2 | 2 | 1 1/2 | 1 | ....... | ....... | ....... | ....... | ....... |
| 52.5 | 3 3/4 | 3 1/2 | 3 1/4 | 2 3/4 | 2 1/4 | 1 3/4 | 1 1/4 | 3/4 | ....... | ....... | ....... | ....... |
| 55.5 | 4 | 3 3/4 | 3 1/2 | 3 | 2 1/2 | 2 | 1 1/2 | 1 | ....... | ....... | ....... | ....... |
| 58.5 | 4 1/4 | 4 | 3 3/4 | 3 1/4 | 2 3/4 | 2 1/4 | 1 3/4 | 1 1/4 | 3/4 | ....... | ....... | ....... |
| 61.5 | 4 1/2 | 4 1/4 | 4 | 3 1/2 | 3 | 2 1/2 | 2 | 1 1/2 | 1 | ....... | ....... | ....... |
| 64.5 | 4 3/4 | 4 1/2 | 4 1/4 | 3 3/4 | 3 1/4 | 2 3/4 | 2 1/4 | 1 3/4 | 1 1/4 | 3/4 | ....... | ....... |
| 67.5 | 5 | 4 3/4 | 4 1/2 | 4 | 3 1/2 | 3 | 2 1/2 | 2 | 1 1/2 | 1 | ....... | ....... |
| 70.5 | ....... | 5 | 4 3/4 | 4 1/4 | 3 3/4 | 3 1/4 | 2 3/4 | 2 1/4 | 1 3/4 | 1 1/4 | 3/4 | ....... |
| 73.5 | ....... | ....... | 5 | 4 1/2 | 4 | 3 1/2 | 3 | 2 1/2 | 2 | 1 1/2 | 1 | ....... |
| 76.5 | ....... | ....... | ....... | 4 3/4 | 4 1/4 | 3 3/4 | 3 1/4 | 2 3/4 | 2 1/4 | 1 3/4 | 1 1/4 | ....... |
| 79.5 | ....... | ....... | ....... | 5 | 4 1/2 | 4 | 3 1/2 | 3 | 2 1/2 | 2 | 1 1/2 | ....... |
| 82.5 | ....... | ....... | ....... | ....... | 4 3/4 | 4 1/4 | 3 3/4 | 3 1/4 | 2 3/4 | 2 1/4 | 1 3/4 | 3/4 |
| 85.5 | ....... | ....... | ....... | ....... | 5 | 4 1/2 | 4 | 3 1/2 | 3 | 2 1/2 | 2 | 1 |

be taken up on the slack side of the drive at the driving sheave. See Fig. 1. If this is not possible, it may be taken up at the driven sheave. See Fig. 2. The outside groove of the driven sheave then should be a loose sheave, and the rope led to it from the tension sheave, whence it is returned to the opposite outside groove on the driving sheave. The weight applied to the tension carriage should be just sufficient to prevent any tendency to slip in the rope. It can be approximated from the value of the tension in the slack side, as calculated by formula [4].

The number of grooves and length of rope that one tension carriage will serve varies with the conditions of the drive. It is advisable to limit the length of the rope in continuous drives of large power, and use two or more ropes, each served by a separate tension carriage. The American Mfg. Co. recommends that not over 10 or 12 laps of rope be allotted to a single tension sheave, and for long drives and those requiring heavy ropes the number of laps per tension carriage should be reduced in proportion to the length and weight of rope between driving and driven sheaves. The maximum length of rope served by one tension carriage should not exceed 3000 ft. under ordinary conditions, and the travel of the carriage should be at least 2% of the length of the rope, except for very long or very short ropes. Table 1 gives data which are useful in the design of rope drives.

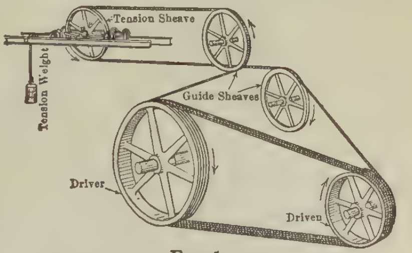


Fig. 1

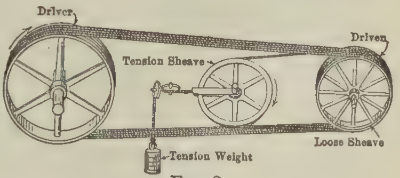


Fig. 2
Continuous Rope Drives

**The English Multiple System**, using separate parallel ropes, has no tension carriage, but depends on the weight of the ropes to provide the necessary tension. The power transmitted by a given drive may be increased at any time by the installation of additional ropes. An advantage of this system is that the breakage of a single rope does not put the drive out of service. Disadvantages are the difficulty of maintaining uniform

**Table 1.—Data of Manila Transmission Rope**

(American Mfg. Co., Brooklyn, N. Y.)

| Diam. of Rope, in. | Diameter² | Approximate Weight, lb. per ft. | Breaking Strength, lb. | Maximum Allowable Tension | Length of Splice, ft. | | | Smallest Sheave Diam., in. | Maximum r.p.m. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | 3 Strands | 4 Strands | 6 Strands | | |
| 3/4 | 0.5625 | 0.21 | 3,950 | 112 | 6 | 8 | ........ | 28 | 760 |
| 7/8 | 0.7656 | 0.27 | 5,400 | 153 | 6 | 8 | ........ | 32 | 650 |
| 1 | 1. | 0.36 | 7,000 | 200 | 7 | 10 | 14 | 36 | 570 |
| 1 1/8 | 1.2656 | 0.45 | 8,900 | 253 | 7 | 10 | 16 | 40 | 510 |
| 1 1/4 | 1.5625 | 0.56 | 10,900 | 312 | 7 | 10 | 16 | 46 | 460 |
| 1 3/8 | 1.8906 | 0.68 | 13,200 | 378 | 8 | 12 | 16 | 50 | 415 |
| 1 1/2 | 2.25 | 0.80 | 15,700 | 450 | 8 | 12 | 18 | 54 | 380 |
| 1 5/8 | 2.6406 | 0.92 | 18,500 | 528 | 8 | 12 | 18 | 60 | 344 |
| 1 3/4 | 3.0625 | 1.08 | 21,400 | 612 | 8 | 12 | 18 | 64 | 330 |
| 2 | 4. | 1.40 | 28,000 | 800 | 9 | 14 | 20 | 72 | 290 |
| 2 1/4 | 5.0625 | 1.80 | 35,400 | 1012 | 9 | 14 | 20 | 82 | 255 |
| 2 1/2 | 6.25 | 2.20 | 43,700 | 1250 | 10 | 16 | 22 | 90 | 230 |

Weight of transmission rope = 0.34 × diam.².
Breaking strength = 7000 × diam.².
Maximum allowable tension = 200 × diam.².
Diameter of smallest practicable sheave = 36 × diam.
Velocity of rope (assumed) = 5400 ft. per min.

tension on the several ropes, with resultant excessive wear on the more heavily loaded ones; the necessity of protecting the drive from the weather; the inability to use it for vertical drives; and the requirement that the distance between sheaves must be sufficiently long to give the necessary tension on the ropes.

**EFFICIENCY OF ROPE DRIVES.**—(E. H. Ahara, *Trans.* A.S.M.E., xxxv, p. 557, 1913.)—The efficiency of rope drives decreases slightly with increasing speeds. It is not affected by variation of center to center distances from 25 to 150 ft. The following figures were obtained in tests of both multiple and continuous drives at 3/4 load with a constant rope speed of 4500 ft. per min., using 1-in. manila rope with a center to center distance of pulleys of 100 ft.:

| No. of ropes in drive | | 1 | 2 | 3 | 4 | 6 | 8 |
|---|---|---|---|---|---|---|---|
| Continuous drive efficiency | Open | 91 | 93 | 94 | 94 | 94 | 94 |
| | Up and over | 58 | 72 | 79 | 80 | 80 | 81 |
| Multiple drive efficiency | Open | 73 | 80 | 84 | 86 | 86 | 86 |
| | Up and over | 56 | 67 | 71 | 72 | 75 | 77 |

The continuous drive had 60-deg. grooves, and the multiple drive 45-deg. grooves. In the open drive, the ropes ran horizontally between two sheaves. In the up and over drive the rope was led upward from the driving sheave, horizontally over two idlers, downward and around the driven sheave, upward and horizontally over two idlers, and thence downward to the driving sheave.

**HORSEPOWER OF ROPE DRIVES.**—Let $T$ = working tension in rope, lb.; $T_1$ and $T_2$ = the total tension in tight and slack sides of rope, respectively; $t_1$ and $t_2$ = effective tension in tight and slack sides of rope, respectively, lb.; $t_c$ = tension due to centrifugal force; $V$ and $v$ = velocity of rope, ft. per min. and ft. per sec., respectively; $w$ = weight of rope per ft., lb.; $d$ = diameter of rope, in.; $S$ = breaking strength of rope, lb. Then

$$T = (t_1 - t_2), \qquad [1]$$

$$t_c = wv^2/g = wv^2/32.2, \qquad [2]$$

$$\text{Hp.} = TV/33{,}000 \qquad [3]$$

Tension $t_1$ may be considered as made up of: (1) tension doing useful work; (2) tension to balance the tension necessary for adhesion; (3) tension due to centrifugal force. Tension $t_2$ may be considered as made up of: (1) tension necessary for adhesion $T_a$; (2) tension due to centrifugal force, $t_c$.

Tension on the tight side is assumed to be double that on the slack side. Then $T_a/T = 1/2$, and $T_a = 1/2\ T$. Since $T_a = t_2 - t_c$, $t_2 = t_c + 1/2\ T$, and

$$T = t_1 - t_c - 1/2\ T, \qquad [4]$$

from which

$$T = 2/3(t_1 - t_c) \qquad [5]$$

Substituting the value of $T$ from [5] in equation [3],

$$\text{Hp.} = 2/3(t_1 - t_c)V/33{,}000 \qquad [6]$$

According to the American Mfg. Co., $t_1$ should be limited to $1/35\ S$, and $S$ may be taken

**Table 2.—Horsepower Transmitted by Single Manila Ropes**

(American Mfg. Co., Brooklyn, N. Y.)

| Diameter of Rope, in. | Velocity, feet per minute | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1000 | 1500 | 2000 | 2500 | 3000 | 3500 | 4000 | 4500 | 5000 | 5500 | 6000 |
| 3/4 | 2.3 | 3.3 | 4.3 | 5.2 | 6.0 | 6.6 | 7.2 | 7.3 | 7.4 | 7.3 | 6.9 |
| 7/8 | 3.0 | 4.5 | 5.9 | 7.0 | 8.2 | 9.0 | 9.6 | 9.8 | 10.0 | 9.6 | 9.0 |
| 1 | 4.0 | 5.9 | 7.7 | 9.2 | 10.6 | 11.8 | 12.7 | 12.9 | 13.0 | 12.7 | 12.0 |
| 1 1/8 | 5.0 | 7.5 | 9.7 | 11.6 | 13.5 | 14.9 | 16.0 | 16.3 | 16.7 | 16.5 | 15.3 |
| 1 1/4 | 6.3 | 9.1 | 12.0 | 14.3 | 16.7 | 18.5 | 20.0 | 20.2 | 20.7 | 20.1 | 18.9 |
| 1 3/8 | 7.5 | 10.8 | 14.4 | 17.4 | 20.0 | 22.1 | 23.7 | 24.5 | 24.6 | 24.0 | 22.3 |
| 1 1/2 | 9.0 | 13.5 | 17.4 | 20.7 | 23.0 | 26.3 | 28.7 | 29.0 | 29.5 | 28.6 | 26.7 |
| 1 5/8 | 10.5 | 15.5 | 20.1 | 24.3 | 27.9 | 30.8 | 32.9 | 34.1 | 34.3 | 33.3 | 31.0 |
| 1 3/4 | 12.3 | 18.0 | 23.6 | 28.2 | 32.7 | 36.4 | 38.5 | 39.4 | 40.5 | 38.7 | 36.0 |
| 2 | 16.0 | 23.2 | 30.6 | 36.8 | 42.5 | 46.7 | 50.0 | 51.7 | 52.8 | 50.6 | 47.3 |
| 2 1/4 | 20.0 | 29.6 | 38.6 | 46.6 | 53.6 | 59.2 | 63.6 | 65.8 | 66.3 | 64.4 | 60.3 |
| 2 1/2 | 25.0 | 36.6 | 47.7 | 57.5 | 66.0 | 71.2 | 78.0 | 80.0 | 81.0 | 79.0 | 73.8 |

**Table 3.—Sag of Ropes in Rope Drives**

| Distance Between Pulleys, ft. | Sag on Driving Side, All Speeds, ft. | Sag on Slack Side | | | | |
|---|---|---|---|---|---|---|
| | | Velocity, feet per minute | | | | |
| | | 3000 | 4000 | 4500 | 5000 | 5500 |
| 30 | 0.19 | 0.45 | 0.39 | 0.36 | 0.33 | 0.30 |
| 40 | .34 | .80 | .69 | .64 | .59 | .53 |
| 50 | .53 | 1.2 | 1.1 | 1.0 | .92 | .84 |
| 60 | .76 | 1.8 | 1.7 | 1.4 | 1.3 | 1.2 |
| 70 | 1.0 | 2.4 | 2.1 | 1.9 | 1.7 | 1.6 |
| 80 | 1.4 | 3.2 | 2.9 | 2.5 | 2.3 | 2.1 |
| 90 | 1.7 | 4.0 | 3.5 | 3.2 | 3.0 | 2.7 |
| 100 | 2.1 | 5.0 | 4.3 | 4.0 | 3.7 | 3.3 |
| 120 | 3.0 | 7.2 | 6.2 | 5.7 | 5.3 | 4.8 |
| 140 | 4.1 | 9.9 | 8.5 | 7.8 | 7.2 | 6.6 |
| 160 | 5.4 | 12.9 | 11.1 | 10.2 | 9.5 | 8.6 |

as 7000 $d^2$; whence, $t_1 = 200\ d^2$. Substituting the values $t_1 = 200\ d^2$, and $t_c = wv^2/32.2$ in [6], we have

$$\text{Hp.} = {}^2\!/_3\ V\{200\ d^2 - (wv^2/32.2)\} \div 33{,}000$$

$$= 0.000\,022\,2\ V\{200\ d^2 - (wv^2/32.2\} \quad \ldots\ldots \quad [7]$$

Table 2 gives the horsepower that will be transmitted by single ropes at various velocities. It is based on a coefficient of friction of 0.31, corresponding to a groove angle of 45° and an arc of contact of 165°. (For theoretical discussion of the effect of arc of contact and coefficient of friction on the transmission of power by belts, bands, ropes, etc., see Belting, p. 24–14.)

Since the value of $T_c$ increases with the velocity of the rope, it is obvious that there is a certain speed at which the maximum horsepower will be transmitted. For a tension in the tight side of 200 lb. per sq. in. this is in the neighborhood of 80 ft. per sec. as shown by Fig. 3.

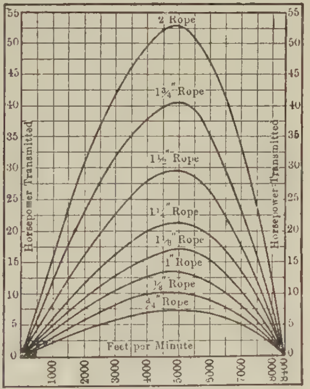


FIG. 3. Horsepower Transmitted by Manila Rope

**Sag of Rope.**—The catenary of the driving side is not affected by the speed or by the diameter of the rope. The deflection of the rope between the pulleys on the slack side varies with each change of load or speed. The deflections $D$ and $D_1$, or sag in the tight and slack sides, respectively, for horizontal drives may be approximated from equations [8] and [9], $t_1$ being taken as $200\,d^2$, and $L$ being distance between sheaves, ft.:

$$D = (wL^2)/8\,t_1 \quad \ldots\ldots \quad [8]; \qquad D_1 = (wL^2)/8\,t_2 \quad \ldots\ldots \quad [9]$$

The value of $t_2$ is found from equations [4] and [5]. Equations [8] and [9] will closely approximate the sag on the driving side for both multiple and continuous systems, and also the sag on the slack side in the continuous system. In the multiple system the equation for the slack side is true only when the ropes are first spliced on the pulleys, as the stretch in new ropes, when they are put in motion, rapidly increases the sag. Table 3 has been computed from the foregoing equation.

**SHEAVES.**—The larger the sheaves, the less the internal wear of the rope. Sheaves should not be less than 40 times the diameter of the rope for economical wear, and as much larger as it is possible to make them. This rule applies also to idler and tension sheaves.

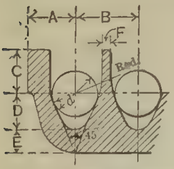


FIG. 4. Driving Sheave Grooves

Fig. 4 and Table 4 show the form and give the dimensions of the grooves in driving and driven sheaves, as recommended by The American Mfg. Co., for the continuous system. The grooves for the multiple rope system are of similar form, but deeper, and the angle of the grooves is 30 deg., instead of 45 deg. Fig. 5 shows the form of grooves recommended for carrier and idler sheaves in the continuous rope system. Grooves should be well polished to avoid wear on the ropes. They also should be free from sand- or blowholes, which tend to cut the rope rapidly.

**COTTON ROPES** are used almost exclusively in Great Britain but only to a limited extent in the U. S., where the preference is for manila rope. The American Mfg. Co. states that cotton ropes are advantageously used as bands or cords on the smaller machine appliances; the fiber, being softer and more flexible than manila hemp, gives good results for small sheaves; but for large drives, where power transmitted is in considerable amounts, cotton rope, as compared with manila, is hardly to be considered, on account of the following disadvantages: It is less durable; it is injuriously affected by the weather, so that for exposed drives, paper-mill work, or use in waterwheel pits, it is unsatisfactory; it is difficult, if not impossible, to splice uniformly; even the best quality cotton rope is much inferior to manila in strength—the breaking strain of the highest grade being but 4000 × diam.$^2$ as against 7000 × diam.$^2$ for manila; while, for the transmission of equal powers, the cost of a cotton rope varies from one-third to one-half more than manila.

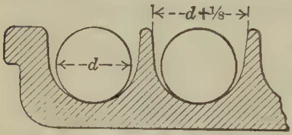


FIG. 5. Idler and Carrier Sheave Grooves

J. Melville Alison (*Jour.* Engg. Inst. of Canada, June, 1920) gives a table for the horsepower transmitted by cotton ropes, from which Table 5 is condensed. He recommends a groove angle in the sheaves of 40 deg.

For many valuable data on rope driving, including arrangement of drives, see Blue Book of Rope Transmission, American Mfg. Co., Brooklyn, N. Y.

**SHEAVE CENTER DISTANCES.**—The distance between the axes of the sheaves of rope drives, where the tight strands are on the bottom, depends on the sheave diameters. The slack strands should clear the tight lower strands by at least 0.1 $L$, where

**Table 4.—Dimensions of Grooves in Rope Sheaves**

| Diameter of Rope, in. | Dimensions in Inches. (See Fig. 4) | | | | | | |
|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | Radius |
| 2 | 2 1/8 | 2 3/4 | 2 | 1 5/8 | 1 | 3/8 | 4 9/16 |
| 1 3/4 | 1 15/16 | 2 1/2 | 1 3/4 | 1 7/16 | 7/8 | 3/8 | 4 3/8 |
| 1 1/2 | 1 11/16 | 2 1/4 | 1 1/2 | 1 1/4 | 3/4 | 3/8 | 4 5/16 |
| 1 3/8 | 1 9/16 | 2 1/8 | 1 3/8 | 1 1/8 | 11/16 | 3/8 | 4 1/8 |
| 1 1/4 | 1 1/2 | 2 1/8 | 1 1/4 | 1 | 5/8 | 3/8 | 4 |
| 1 1/8 | 1 3/8 | 2 | 1 1/8 | 15/16 | 9/16 | 3/8 | 3 7/8 |
| 1 | 1 1/4 | 1 7/8 | 1 | 13/16 | 1/2 | 3/8 | 3 7/8 |
| 7/8 | 1 1/16 | 1 1/2 | 7/8 | 3/4 | 7/16 | 1/4 | 2 15/16 |
| 3/4 | 15/16 | 1 3/8 | 3/4 | 5/8 | 3/8 | 1/4 | 2 15/16 |

**Table 5.—Horsepower Transmitted by 3-Strand Cotton Rope**

| Diameter of Rope, in. | Velocity, feet per minute | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1000 | 1500 | 2000 | 2500 | 3000 | 3500 | 4000 | 4500 | 5000 | 5500 | 6000 |
| 1/2 | 1.0 | 1.6 | 2.1 | 2.6 | 3.1 | ...... | ...... | ...... | ...... | ...... | ...... |
| 5/8 | 1.6 | 2.6 | 3.6 | 4.2 | 4.7 | 5.2 | ...... | ...... | ...... | ...... | ...... |
| 3/4 | 2.3 | 3.8 | 5.0 | 6.0 | 7.0 | 7.6 | 8.1 | ...... | ...... | ...... | ...... |
| 7/8 | 3.2 | 5.2 | 6.7 | 8.2 | 9.6 | 10.6 | 11.3 | 11.8 | 12.3 | ...... | ...... |
| 1 | 4.2 | 6.7 | 8.7 | 10.7 | 12.3 | 13.6 | 14.6 | 15.6 | 16.4 | 16.9 | ...... |
| 1 1/8 | 5.3 | 8.3 | 11.2 | 13.7 | 15.7 | 17.2 | 18.5 | 19.5 | 20.5 | 21.5 | 22.3 |
| 1 1/4 | 6.6 | 10.2 | 13.7 | 16.9 | 19.3 | 21.3 | 22.9 | 24.4 | 25.5 | 26.5 | 27.5 |
| 1 3/8 | 8.0 | 12.5 | 16.6 | 20.1 | 23.4 | 25.7 | 27.7 | 29.3 | 30.8 | 32.3 | 33.5 |
| 1 1/2 | 9.5 | 15.0 | 19.8 | 24.1 | 27.9 | 30.8 | 33.0 | 35.0 | 36.7 | 38.2 | 39.6 |
| 1 5/8 | 11.2 | 17.5 | 23.2 | 28.5 | 32.7 | 35.9 | 38.8 | 41.0 | 43.0 | 45.0 | 46.5 |
| 1 3/4 | 13.0 | 20.3 | 27.0 | 32.9 | 38.0 | 41.9 | 45.0 | 47.5 | 50.0 | 52.0 | 54.0 |
| 1 7/8 | 14.9 | 23.5 | 31.0 | 37.9 | 43.6 | 48.3 | 52.1 | 54.4 | 57.4 | 59.9 | 61.9 |
| 2 | 17.0 | 26.5 | 35.2 | 42.9 | 49.6 | 54.7 | 58.8 | 62.3 | 65.3 | 67.9 | 70.4 |

$L$ = center to center distance of the sheaves. Where the tight strand is on top, $L$ may be as great as 100 ft., provided that shafts and bearings are sufficiently rugged to withstand the strain of the extra weight of rope. It is not advisable to make the distance between the faces of the sheaves less than 25 ft.

## 2. WIRE ROPE TRANSMISSION

The electrical transmission of energy has largely displaced wire rope as a power transmission medium. For the relatively few installations that require rope, Table 6 will provide the data necessary to calculate the drive. For data on the various types of wire rope available for power transmission, including construction and breaking strength, see pp. 6–34 to 6–43.

**WIRE ROPE SHEAVES.**—Sheaves used with wire rope for the transmission of power are of the form shown in Fig. 6. Dimensions are given in Table 7. The bottom of the sheave is filled with blocks of rubber or leather. The minimum diameter of sheave that should be used with 6 × 7 wire rope is 100 $d$, where $d$ = diameter of rope. With 6 × 19 rope, the minimum sheave diameter should be 66 $d$. Sheaves must be set true on the shafts, and at right angles to the line of transmission to avoid wear on the rope and loss of power. Guide sheaves and supporting sheaves should not be used unless absolutely necessary, for the same reasons. In designing wire rope transmissions to transmit a given horsepower, a choice frequently can be made between small sheaves running at high speed and large sheaves running at a lower speed. Since the wear on the rope always is greater in bending over small sheaves, economy in life of rope will be served by selecting the larger sheaves and lower speeds.

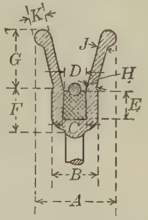


FIG. 6. Wire Rope Sheave Groove

**Table 6.—Horsepower Transmitted by Wire Rope**

American Steel & Wire Co., Cleveland, O.

| Horsepower | Diameter of Rope, in. | Diameter of Sheave, ft. | Revolutions per Minute | Horsepower | Diameter of Rope, in. | Diameter of Sheave, ft. | Revolutions per Minute | Horsepower | Diameter of Rope, in. | Diameter of Sheave, ft. | Revotions per Minute |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3 | 3/8 | 3 | 80 | 25 | 9/16 | 7 | 100 | 84 | 5/8 | 9 | 140 |
| 3 1/2 | 3/8 | 3 | 100 | 26 | 5/8 | 8 | 80 | 85 | 11/16 | 10 | 100 |
| 4 | 3/8 | 3 | 120 | 30 | 9/16 | 7 | 120 | 93 | 11/16 | 12 | 80 |
| 4 | 3/8 | 4 | 80 | 32 | 5/8 | 8 | 100 | 96 | 5/8 | 10 | 120 |
| 4 1/2 | 3/8 | 3 | 140 | 35 | 9/16 | 7 | 140 | 99 | 3/4 | 12 | 80 |
| 5 | 3/8 | 4 | 100 | 39 | 5/8 | 8 | 120 | 102 | 11/16 | 10 | 120 |
| 6 | 3/8 | 4 | 120 | 45 | 5/8 | 8 | 140 | 112 | 5/8 | 10 | 140 |
| 7 | 3/8 | 4 | 140 | 47 | 9/16 | 9 | 80 | 116 | 11/16 | 12 | 100 |
| 9 | 7/16 | 5 | 80 | 48 | 5/8 | 9 | 80 | 119 | 11/16 | 10 | 140 |
| 11 | 7/16 | 5 | 100 | 58 | 9/16 | 9 | 100 | 124 | 3/4 | 12 | 100 |
| 13 | 7/16 | 5 | 120 | 60 | 5/8 | 9 | 100 | 140 | 11/16 | 12 | 120 |
| 14 | 1/2 | 6 | 80 | 64 | 5/8 | 10 | 80 | 141 | 1 | 14 | 80 |
| 15 | 7/16 | 5 | 140 | 68 | 11/16 | 10 | 80 | 148 | 1 1/8 | 14 | 80 |
| 17 | 1/2 | 6 | 100 | 69 | 9/16 | 9 | 120 | 149 | 3/4 | 12 | 120 |
| 20 | 1/2 | 6 | 120 | 73 | 5/8 | 9 | 120 | 173 | 7/8 | 12 | 120 |
| 20 | 9/16 | 7 | 80 | 80 | 5/8 | 10 | 100 | 176 | 1 | 14 | 100 |
| 23 | 1/2 | 6 | 140 | 82 | 9/16 | 9 | 140 | 185 | 1 1/8 | 14 | 100 |

**Table 7.—Dimensions of Sheaves for Wire Rope Transmission**

| Diam. of Rope, in. | Diam. of Sheave, ft. | Dimensions of Rim, in. (See Fig. 6) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | $A$ | $B$ | $C$ | $D$ | $E$ | $F$ | $G$ | $H$ | $J$ | $K$ |
| 1/2 | 3 | 3 | 1 3/4 | 3/4 | 5/8 | 3/4 | 1 3/8 | 2 | 3/8 | 1/4 | 9/16 |
| 5/8 | 4 | 3 1/2 | 2 1/8 | 7/8 | 3/4 | 7/8 | 1 1/2 | 2 1/2 | 7/16 | 5/16 | 5/8 |
| 3/4 | 5 | 4 | 2 3/8 | 1 | 7/8 | 1 | 1 3/4 | 3 | 1/2 | 3/8 | 3/4 |
| 7/8 | 6 | 4 1/2 | 2 5/8 | 1 1/8 | 1 | 1 1/8 | 2 | 3 1/2 | 1/2 | 3/8 | 3/4 |
| 1 | 7 | 4 3/4 | 2 3/4 | 1 1/4 | 1 1/8 | 1 1/4 | 2 1/8 | 3 3/4 | 9/16 | 7/16 | 7/8 |
| 1 1/8 | 8 | 5 | 2 7/8 | 1 3/8 | 1 1/4 | 1 3/8 | 2 1/4 | 4 | 5/8 | 1/2 | 1 |
| 1 1/4 | 9 | 5 1/4 | 3 1/8 | 1 1/2 | 1 3/8 | 1 1/2 | 2 1/2 | 4 1/4 | 5/8 | 1/2 | 1 |
| 1 3/8 | 10 | 5 1/2 | 3 1/4 | 1 5/8 | 1 1/2 | 1 5/8 | 2 5/8 | 4 1/2 | 11/16 | 9/16 | 1 1/8 |
| 1 1/2 | 12 | 6 | 3 5/8 | 1 3/4 | 1 5/8 | 1 3/4 | 3 | 4 3/4 | 3/4 | 5/8 | 1 1/4 |

# COUPLINGS AND CLUTCHES

Couplings are of two general classes: 1. Fast or permanent couplings; 2. Releasing couplings, generally known as clutches. Permanent couplings may connect parallel shafts, having a common axis, or otherwise; they may connect shafts having intersecting axes; or they may connect shafts of inaccurate alignment. The positive clutch consists of a part rigidly fastened to the shaft and a second part free to slide along a spline on the shaft. The abutting faces of the coupling engage through interlocking jaws, or drive by friction. For an excellent discussion on couplings and clutches, see Leutwiler's Machine Design. Also see *Trans.* A.S.M.E., xxx, p. 39, 1908.

## 1. COUPLINGS

**BOX COUPLINGS** prevent relative motion of the shafts by means of a key. Unwin gives the proportions shown in Fig. 1.

**A FRICTION COUPLING** that is both simple and effective for small shafts is shown in Fig. 2, which also gives the proportions.

**CLAMP SHAFT COUPLING** (Table 1).—Halsey's Handbook gives the proportions for the clamp shaft coupling shown in Fig. 3. These correspond closely with the proportions of commercial couplings.

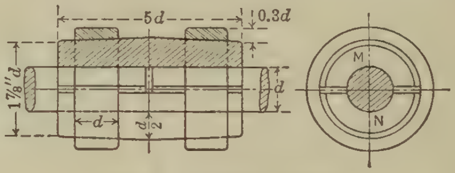
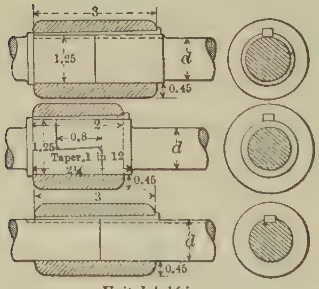


FIG. 2. Friction Coupling



Unit d + ⅛ in.

FIG. 1. Box Coupling

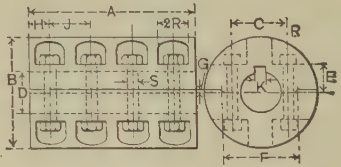


FIG. 3. Clamp Shaft Coupling

**Table 1.—Dimensions of Clamp Shaft Couplings**

(See Fig. 3. Dimensions in inches)

| *D* | *A* | *B* | *C* | *E* | *F* | *G* | *H* | *J* | *K* | *R* | *S* | Weight, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 7/16 | 6 | 4 1/4 | 2 1/2 | 1 | 3 3/8 | 1/8 | 1 1/2 | ...... | 5/16 | 15/16 | 5/8 | 18.7 |
| 1 11/16 | 7 | 4 7/8 | 3 | 1 3/16 | 4 | 3/16 | 1 3/4 | ...... | 7/16 | 1 1/16 | 3/4 | 29.0 |
| 1 15/16 | 8 | 5 1/2 | 3 1/4 | 1 3/8 | 4 1/4 | 1/4 | 2 | ...... | 7/16 | 1 1/16 | 3/4 | 42.7 |
| 2 3/16 | 9 | 6 1/8 | 3 1/2 | 1 1/2 | 4 1/2 | 1/4 | 2 1/4 | ...... | 9/16 | 1 1/16 | 3/4 | 57.9 |
| 2 7/16 | 10 | 6 3/4 | 3 7/8 | 1 5/8 | 5 1/4 | 1/4 | 2 1/2 | ...... | 9/16 | 1 3/16 | 7/8 | 78.9 |
| 2 11/16 | 11 | 7 3/8 | 4 | 1 7/8 | 5 | 1/4 | 1 3/8 | 2 3/4 | 9/16 | 1 1/16 | 3/4 | 94.6 |
| 2 15/16 | 12 | 8 | 4 1/4 | 2 | 5 3/8 | 1/4 | 1 1/2 | 3 | 11/16 | 1 1/16 | 3/4 | 125.2 |
| 3 3/16 | 13 | 8 5/8 | 4 1/2 | 2 1/8 | 5 1/2 | 1/4 | 1 5/8 | 3 1/4 | 11/16 | 1 1/16 | 3/4 | 149.5 |
| 3 7/16 | 14 | 9 | 4 3/4 | 2 1/4 | 6 | 1/4 | 1 3/4 | 3 1/2 | 11/16 | 1 3/16 | 7/8 | 181.7 |
| 3 11/16 | 15 | 9 3/4 | 5 | 2 1/2 | 6 1/4 | 1/4 | 1 7/8 | 3 3/4 | 11/16 | 1 3/16 | 7/8 | 228.6 |
| 3 15/16 | 16 | 10 1/4 | 5 1/2 | 2 3/4 | 7 | 1/4 | 2 | 4 | 13/16 | 1 5/16 | 1 | 277.9 |
| 4 7/16 | 18 | 11 1/2 | 6 | 3 | 7 3/4 | 1/4 | 2 1/4 | 4 1/2 | 13/16 | 1 5/16 | 1 | 389.8 |
| 4 15/16 | 20 | 12 5/8 | 6 3/4 | 3 3/8 | 8 1/2 | 1/4 | 2 1/2 | 5 | 13/16 | 1 9/16 | 1 1/4 | 496.7 |

**HYDROELECTRIC UNIT COUPLINGS.**—Integrally forged shaft couplings for hydroelectric units have been standardized by the A.S.A. as shown in Fig. 4. Dimensions are given in Table 2.

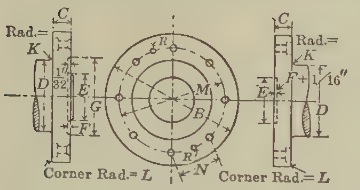


Fig. 4. Integrally-forged Hydroelectric Unit Coupling

**FLANGE COUPLINGS** have not been standardized. The dimensions of stock couplings made by different manufacturers vary considerably. Fig. 5 and Table 3 represent good practice. In these couplings, the center line of the keyway should lie on a diameter through a coupling bolt as shown. Couplings for shafts of 10 in. diameter or over should have two keys 90 deg. apart.

While rules for design are empirical, the resistance of the bolts to the turning movement

**Table 2.—Dimensions of Integrally-forged Shaft Couplings for Hydroelectric Units**
(A.S.A. B49, 1932)
All dimensions in inches. See Fig. 4

| Shaft Diam., *D* | Flange | | | | | | | | | Bolts | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Diam., *B* | Thickness *C* | | Rabbet Diam., *E* | Height *F* | Relief Diam., *G* | Fillet Radius, *H* | Corner Radius, *L* | No. | Diam. | Bolt Circle Diam., *M* | Center Distance of Bolts, *N* | Size of Nuts |
| | | Shaft Subject to Bending | Shaft Not Subject to Bending | | | | | | | | | | |
| 3 1/2 | 7 1/2 | 7/8 | 7/8 | 2 1/8 | 3/16 | ...... | 3/16 | 1/16 | 6 | 7/8 | 5 5/8 | 2.81 | 7/8 |
| 4 | 8 1/8 | 1 | 1 | 2 3/8 | 3/16 | ...... | 3/16 | 1/16 | 6 | 7/8 | 6 1/4 | 3.12 | 7/8 |
| 4 1/2 | 9 | 1 1/8 | 1 1/8 | 2 3/4 | 3/16 | ...... | 3/16 | 1/16 | 6 | 1 | 6 7/8 | 3.43 | 1 |
| 5 | 10 1/4 | 1 1/4 | 1 1/4 | 3 | 3/16 | ...... | 3/16 | 1/16 | 6 | 1 1/4 | 7 3/4 | 3.87 | 1 1/4 |
| 5 1/2 | 10 7/8 | 1 3/8 | 1 3/8 | 3 1/4 | 3/16 | ...... | 1/4 | 1/16 | 6 | 1 1/4 | 8 3/8 | 4.18 | 1 1/4 |
| 6 | 11 1/2 | 1 1/2 | 1 1/2 | 3 5/8 | 3/16 | 6 5/8 | 1/4 | 1/16 | 8 | 1 1/4 | 9 | 3.44 | 1 1/4 |
| 6 1/2 | 12 3/4 | 1 5/8 | 1 5/8 | 3 7/8 | 3/16 | 7 1/8 | 1/4 | 1/16 | 6 | 1 1/2 | 9 3/4 | 4.87 | 1 1/2 |
| 7 | 13 1/2 | 1 3/4 | 1 3/4 | 4 1/4 | 1/4 | 7 3/4 | 3/8 | 3/32 | 8 | 1 1/2 | 10 1/2 | 4.02 | 1 1/2 |
| 7 1/2 | 14 | 1 7/8 | 1 3/4 | 4 1/2 | 1/4 | 8 1/4 | 3/8 | 3/32 | 8 | 1 1/2 | 11 | 4.21 | 1 1/2 |
| 8 | 15 | 2 | 1 7/8 | 4 3/4 | 1/4 | 8 3/4 | 1/2 | 3/32 | 8 | 1 3/4 | 12 | 4.58 | 1 1/2 |
| 8 1/2 | 15 1/2 | 2 1/8 | 1 7/8 | 5 1/8 | 1/4 | 9 3/8 | 1/2 | 3/32 | 8 | 1 3/4 | 12 1/2 | 4.78 | 1 1/2 |
| 9 | 16 1/4 | 2 1/4 | 2 | 5 3/8 | 1/4 | 9 7/8 | 5/8 | 3/32 | 10 | 1 3/4 | 13 1/4 | 4.10 | 1 1/2 |
| 10 | 18 | 2 1/2 | 2 1/4 | 6 | 1/4 | 11 | 3/4 | 3/32 | 10 | 2 | 14 1/2 | 4.47 | 1 3/4 |
| 11 | 19 3/4 | 2 3/4 | 2 1/4 | 6 1/2 | 1/4 | 12 1/8 | 1 | 1/8 | 10 | 2 | 16 1/4 | 5.02 | 1 3/4 |
| 12 | 22 | 3 | 2 1/2 | 7 1/4 | 5/16 | 13 1/4 | 1 | 1/8 | 10 | 2 1/4 | 18 | 5.56 | 2 |
| 13 | 23 1/2 | 3 1/4 | 2 3/4 | 7 3/4 | 5/16 | 14 3/8 | 1 1/4 | 1/8 | 12 | 2 1/4 | 19 1/2 | 5.05 | 2 |
| 14 | 25 | 3 1/2 | 3 | 8 1/2 | 5/16 | 15 3/8 | 1 1/4 | 1/8 | 12 | 2 1/2 | 20 1/2 | 5.32 | 2 1/4 |
| 15 | 27 | 3 3/4 | 3 1/4 | 9 | 5/16 | 16 1/2 | 1 1/2 | 1/8 | 12 | 2 1/2 | 22 1/2 | 5.83 | 2 1/4 |
| 16 | 29 | 4 | 3 1/4 | 9 1/2 | 5/16 | 17 5/8 | 1 1/2 | 3/16 | 12 | 2 3/4 | 24 | 6.21 | 2 1/2 |
| 17 | 30 | 4 1/4 | 3 1/2 | 10 1/4 | 5/16 | 18 3/4 | 1 1/2 | 3/16 | 12 | 2 3/4 | 25 | 6.47 | 2 1/2 |
| 18 | 32 1/2 | 4 1/2 | 3 3/4 | 10 3/4 | 5/16 | 19 7/8 | 1 3/4 | 3/16 | 12 | 3 | 27 | 6.99 | 2 3/4 |
| 19 | 34 | 4 3/4 | 4 | 11 1/2 | 5/16 | 20 7/8 | 1 3/4 | 3/16 | 14 | 3 | 28 1/2 | 6.33 | 2 3/4 |
| 20 | 35 1/4 | 5 | 4 | 12 | 3/8 | 22 | 1 3/4 | 1/4 | 14 | 3 | 29 3/4 | 6.61 | 2 3/4 |
| 21 | 37 1/4 | 5 1/4 | 4 1/4 | 12 1/2 | 3/8 | 23 1/8 | 2 | 1/4 | 14 | 3 1/4 | 31 1/4 | 6.96 | 3 |
| 22 | 39 | 5 1/2 | 4 1/2 | 13 1/4 | 3/8 | 24 1/8 | 2 | 1/4 | 14 | 3 1/2 | 32 1/2 | 7.23 | 3 1/4 |
| 23 | 40 | 5 3/4 | 4 3/4 | 13 3/4 | 3/8 | 25 3/8 | 2 | 1/4 | 14 | 3 1/2 | 33 1/2 | 7.45 | 3 1/4 |
| 24 | 42 | 6 | 4 3/4 | 14 1/2 | 3/8 | 26 3/8 | 2 | 1/4 | 14 | 3 3/4 | 35 | 7.78 | 3 1/2 |
| 25 | 43 1/2 | 6 1/4 | 5 | 15 | 3/8 | 27 1/2 | 2 1/2 | 1/4 | 14 | 3 3/4 | 36 1/2 | 8.10 | 3 1/2 |
| 26 | 45 1/2 | 6 1/2 | 5 1/4 | 15 1/2 | 7/16 | 28 5/8 | 2 1/2 | 1/4 | 14 | 4 | 38 | 8.45 | 3 3/4 |
| 27 | 47 | 6 3/4 | 5 1/2 | 16 1/4 | 7/16 | 29 3/4 | 2 1/2 | 1/4 | 16 | 4 | 39 1/2 | 7.70 | 3 3/4 |
| 28 | 24 1/2 | 7 | 5 3/4 | 16 3/4 | 7/16 | 30 7/8 | 3 | 1/4 | 16 | 4 | 41 | 8.00 | 3 3/4 |
| 29 | 50 1/2 | 7 1/4 | 5 3/4 | 17 1/2 | 7/16 | 31 7/8 | 3 | 1/4 | 16 | 4 1/4 | 42 1/2 | 8.29 | 4 |
| 30 | 51 3/4 | 7 1/2 | 6 | 18 | 7/16 | 33 | 3 | 3/8 | 16 | 4 1/4 | 43 1/2 | 8.48 | 4 |
| 31 | 53 1/2 | 7 3/4 | 6 1/4 | 18 1/2 | 1/2 | 34 | 3 | 3/8 | 16 | 4 1/2 | 44 3/4 | 8.73 | 4 1/4 |
| 32 | 55 | 8 | 6 1/2 | 19 1/4 | 1/2 | 35 1/4 | 3 | 3/8 | 16 | 4 1/2 | 46 1/4 | 9.02 | 4 1/4 |
| 33 | 56 1/2 | 8 1/4 | 6 3/4 | 19 3/4 | 1/2 | 36 1/4 | 3 1/2 | 3/8 | 18 | 4 1/2 | 47 3/4 | 8.32 | 4 1/4 |
| 34 | 58 1/4 | 8 1/2 | 7 | 20 1/2 | 1/2 | 37 3/8 | 3 1/2 | 3/8 | 18 | 4 1/2 | 49 1/2 | 8.56 | 4 1/4 |
| 35 | 60 | 8 3/4 | 7 | 21 | 1/2 | 38 1/2 | 3 1/2 | 3/8 | 18 | 4 3/4 | 50 3/4 | 8.81 | 4 1/2 |
| 36 | 61 3/4 | 9 | 7 1/4 | 21 1/2 | 1/2 | 39 5/8 | 3 1/2 | 3/8 | 18 | 4 3/4 | 52 1/2 | 9.11 | 4 1/2 |
| 37 | 63 1/2 | 9 1/4 | 7 1/2 | 22 1/4 | 1/2 | 40 3/4 | 3 1/2 | 3/8 | 18 | 5 | 53 3/4 | 9.33 | 4 3/4 |
| 38 | 64 3/4 | 9 1/2 | 7 3/4 | 23 | 1/2 | 41 3/4 | 3 1/2 | 3/8 | 18 | 5 | 55 | 9.55 | 4 3/4 |
| 39 | 66 1/2 | 9 3/4 | 7 3/4 | 23 1/2 | 1/2 | 43 | 3 1/2 | 3/8 | 18 | 5 1/4 | 56 1/4 | 9.77 | 5 |
| 40 | 68 | 10 | 8 | 24 | 1/2 | 44 | 3 1/2 | 3/8 | 18 | 5 1/4 | 57 3/4 | 10.03 | 5 |

Couplings for shaft diameters 7 in. and larger have two jack-bolts *R*. Diameters of bolts are as follows: Shaft diameters, 7 to 12 in., inclusive, 3/4 in.; 13 to 24 in., inclusive, 1 in.; 25 to 38 in., inclusive, 1 1/4 in.; 39 and 40 in., 1 1/2.

of the coupling should exceed the torsional strength of the shaft. The load should be evenly distributed among all the bolts, which will be accomplished by accurately fitting them. Kimball and Barr (Elements of Machine Design) give the following formulas for the diameter and number of bolts:

$$b = 0.5\sqrt{d^3/n}; \quad n = 3 + (d/2)$$

where $b$ = diam. of bolt, in.; $d$ = diam. of shaft, in.; $n$ = number of bolts.

**THE SELLERS COUPLING,** Fig. 6, consists of an outer sleeve bored taper from each end; a split cone bushing fits each end. Through bolts inserted through openings draw the cones toward each other, thus clamping them on the shaft with great force. A key also is provided. If the outer sleeve is made in two parts with an interlocking joint, the coupling is easily adjusted. Table 4 gives proportions.

**FLEXIBLE COUPLINGS** are used to connect shafts that may be subject no more or less misalignment. They are of various forms. Several of the principal types are illustrated below.

**The Flexible Shaft Coupling,** Fig. 7, is suitable for connecting shafts requiring more than ordinary flexibility. The torque, $T$, that can be transmitted is limited by the shearing strength of the flexible steel plate $P$ to resist ring shear at the radius $R$. If $f_s$ = allowable shearing stress of the plate, lb. per sq. in., $T = 2\pi R t f_s$. Usually this value of $T$ will be much less than $^1/_{16}\,\pi d^3 f_s$, the strength of the shaft.

**Leather-link Couplings** are used to connect generators to engines or motors to centrifugal pumps, blowers, etc. They then act as insulators between the connected shafts. Moisture, oil, grit or dust are detrimental to them. The torque of the driving shaft is transmitted to the driven shaft through a combination of flanged cups, links and bolts. Flexibility is obtained by making alternate holes in the flanges large enough to give the bolt sufficient

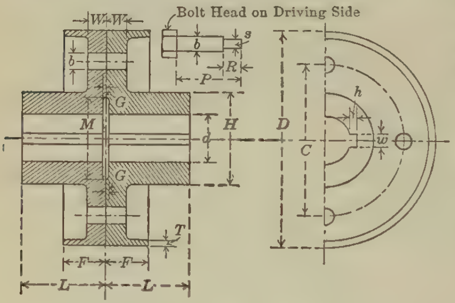


Fig. 5. Flange Coupling

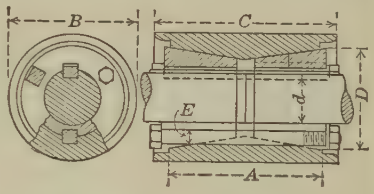


Fig. 6. Sellers Coupling

**Table 3.—Dimensions of Flange Couplings**

| Shaft Diam., d | Dimensions, in. See Fig. 5 | | | | | | | | Bolts | | | | | | Keyway | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $D$ | $L$ | $H$ | $F$ | $W$ | $T$ | $G$ | $M$ | Diam. Bolt Circle, $C$ | Number | Diameter | | Length | | $w$ | $h$ |
| | | | | | | | | | | | $b$ | $s$ | $P$ | $R$ | | |
| 1 | 5 5/8 | 2 1/4 | 2 1/8 | 1 1/8 | 9/16 | 3/16 | 3/32 | 2 | 3 1/8 | 3 | 3/8 | ..... | 1 1/2 | ...... | 1/4 | 1/8 |
| 1 1/4 | 6 1/2 | 2 1/2 | 2 1/2 | 1 3/8 | 5/8 | 3/16 | 3/32 | 2 5/16 | 4 1/2 | 3 | 1/2 | ..... | 1 3/4 | ...... | 5/16 | 5/32 |
| 1 1/2 | 7 1/2 | 2 7/8 | 3 | 1 9/16 | 11/16 | 7/32 | 1/8 | 2 5/8 | 5 3/16 | 3 | 5/8 | 1/2 | 2 | 11/16 | 3/8 | 3/16 |
| 2 | 8 3/4 | 3 1/2 | 3 3/4 | 1 11/16 | 13/16 | 1/4 | 1/8 | 3 3/8 | 6 1/4 | 4 | 5/8 | 1/2 | 2 1/4 | 11/16 | 1/2 | 1/4 |
| 3 | 11 1/4 | 4 3/4 | 5 3/8 | 2 1/8 | 1 1/16 | 5/16 | 5/32 | 4 3/4 | 8 1/4 | 4 | 3/4 | 5/8 | 2 7/8 | 7/8 | 3/4 | 3/8 |
| 4 | 13 1/2 | 6 | 7 | 2 1/2 | 1 5/16 | 3/8 | 3/16 | 6 | 10 1/8 | 4 | 7/8 | 3/4 | 3 1/2 | 1 | 1 | 1/2 |
| 5 | 16 | 7 1/4 | 8 5/8 | 2 15/16 | 1 1/2 | 7/16 | 1/4 | 7 1/2 | 12 1/8 | 6 | 1 | 7/8 | 4 | 1 1/4 | 1 1/4 | 5/8 |
| 6 | 19 | 8 1/2 | 10 1/4 | 3 3/8 | 1 3/4 | 1/2 | 1/4 | 8 3/4 | 14 3/8 | 6 | 1 1/4 | 1 1/8 | 4 3/4 | 1 3/8 | 1 1/2 | 3/4 |
| 7 | 21 1/4 | 9 3/4 | 12 | 3 5/8 | 2 | 5/8 | 5/16 | 10 1/2 | 16 1/4 | 6 | 1 1/4 | 1 1/8 | 5 1/4 | 1 3/8 | 1 5/8 | 7/8 |
| 8 | 24 | 11 | 13 1/2 | 4 3/16 | 2 1/4 | 11/16 | 3/8 | 11 1/2 | 18 3/8 | 8 | 1 1/2 | 1 3/8 | 6 | 1 5/8 | 1 3/4 | 7/8 |
| 10 | 28 | 13 1/2 | 16 3/4 | 4 11/16 | 2 3/4 | 7/8 | 7/16 | 14 1/4 | 21 7/8 | 8 | 1 1/2 | 1 3/8 | 7 | 1 5/8 | 2 | 1 |
| 12 | 33 1/4 | 16 | 20 | 5 9/16 | 3 1/2 | 1 | 1/2 | 17 | 26 | 8 | 1 3/4 | 1 5/8 | 8 1/4 | 1 7/8 | 2 1/4 | 1 |

**Table 4.—Dimensions of Sellers Couplings.** (See Fig. 6)

| $d$ = | 1 1/2 | 1 3/4 | 2 | 2 1/4 | 2 1/2 | 2 3/4 | 3 | 3 1/2 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $A$ | 4 1/2 | 5 1/8 | 6 | 6 3/4 | 7 1/2 | 8 1/4 | 8 1/2 | 10 1/2 | 12 | 14 1/4 | 18 |
| $B$ | 4 1/2 | 5 1/4 | 6 1/8 | 6 1/2 | 7 1/4 | 7 5/8 | 8 1/2 | 9 3/4 | 11 | 12 7/8 | 14 1/2 |
| $C$ | 5 5/8 | 6 5/8 | 7 3/4 | 8 1/2 | 9 1/2 | 10 1/4 | 11 3/8 | 13 1/8 | 14 7/8 | 18 1/4 | 21 1/4 |
| $D$ | 3 1/2 | 4 | 4 3/4 | 5 | 5 3/4 | 6 | 6 3/4 | 7 3/4 | 8 3/4 | 10 1/4 | 11 1/4 |
| $E$ | 7/16 | 1/2 | 5/8 | 5/8 | 3/4 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 1/4 |

play to transmit the torque through the leather link. Fig. 8 shows such a coupling made by the Bruce-Macbeth Co., Cleveland. Dimensions and capacities are given in Table 5.

**Leather-laced Couplings** consist of two cast-iron flanges, on each of which is bolted a steel ring with prongs. An endless belt, laced over the prongs, connects the shafts. The machine can be disconnected without unlacing the belt. Fig. 9 shows such a coupling as made by the General Electric Co. Table 6 gives dimensions and capacities.

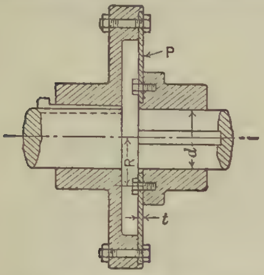


Fig. 7. Flexible Shaft Coupling

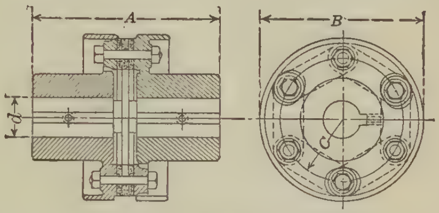


Fig. 8. Leather-link Coupling

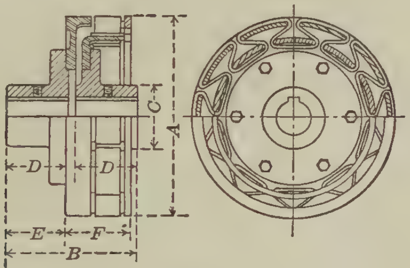


Fig. 9. Leather-laced Coupling

Table 5.—Dimensions and Capacities of Leather-link Couplings. (See Fig. 8)

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Hp. at 100 r.p.m. | 1.5 | 2 | 6 | 10 | 15 | 30 | 50 | 100 | 200 |
| Bore, *d*, in. | 3/4 | 1 3/16 | 1 11/16 | 1 15/16 | 2 3/16 | 2 15/16 | 3 7/16 | 4 7/16 | 5 1/2 |
| Max. r.p.m. | 2400 | 2000 | 1800 | 1600 | 1500 | 1250 | 1000 | 850 | 750 |
| Weight, lb. | 15 | 25 | 65 | 110 | 210 | 335 | 560 | 1270 | 1790 |
| Dimension, *A*, in. | 5 | 6 | 8 | 10 | 13 | 15 | 18 | 26 | 30 |
| " *B*, in. | 5 5/8 | 7 | 10 1/2 | 16 | 20 | 24 | 29 | 34 | 40 |
| " *C*, in. | 2 | 2 1/2 | 4 | 6 | 8 | 10 | 12 | 14 | 16 |

Table 6.—Dimensions and Capacities of Leather-laced Couplings

| Bore, *d*, in. | Hp. at 100 r.p.m. | Max. r.p.m. | Wt., lb. | Dimensions in. See Fig. 9 | | | | | | Key, in. |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | *A* | *B* | *C* | *D* | *E* | *F* | |
| 2 1/2 | 16 | 1200 | 160 | 15 1/2 | 10 | 5 | 4 13/16 | 4 11/16 | 4 3/4 | 1/2 × 1/2 |
| 3 | 27.7 | 900 | 263 | 18 1/2 | 12 | 6 | 5 13/16 | 5 11/16 | 5 1/16 | 3/4 × 3/4 |
| 3 1/2 | 27.7 | 900 | 256 | 18 1/2 | 12 | 6 | 5 13/16 | 5 11/16 | 5 1/16 | 3/4 × 3/4 |
| 4 | 66 | 750 | 494 | 24 1/2 | 14 | 8 | 6 13/16 | 6 11/16 | 6 1/4 | 1 × 1 |
| 4 1/2 | 66 | 750 | 482 | 24 1/2 | 14 | 8 | 6 13/16 | 6 11/16 | 6 1/4 | 1 × 1 |
| 5 | 128 | 600 | 883 | 30 1/2 | 16 | 10 | 7 13/16 | 7 11/16 | 7 1/4 | 1 1/4 × 1 1/4 |
| 5 1/2 | 128 | 600 | 868 | 30 1/2 | 16 | 10 | 7 13/16 | 7 11/16 | 7 1/4 | 1 1/4 × 1 1/4 |
| 6 | 222 | 450 | 1329 | 37 | 18 | 12 | 8 13/16 | 8 11/16 | 7 5/8 | 1 1/2 × 1 1/2 |
| 6 1/2 | 222 | 450 | 1307 | 37 | 18 | 12 | 8 13/16 | 8 11/16 | 7 5/8 | 1 1/2 × 1 1/2 |
| 7 | 352 | 350 | 2076 | 43 | 20 | 14 | 9 13/16 | 9 11/16 | 8 3/8 | 1 1/2 × 1 1/2 |
| 7 1/2 | 352 | 350 | 2046 | 43 | 20 | 14 | 9 13/16 | 9 11/16 | 8 3/8 | 1 1/2 × 1 1/2 |
| 8 | 526 | 300 | 2767 | 49 | 24 | 16 | 11 13/16 | 11 11/16 | 9 1/8 | 1 1/2 × 1 3/4 |
| 8 1/2 | 526 | 300 | 2727 | 49 | 24 | 16 | 11 13/16 | 11 11/16 | 9 1/8 | 1 1/2 × 1 3/4 |
| 9 | 748 | 250 | 3917 | 55 | 28 | 18 | 13 13/16 | 13 11/16 | 9 11/16 | 1 3/4 × 2 |
| 9 1/2 | 748 | 250 | 3865 | 55 | 28 | 18 | 13 13/16 | 13 11/16 | 9 11/16 | 1 3/4 × 2 |
| 10 | 1027 | 200 | 5120 | 61 | 32 | 20 | 15 13/16 | 15 11/16 | 10 5/16 | 1 3/4 × 2 |

**The Westinghouse Flexible Coupling** (Fig. 10) consists of two flanged hubs *F*, with rectangular slots cut in the circumference. In each slot is placed a flexible torque-transmitting element comprising a rectangular steel plate *B*, with cushioning material *C* on each side of it. Snap rings prevent radial movement of these elements. Steel cover plates enclose the assembled coupling. Dimensions and capacities are given in Table 7.

**The Meriam Flexible Coupling,** Fig. 11, consists of two hubs connected by hardened steel rollers embedded in a mixture of ground cork, hard grease and graphite. Table 8 gives dimensions and capacities, and Table 9 service factors for different types of applications. These factors are based on electric motor drive and should be increased about 50% for gas or oil engine drive.

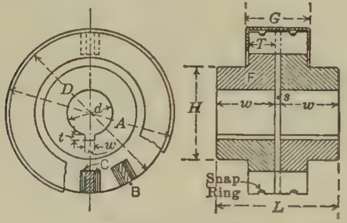


FIG. 10. Westinghouse Flexible Coupling

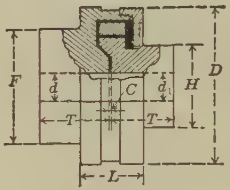


FIG. 11. Meriam Flexible Coupling

**Table 7.—Dimensions and Capacities of Westinghouse Flexible Couplings**

(Westinghouse Electric & Mfg. Co., Pittsburgh, Pa.)

| Hp. per 100 r.p.m. | Shaft diam., *d* |  | Max. rev. r.p.m. | Dimensions, in. (See Fig. 10.) |  |  |  |  |  |  |  |  |  |  |  |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  |  |  |  |  |  |  |  |  | Keyway |  | Slots |  |
|  | Min. | Max. |  | *A* | *L* | *D* | *H* | *T* | *W* | *S* | *G* | *t* | *w* | No. | Size |
| 1.87 | 1 | 1 7/16 | 7300 | 4 3/16 | 2 9/16 | 4 | 2 1/8 | 5/8 | 1 1/4 | 1/16 | 1 7/16 | 5/32 | 1/4 | 5 | 3/4 × 3/4 |
| 3.64 | 1 1/4 | 1 13/16 | 5150 | 5 15/16 | 3 3/16 | 5 3/4 | 2 3/4 | 13/16 | 1 9/16 | 1/16 | 1 13/16 | 5/32 | 1/4 | 6 | 3/4 × 3/4 |
| 7.10 | 1 9/16 | 2 1/4 | 4575 | 6 11/16 | 4 1/16 | 6 1/2 | 3 1/2 | 1 | 2 | 1/16 | 2 3/16 | 7/32 | 3/8 | 6 | 3/4 × 1 |
| 14.95 | 2 | 2 7/8 | 3850 | 7 15/16 | 5 1/8 | 7 3/4 | 4 3/8 | 1 1/4 | 2 1/2 | 1/8 | 2 3/4 | 9/32 | 1/2 | 8 | 1 1/16 × 1 1/2 |
| 29.20 | 2 1/2 | 3 5/8 | 3350 | 9 3/16 | 6 3/8 | 9 | 5 1/2 | 1 9/16 | 3 1/8 | 1/8 | 3 3/8 | 11/32 | 5/8 | 8 | 1 1/8 × 1 1/2 |
| 57.30 | 3 1/8 | 4 1/2 | 2750 | 11 3/16 | 7 7/8 | 11 | 6 3/4 | 1 7/8 | 3 7/8 | 1/8 | 4 | 13/32 | 3/4 | 8 | 1 1/4 × 1 3/4 |
| 119.80 | 4 | 5 3/4 | 2200 | 14 | 10 1/8 | 13 3/4 | 8 5/8 | 2 5/16 | 5 | 1/8 | 5 | 13/32 | 3/4 | 12 | 1 1/2 × 2 |
| 234.0 | 5 | 7 1/4 | 1825 | 16 3/4 | 12 11/16 | 16 1/2 | 10 7/8 | 2 13/16 | 6 1/4 | 3/16 | 6 | 13/32 | 1 | 14 | 1 5/8 × 2 |
| 456.0 | 6 1/4 | 9 | 1475 | 20 3/4 | 15 11/16 | 20 1/2 | 13 1/2 | 3 1/4 | 7 3/4 | 3/16 | 6 7/8 | 15/32 | 1 1/4 | 16 | 1 3/4 × 2 1/2 |
| 957.0 | 8 | 11 1/2 | 1200 | 25 3/8 | 20 1/4 | 25 | 17 1/4 | 3 7/8 | 10 | 1/4 | 8 1/4 | 17/32 | 1 1/2 | 16 | 2 1/2 × 3 |
| 1870.0 | 10 | 14 1/2 | 925 | 33 3/8 | 25 1/4 | 33 | 21 3/4 | 4 | 12 1/2 | 1/4 | 8 1/2 | 21/32 | 2 | 22 | 2 1/2 × 3 |

**Table 8.—Dimensions and Capacities of Meriam Flexible Couplings**

(Hill Clutch Co., Cleveland)

| Hp. at 100 r.p.m. | Bore, max. |  | Dimensions, in. See Fig. 11 |  |  |  |  |  |
|---|---|---|---|---|---|---|---|---|
|  | *d*, in. | $d_1$, in. | *D* | *H* | *F* | *T* | *L* | *C* |
| 1 | 1 1/8 | 1 5/8 | 4 | 2 1/8 | 3 1/2 | 1 3/4 | 1 9/16 | 3/16 |
| 2 | 1 1/2 | 2 | 5 1/2 | 3 | 4 | 2 1/4 | 1 5/8 | 1/4 |
| 8 | 2 | 2 1/2 | 6 1/2 | 4 | 4 1/4 | 2 3/4 | 2 1/2 | 1/4 |
| 15 | 2 1/2 | 3 | 9 | 5 | 6 | 3 1/2 | 3 3/8 | 1/4 |
| 25 | 3 | 3 1/2 | 10 1/2 | 6 | 7 | 4 | 3 1/2 | 3/8 |
| 55 | 4 | 4 1/2 | 13 | 8 | 8 3/4 | 5 | 4 1/4 | 3/8 |
| 110 | 5 | 6 | 16 | 10 | 12 | 6 1/2 | 5 | 1/2 |

**Table 9.—Service Factors for Meriam Couplings**

| Driven Load | Motive Power Factor | Driven Load | Motive Power Factor |
|---|---|---|---|
| Blower, fan or centrifugal........ | 1.0 | Mine hoists, elevators, cranes.... | 2.0 |
| " positive or pressure...... | 2.0 | Pumps |  |
| Centrifugal pump............... | 2.0 | Triplex single-acting........... | 2.0 |
| Compressors.................... | 3.0 | Duplex double-acting.......... | 1.5 |
| Conveyors, belt and chain........ | 2.0 | Pulp grinder................... | 2.0 |
| Generator (uneven load).......... | 1.5 | Tube mills..................... | 2.0 |
| Line shaft...................... | 1.5 | Tumbling barrels (foundry)...... | 2.0 |
|  |  | Woodworking machinery.......... | 2.0 |

**The Oldham Coupling** is used to connect two parallel shafts whose axes are not in the same straight line. The elements of the coupling are shown in Fig. 12. The discs *A* are keyed to the shaft ends, and between them the disc *B* is fitted. Diametral feathers *C* on discs *A* fit in corresponding slots in disc *B*. The feathers on the discs *A* are at right angles to each other. Unwin gives proportions as follows (See Fig. 13), $k = (d + {}^1/_2 \text{ in.})$: $D = 5\,k$; $S = 1.5\,k$; $W = 0.7\,k$; $H = 1.75\,k$; $T = 0.25\,k$; $a = 0.25\,k$; $b = 0.4\,k$. Table 10 gives dimensions and capacities of one commercial type.

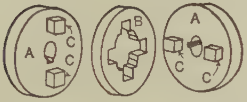


**Fig. 12.** Oldham Flexible Coupling

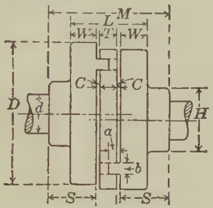


**Fig. 13.** Oldham Flexible Coupling

**Table 10.—Dimensions and Capacities of Oldham Flexible Couplings**

(Dodge Mfg. Co., Mishawaka, Ind.)

| Hp. at 100 r.p.m. | *d*, max. | Dimensions, in. See Fig. 13 | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *D* | *L* | *M* | *W* | *T* | *C* | | *H* | *S* |
| | | | | | | | Min. | Max. | | |
| 1/2 | 1 1/8 | 5 | 2 13/16 | 3 7/16 | 1 | 1/2 | 1/8 | 3/16 | 2 | 1 5/16 |
| 1 | 1 3/8 | 5 1/2 | 3 | 4 1/4 | 1 1/16 | 11/16 | 1/16 | 1/8 | 2 3/8 | 1 11/16 |
| 1 1/2 | 1 5/8 | 6 | 3 3/16 | 5 1/16 | 1 1/8 | 3/4 | 1/16 | 1/8 | 2 3/4 | 2 7/16 |
| 2 3/4 | 2 | 7 1/4 | 3 5/8 | 6 3/4 | 1 1/4 | 7/8 | 1/16 | 3/16 | 3 3/4 | 2 13/16 |

**The Morse Flexible Coupling,** Fig. 14, comprises two forged steel hub members mounted on the shafts to be connected, between which is a floating center to which each hub is separately connected through a pair of resilient rubber trunnion blocks 90 deg. apart. The trunnion blocks are set under pressure in a two-piece riveted pressed steel housing, and are permanently fastened to steel bushings. The steel cores in opposite pairs are rigidly bolted to end hub members of the coupling. Flexibility is provided by the resiliency of the rubber cores. Table 11 gives capacities and dimensions.

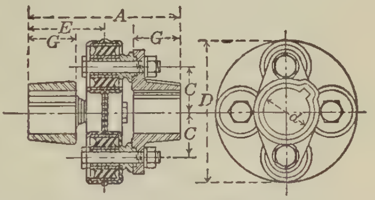


**Fig. 14.** Morse Flexible Coupling

**Table 11.—Dimensions and Capacities of Morse Flexible Couplings**

(Morse Chain Co., Ithaca, N. Y.)

| Hp. at 100 r.p.m. | Bore, *d*, max. | Dimensions, in. See Fig. 14 | | | | |
|---|---|---|---|---|---|---|
| | | *D* | *A* | *C* | *E* | *G* |
| 0.06 | 1/2 | 2 1/8 | 1 3/4 | 21/32 | 7/8 | 9/16 |
| .15 | 1/2 | 2 1/8 | 1 3/4 | 21/32 | 7/8 | 9/16 |
| .25 | 3/4 | 2 1/2 | 2 1/4 | 13/16 | 1 1/8 | 3/4 |
| .38 | 7/8 | 3 | 2 3/4 | 31/32 | 1 3/8 | 7/8 |
| .60 | 1 1/16 | 3 1/2 | 3 1/8 | 1 1/8 | 1 9/16 | 1 |
| .97 | 1 1/4 | 4 | 4 1/8 | 1 9/32 | 2 1/16 | 1 3/16 |
| 1.57 | 1 1/2 | 5 | 4 7/8 | 1 19/32 | 2 7/16 | 1 7/16 |
| 3.15 | 1 13/16 | 6 | 6 3/8 | 1 29/32 | 3 3/16 | 2 |
| 5.40 | 2 1/8 | 7 | 7 5/16 | 2 5/32 | 3 21/32 | 2 3/8 |
| 7.40 | 2 1/2 | 8 | 8 1/4 | 2 15/32 | 4 1/8 | 2 11/16 |
| 10.00 | 2 3/4 | 9 | 9 3/4 | 2 25/32 | 4 7/8 | 3 1/4 |
| 13.80 | 3 | 10 | 11 | 3 3/32 | 5 1/2 | 3 21/32 |
| 20.00 | 3 1/2 | 12 | 12 1/2 | 3 11/16 | 6 1/4 | 4 1/16 |

**Slip Couplings** are used to protect motors against overloads, being designed to slip when the load transmitted is excessive. The slip coupling, Fig. 15, is a form of single disc friction coupling. The flanged hub, *A*, keyed to the driving shaft, has a plate *B* bolted

**Table 12.—Proportions of Slip Couplings**

| *D*, Maximum, in. | Hp. at 100 r.p.m. | Dimensions, inches. See Fig. 15 | | | | | | | | | | | | | | No. of Springs |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | *a* | *b* | *c* | *d* | *e* | *f* | *g* | *h* | *k* | *l* | *m* | *n* | *p* | *q* | |
| 3 | 6 | 18 1/2 | 15 1/2 | 14 | 11 | 7 5/8 | 7 7/8 | 6 | 3 7/8 | 3 3/4 | 3 1/4 | 1 5/8 | 1 1/2 | 1 1/4 | 3/4 | 6 |
| 3 1/2 | 8 | 20 1/2 | 17 1/2 | 16 | 12 | 8 1/2 | 8 1/8 | 7 | 4 5/8 | 4 1/2 | 3 3/8 | 1 3/4 | 1 1/2 | 1 3/8 | 3/4 | 6 |
| 4 | 10 | 21 1/2 | 18 1/2 | 17 | 13 | 9 3/8 | 8 3/8 | 8 | 5 3/8 | 5 | 3 1/2 | 2 | 1 5/8 | 1 1/2 | 3/4 | 7 |
| | 15 | 23 1/2 | 20 1/2 | 19 | 13 | 9 3/8 | 8 3/8 | 8 | 5 3/8 | 5 | 3 1/2 | 2 | 1 5/8 | 1 1/2 | 3/4 | 10 |
| | 20 | 23 3/4 | 20 3/4 | 19 | 13 | 9 1/2 | 8 7/8 | 8 | 5 1/4 | 5 | 3 5/8 | 2 | 1 3/4 | 1 5/8 | 7/8 | 8 |
| | 25 | 24 3/4 | 21 3/4 | 20 | 13 | 9 1/2 | 8 7/8 | 8 | 5 1/4 | 5 | 3 5/8 | 2 | 1 3/4 | 1 5/8 | 7/8 | 10 |
| 4 1/2 | 30 | 26 3/4 | 23 3/4 | 22 | 15 | 10 1/2 | 8 7/8 | 9 | 6 1/4 | 5 3/4 | 3 5/8 | 2 1/4 | 1 3/4 | 1 5/8 | 7/8 | 12 |
| | 35 | 27 3/4 | 24 3/4 | 23 | 15 | 10 1/2 | 8 7/8 | 9 | 6 1/4 | 5 3/4 | 3 5/8 | 2 1/4 | 1 3/4 | 1 5/8 | 7/8 | 12 |
| | 40 | 28 3/4 | 25 3/4 | 24 | 15 | 10 1/2 | 8 7/8 | 9 | 6 1/4 | 5 3/4 | 3 5/8 | 2 1/4 | 1 3/4 | 1 5/8 | 7/8 | 13 |
| | 45 | 29 | 26 | 24 | 15 | 10 3/4 | 9 7/8 | 9 | 6 | 5 3/4 | 4 | 2 1/4 | 2 | 1 3/4 | 1 1/8 | 10 |
| 5 | 50 | 30 | 27 | 25 | 16 | 11 1/2 | 9 7/8 | 10 | 6 3/4 | 6 1/4 | 4 | 2 1/2 | 2 | 1 3/4 | 1 1/8 | 10 |
| | 60 | 32 | 29 | 27 | 16 | 11 1/2 | 9 7/8 | 10 | 6 3/4 | 6 1/4 | 4 | 2 1/2 | 2 | 1 3/4 | 1 1/8 | 12 |
| | 70 | 33 | 30 | 28 | 17 | 11 1/2 | 9 7/8 | 10 | 6 3/4 | 6 1/4 | 4 | 2 1/2 | 2 | 1 3/4 | 1 1/8 | 13 |
| | 80 | 34 1/2 | 31 | 29 | 17 | 11 3/4 | 11 1/4 | 10 | 6 1/2 | 6 1/4 | 4 1/2 | 2 1/2 | 2 1/4 | 1 7/8 | 1 1/8 | 12 |
| 5 1/2 | 90 | 35 1/2 | 32 | 30 | 18 | 12 1/4 | 11 3/8 | 11 | 6 7/8 | 6 3/4 | 4 5/8 | 2 1/2 | 2 1/4 | 1 7/8 | 1 1/8 | 13 |
| 6 | 100 | 36 1/2 | 33 | 31 | 19 | 13 1/4 | 11 7/8 | 12 | 7 1/2 | 7 1/2 | 5 | 2 1/2 | 2 1/2 | 2 | 1 1/8 | 12 |

**Dimensions of Springs**

| Hp. Transmitted | Outside Diam., in. | Length, in. | Diam. of Wire, in. | Pitch of Coils, in. | Hp. Transmitted | Outside Diam., in. | Length, in. | Diam. of Wire, in. | Pitch of Coils, in. | Hp. Transmitted | Outside Diam., in. | Length, in. | Diam. of Wire, in. | Pitch of Coils, in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 1 3/4 | 1 3/4 | 1/4 | 1/2 | 30 | 2 | 1 3/4 | 5/16 | 9/16 | 60 | 2 | 1 3/4 | 3/8 | 9/16 |
| 8 | 1 1/2 | 1 3/4 | 1/4 | 1/2 | 35 | 1 3/4 | 1 3/4 | 5/16 | 9/16 | 70 | 2 | 1 3/4 | 3/8 | 9/16 |
| 10 | 1 1/2 | 1 3/4 | 1/4 | 1/2 | 40 | 1 3/4 | 1 3/4 | 5/16 | 9/16 | 80 | 2 1/2 | 2 1/2 | 7/16 | 5/8 |
| 15 | 1 1/2 | 1 3/4 | 1/4 | 1/2 | 45 | 2 | 1 3/4 | 3/8 | 9/16 | 90 | 2 1/2 | 2 1/2 | 7/16 | 5/8 |
| 20 | 1 3/4 | 1 3/4 | 5/16 | 9/16 | 50 | 2 | 1 3/4 | 3/8 | 9/16 | 100 | 2 1/4 | 2 1/2 | 7/16 | 5/8 |
| 25 | 1 3/4 | 1 3/4 | 5/16 | 9/16 | | | | | | | | | | |

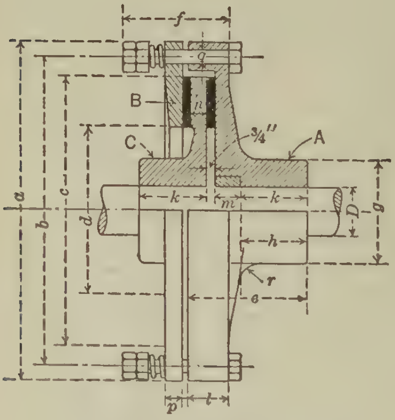


Fig. 15. Slip Coupling

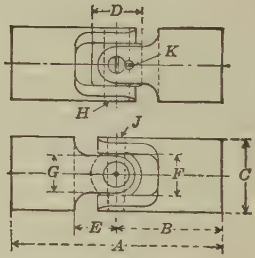


Fig. 16. Hooke Joint

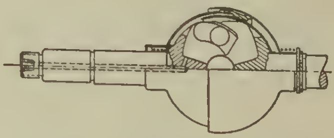


Fig. 17. Weiss Universal Joint

to its rim. Between *A* and *B*, and separated from them by fiber discs, is the flanged hub *C*, keyed to the driven shaft. The bolts connecting the plates *B* with the hub *A* have springs which create a pressure on both faces of the hub *C*. Table 12 (Leutwiler's Machine Design) gives the proportions and other data relative to this type of clutch.

**THE UNIVERSAL JOINT,** also known as a Hooke joint, is used to connect nonparallel shafts which intersect. The joint consists of two U-shaped yokes, fastened to the end of the shafts that are to be connected. A cross-shaped piece, located within these yokes, has four trunnions fitted into bearings on the yokes. Fig. 16 shows the Bocorselski universal joint, as made by the Baush Machine Tool Co., Springfield Mass. Its general dimensions are given in Table 13.

Another form of universal joint, shown in Fig. 17, is the Weiss high-angle joint made by the Weiss Engineering Co., New York. It is designed to maintain a constant velocity, irrespective of the angle of the shafts. In the ordinary universal joint, the angular velocity fluctuates widely as the angle increases. In the Weiss joint, 4 to 8 or more steel balls are placed in intersecting races cut in the joint members. The races guide the balls to precisely half the shaft angle, from zero to its desired maximum, and also compensate for axial movement of the shaft with changes in angle. This joint was primarily developed for automotive use, but has been used in other applications.

**Table 13.—Proportions of Hooke Universal Joints**

| Size | Dimensions, inches. See Fig. 16 | | | | | | | Diameters, inches | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *I* | *J* |
| 1/4 | 1 1/4 | 5/8 | 1/4 | 3/16 | 5/32 | 9/64 | 1/8 | 0.076 | 0.0465 | ....... |
| 3/8 | 1 3/4 | 7/8 | 3/8 | 17/64 | 7/32 | 7/32 | 3/16 | 0.1065 | 0.0595 | ....... |
| 1/2 | 2 | 1 | 1/2 | 11/32 | 9/32 | 9/32 | 1/4 | 0.167 | 0.096 | ....... |
| 5/8 | 2 1/4 | 1 1/8 | 5/8 | 13/32 | 11/32 | 11/32 | 5/16 | 7/32 | 5/32 | ....... |
| 3/4 | 2 11/16 | 1 11/32 | 3/4 | 1/2 | 13/32 | 7/16 | 3/8 | 1/4 | 11/64 | ....... |
| 7/8 | 3 1/4 | 1 5/8 | 7/8 | 19/32 | 15/32 | 1/2 | 7/16 | 9/32 | 3/16 | ....... |
| 1 | 3 3/8 | 1 11/16 | 1 | 11/16 | 9/16 | 9/16 | 1/2 | 5/16 | 7/32 | 1/8 |
| 1 1/4 | 3 3/4 | 1 7/8 | 1 1/4 | 7/8 | 11/16 | 11/16 | 5/8 | 3/8 | 1/4 | 1/8 |
| 1 1/2 | 4 1/4 | 2 1/8 | 1 1/2 | 1 | 27/32 | 27/32 | 3/4 | 1/2 | 11/32 | 1/8 |
| 1 3/4 | 4 1/2 | 2 1/4 | 1 3/4 | 1 3/16 | 15/16 | 31/32 | 7/8 | 9/16 | 3/8 | 1/8 |
| 2 | 5 7/16 | 2 23/32 | 2 | 1 7/16 | 1 3/32 | 1 3/16 | 1 1/16 | 11/16 | 7/16 | 1/8 |
| 2 1/2 | 7 | 3 1/2 | 2 1/2 | 1 3/4 | 1 13/32 | 1 7/16 | 1 5/16 | 3/4 | 1/2 | 1/8 |
| 3 | 9 | 4 1/2 | 3 | 2 1/8 | 1 21/32 | 1 5/8 | 1 7/16 | 7/8 | 5/8 | 1/4 |
| 4 | 10 5/8 | 5 5/16 | 4 | 2 3/4 | 2 3/16 | 2 1/16 | 1 7/8 | 1 1/16 | 3/4 | 1/4 |

## 2. CLUTCHES

Clutches are used to connect shafts to a driving mechanism as a motor, engine or line shaft, and to disconnect them at will. They may be classed as: 1. Jaw clutches, in which jaws of opposed members of the clutch interlock; 2. Friction clutches, in which the driving force is transmitted by the friction of surfaces in contact; 3. Magnetic clutches, in which a magnetizing coil on one member pulls the armature, carrying a friction lining, on the other member into contact with a friction ring on the first member, and thus transmits the torque.

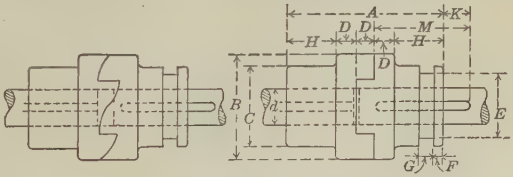


FIG. 18. Jaw Clutches

**JAW CLUTCHES** (Fig. 18) are used for slow-moving shafts, where sudden starting action is not objectionable and when the inertia of the driven parts is relatively small. The various forms of such clutches differ only in the shapes

**Table 14.—Dimensions of Square- and Spiral-jaw Clutches**

(Dodge Mfg. Co., Mishawaka, Ind.)

| Shaft Diam., *d* | Dimensions, in. See Fig. 18 | | | | | | | | | | Wt., lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *K* | *M* | |
| 15/16 | 4 3/4 | 3 | 2 | 1/2 | 1 5/8 | 1/2 | 1/4 | 1 5/8 | 3/4 | 2 7/8 | 18 |
| 1 3/16 | 5 3/8 | 3 1/2 | 2 1/2 | 5/8 | 2 1/8 | 1/2 | 1/4 | 1 3/4 | 1 | 3 3/8 | 22 |
| 1 7/16 | 6 1/4 | 4 1/8 | 3 | 3/4 | 2 1/2 | 9/16 | 3/8 | 2 | 1 1/8 | 3 7/8 | 28 |
| 1 11/16 | 6 7/8 | 4 3/4 | 3 1/2 | 7/8 | 2 7/8 | 9/16 | 3/8 | 2 1/8 | 1 3/8 | 4 3/8 | 35 |
| 1 15/16 | 7 1/2 | 5 3/8 | 4 | 1 | 3 1/4 | 5/8 | 3/8 | 2 1/4 | 1 1/2 | 4 3/4 | 44 |
| 2 3/16 | 8 1/8 | 6 | 4 1/2 | 1 1/8 | 3 5/8 | 5/8 | 3/8 | 2 3/8 | 1 3/4 | 5 1/4 | 58 |
| 2 7/16 | 8 3/4 | 6 3/4 | 5 | 1 1/4 | 4 1/8 | 5/8 | 7/16 | 2 1/2 | 1 7/8 | 5 5/8 | 75 |
| 2 11/16 | 9 3/8 | 7 1/8 | 5 1/2 | 1 3/8 | 4 5/8 | 11/16 | 7/16 | 2 5/8 | 2 | 6 | 81 |
| 2 15/16 | 10 | 7 3/4 | 6 | 1 1/2 | 5 1/8 | 11/16 | 1/2 | 2 3/4 | 2 1/4 | 6 1/2 | 104 |
| 3 7/16 | 11 1/2 | 8 1/4 | 7 | 1 3/4 | 5 7/8 | 3/4 | 1/2 | 3 1/8 | 2 1/2 | 7 3/8 | 136 |
| 3 15/16 | 12 1/2 | 9 1/2 | 8 | 2 | 6 3/4 | 13/16 | 5/8 | 3 1/4 | 2 7/8 | 8 1/8 | 190 |

### Table 15.—Application Factors for Dry-plate Friction Clutches

| Driven Mechanism | Factor | Driven Mechanism | Factor |
|---|---|---|---|
| Air compressors (depending on cycles) | 2.00 to 2.50 | Generator, steady load | 1.00 |
| Ammonia compressors | 2.50 | Hammer mills | 2.00 |
| Blowers | 1.00 | Hoists, elevators, cranes, shovels | 2.00 |
| Brick making and similar machinery | 3.00 | Line shaft (driven by belt or gears) | 1.50 |
| Centrifugal pumps | 1.00 | Printing presses, textile looms, etc. | 1.50 |
| Crushers, veneers, hogs, ball mills, etc. | 2.00 | Pulp grinders, screens, etc. | 1.75 |
| Double-acting pump | 1.50 | Rock and stone crushers | 3.00 |
| Driers and calenders | 2.00 | Single-acting pump | 2.00 |
| Generator, fluctuating load | 1.50 | Woodworking machinery, general | 1.75 |

### Table 16.—Dimensions and Capacities of Single Plate Friction Clutches

(Twin Disc Clutch Co., Racine, Wis.)

| Working Torque, in.-lb. | Hp per 100 r.p.m. | Hub Bore Max., $d$ | Sleeve Bore Max., $d_1$ | Dimensions, in. See Fig. 19 | | | | | | | | | No. of Teeth | Approx. Weight, lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $A$ | $B$ | $C$ | $D$ | $E$ | $F$ | $G$ | $H$ | $J$ | | |
| 1,890 | 3 | 2 | 2 1/4 | 7 1/4 | 6 1/2 | 5 7/16 | 7/8 | 2 1/2 | 2 3/4 | 3/4 | 3/4 | 4 3/8 | 42 | 14 |
| 2,205 | 3 1/2 | 2 7/16 | 2 7/16 | 8 3/4 | 8 | 5 7/16 | 7/8 | 2 1/2 | 2 3/4 | 3/4 | 3/4 | 5 1/8 | 51 | 18 |
| 2,835 | 4 1/2 | 2 7/16 | 2 7/16 | 10 3/4 | 10 | 5 9/16 | 7/8 | 2 5/8 | 2 3/4 | 3/4 | 3/4 | 5 1/8 | 63 | 35 |
| 4,095 | 6 1/2 | 2 7/16 | 2 7/16 | 12 1/4 | 11 3/8 | 5 13/16 | 7/8 | 2 7/8 | 2 3/4 | 3/4 | 3/4 | 5 1/8 | 72 | 45 |
| 6,930 | 11 | 2 1/2 | 3 | 15 5/32 | 14 | 7 9/16 | 1 1/2 | 4 | 3 1/8 | 3/4 | 3/4 | 7 1/2 | 59 | 85 |
| 11,655 | 18 1/2 | 3 1/2 | 3 1/2 | 17 5/32 | 16 | 9 | 1 7/8 | 4 1/4 | 4 3/16 | 7/8 | 1 | 7 1/4 | 67 | 100 |
| 16,600 | 26 | 3 1/2 | 3 1/2 | 19 5/32 | 18 | 9 1/4 | 1 3/4 | 4 5/8 | 4 3/16 | 7/8 | 1 | 7 1/4 | 75 | 130 |
| 22,185 | 35 | 5 | 5 | 22 5/32 | 21 | 10 | 1 3/4 | 5 1/16 | 4 | 1 1/8 | 1 1/8 | 9 3/4 | 87 | 160 |
| 25,210 | 40 | 5 | 5 | 25 5/32 | 24 | 12 7/16 | 1 3/4 | 5 7/8 | 5 1/2 | 1 1/8 | 1 1/8 | 9 3/4 | 99 | 200 |

### Table 17.—Dimensions and Capacities of Multiple-disc Clutches

(Morse Chain Co., Ithaca, N. Y.)

| Hp. Rating at 500 r.p.m. | Dimensions, in. See Fig. 21 | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Shaft Diam. | | $B$ | $C$ | $D$ | $E$ | $F$ | $G$, Single Clutch | | $H$, Double Clutch | |
| | $d$ | $d_1$ | | | | | | Dry | In Oil | Dry | In Oil |
| 2 | 7/8 | 0.812 | 2 3/4 | 1 15/16 | 2 | 2 1/8 | 0.750 | 3.689 | 3.343 | 5.316 | 4.624 |
| 3 | 1 1/8 | 1.062 | 3 | 2 3/16 | 2 1/2 | 2 5/8 | 0.750 | 3.838 | 3.421 | 5.611 | 4.799 |
| 5 | 1 3/8 | 1.250 | 3 1/2 | 2 5/8 | 3 | 3 1/8 | 1.000 | 4.159 | 4.094 | 5.942 | 5.812 |
| 8 | 1 5/8 | 1.500 | 3 3/4 | 2 7/8 | 3 1/2 | 3 11/16 | 1.000 | 4.514 | 4.094 | 6.652 | 5.812 |
| 15 | 1 7/8 | 1.687 | 4 3/4 | 3 3/4 | 4 1/2 | 4 3/4 | 1.250 | 5.566 | 5.146 | 8.006 | 7.166 |
| 25 | 2 1/4 | 2.062 | 5 1/8 | 4 1/8 | 5 1/2 | 5 7/8 | 1.250 | 5.946 | 5.406 | 8.766 | 7.686 |

### Table 18.—Dimensions and Capacities of Radial Contracting Clutches

(Hill Clutch Co., Cleveland)

| Hp. at 100 r.p.m. | Torque at 100 r.p.m., lb.-ft. | No. of Arms | Equiv. Shaft Diam., in. | Bore, in. | | Dimensions, in. See Fig. 20 | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Standard | Max. | $A$ | $H$, Open | $B$ | $C$ | $D$ | $E$ | $F$ | $G$ |
| 9 | 473 | 3 | 1 3/16 | 2 3/4 | 3 1/2 | 12 | 15 | 3 | 4 1/8 | 5 | 3/4 | 1 7/8 | 1 1/2 |
| 12 | 632 | 3 | 1 15/16 | 3 | 4 | 41 | 17 1/2 | 3 1/2 | 4 3/4 | 5 1/4 | 3/4 | 2 1/8 | 1 1/2 |
| 15 | 788 | 3 | 2 3/16 | 3 1/4 | 4 1/2 | 16 | 20 | 4 | 5 | 5 1/4 | 3/4 | 2 1/8 | 1 1/2 |
| 20 | 1,049 | 3 | 2 7/16 | 3 1/2 | 5 | 18 | 22 1/2 | 4 1/2 | 5 5/8 | 7 1/4 | 1 3/16 | 2 5/8 | 2 |
| 27 | 1,417 | 3 | 2 11/16 | 4 | 5 1/2 | 20 | 24 3/4 | 5 | 6 3/8 | 7 1/2 | 1 3/16 | 2 3/4 | 2 1/2 |
| 35 | 1,833 | 3 | 2 15/16 | 4 | 6 | 22 | 27 1/2 | 5 1/2 | 7 1/8 | 8 | 1 3/16 | 3 3/8 | 2 1/2 |
| 45 | 2,361 | 3 | 3 3/16 | 4 1/2 | 7 | 24 | 29 1/2 | 6 | 7 1/2 | 8 1/2 | 1 3/16 | 3 7/8 | 2 1/2 |
| 60 | 3,139 | 4 | 3 7/16 | 4 1/2 | 7 | 27 | 29 1/2 | 6 | 7 1/2 | 8 7/8 | 1 1/4 | 3 7/8 | 2 1/2 |
| 75 | 3,935 | 4 | 3 11/16 | 5 | 7 1/2 | 30 | 33 | 6 3/4 | 8 | 8 3/4 | 1 3/8 | 3 5/8 | 2 1/4 |
| 90 | 4,724 | 4 | 3 15/16 | 5 | 9 | 34 | 36 1/2 | 7 | 8 1/2 | 9 5/8 | 1 3/8 | 4 1/4 | 2 1/4 |
| 110 | 5,773 | 4 | 4 3/16 | 5 1/2 | 9 | 38 | 41 | 7 1/2 | 9 1/4 | 9 5/8 | 1 3/8 | 4 | 2 1/2 |
| 140 | 7,344 | 4 | 4 11/16 | 6 | 10 | 42 | 46 | 8 | 9 3/4 | 10 7/8 | 1 3/8 | 5 3/8 | 2 1/2 |
| 175 | 9,174 | 4 | 4 15/16 | 6 | 11 | 48 | 50 | 9 | 11 | 11 | 1 3/8 | 5 1/2 | ..... |
| 230 | 12,072 | 4 | 5 1/2 | 6 1/2 | 12 | 48 | 57 | 10 | 12 | 11 | 1 3/8 | 5 1/2 | ..... |

or angles of the engaging surfaces. One-half of the coupling is fitted with a feather key, permitting it to be engaged with or disengaged from the other half at will.

If $N$ = number of teeth; $R$ = mean radius of clutch tooth; $A$ = area of engaging face of one tooth; $A'$ = area of the tooth subjected to shear; $f_c$ = allowable unit crushing stress of the clutch jaw; $f_s$ = allowable shearing stress of the shaft, then the relation between strength of the shaft and strength of clutch is

$$ {}^1/_{16} f_s \pi d^3 = R N A f_c = R N A' F_s $$

The spiral form, Fig. 18*a*, is more easily put into gear, but the motion of the shaft always must be in the same direction. Low and Bevis give the following proportions (See Fig. 18*b*) for designing cast-iron jaw couplings: $B = 2d + 2$; $C = 1.7d + 1$; $H = 0.5d + 0.5$; $F = 0.3d + 0.3$; $G = 0.4d + 0.4$; clearance between clutch jaws and shaft $= 0.1d + 0.1$. These equations give a heavier clutch than most commercial jaw clutches. Table 14 gives the proportions used by one maker.

**FRICTION CLUTCHES,** which transmit power by the friction of surfaces in contact may be classed: **1.** Disc or plate clutches, either single or multiple; 2. Radial expanding or contracting band or jaw clutches; 3. Conical clutches. The equations used for friction brakes also are applicable to clutches. See p. 15–15.

All types of friction clutches should have torque capacities of double the torque delivered to them in order to start the load from rest without excessive slip. That is, the torque to be transmitted should be multiplied by a static load factor of 2. If the torque delivered to the clutch is irregular, as in certain types of internal combustion engines, the static load factor should be increased from 2 to possibly 3 or 4.

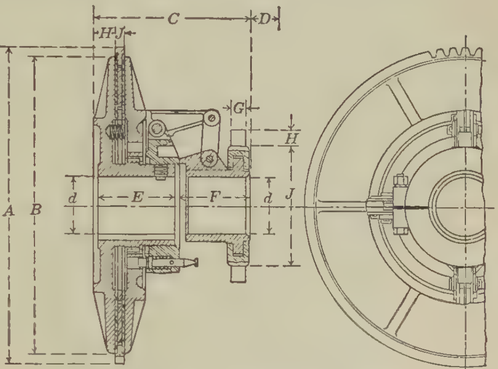


Fig. 19. Single Plate Friction Clutch

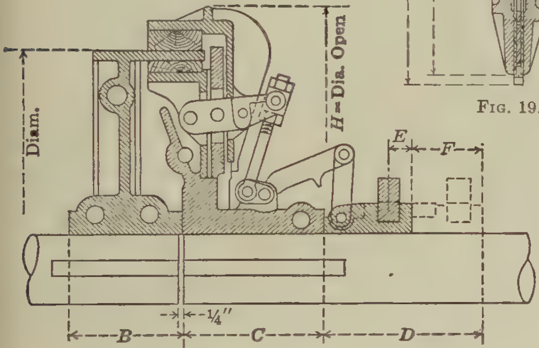


Fig. 20. Radial Jaw Friction Clutch

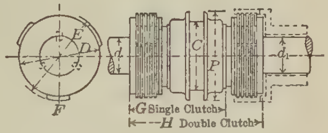


Fig. 21. Multiple-disc Friction Clutch

Additional factors should be applied depending on the character of the load. Table 15 gives factors recommended by the Twin Disc Clutch Co. as a guide in the selection of a clutch based on the characteristics of the driven mechanism.

Example.—Required the torque capacity of a friction clutch to drive woodworking machinery, the prime mover being a 4-cylinder, 40-Hp. Diesel engine running at 1000 r.p.m. *Solution.*—Engine torque $= \{(40 \times 33{,}000)/1000 \times 2\pi)\} \times 12 = 2521$ in.-lb. Diesel engine factor = 3; woodworking machinery factor = 1.75; torque capacity of clutch = 2521 (1.75 + 3.0) = 11,975 in.-lb.

General dimensions and torque capacities of typical heavy-duty single-disc clutches are given in Table 16. Similar information concerning a line of typical multiple-disc clutches is given in Table 17, while Table 18 shows the capacities of one type of radial jaw clutches.

Various materials are used for friction surfaces, but in machine tool applications the tendency is to use hardened steel or phosphor-bronze friction plates. In radial expansion or contraction clutches, the friction surfaces frequently are hard maple and cast iron. Single-plate clutches generally operate without lubricant between friction surfaces (dry type). Multiple-disc clutches are built of the dry type, and also to operate in a bath or spray of oil (wet type).

**Selection of Clutch.**—The type of clutch to be used depends on the service, character of load, space limitations, character of power input, ability to radiate heat and other factors. Single-plate disc clutches generally are used with cranes, gasoline-driven tractors, shovels and similar heavy duty equipment. They also are used in automobiles, for line shaft drives, as are clutches of the radial type, and for various machine applications.

Multiple-disc clutches are used where space is limited, as in machine tool applications, since the same amount of power can be transmitted with two friction discs of a given diameter, as with one disc of twice the diameter. Makers of clutches should be consulted in selecting a clutch for given conditions of service.

**Mounting of Clutches.**—Clutches in industrial work generally are mounted by one of the following methods: 1. The driving plate, to which the friction member is attached, is bolted to the pulley or fly-wheel delivering power to the clutch. The driven element floats on the shaft, either on a splined bushing keyed to the shaft or on splines in the shaft itself. 2. The driving plates and friction discs float on the driving element, which transmits power to it through driving pins, or through an internal toothed ring meshing with gear teeth in the periphery of the driving plate. In some designs a series of projections on the driving plate engage corresponding notches on the driving member. The driven element is rigidly attached to the shaft.

Clutch mountings in automotive work are designed for the special conditions under which they must work.

**The Conical Friction Clutch** consists of the frustum of a cone, so fitted to a shaft by means of a feather key that it can be pushed into an opposite engaging surface rigidly attached to the other shaft. Such clutches require some force to put them into gear. This force, acting parallel to the shaft, produces an undesirable end thrust. Cone clutches may be faced with leather, asbestos fabric, or equipped with cork inserts to increase their driving capacity. The inclination $\alpha$ of the slant side is highly important, in order to avoid "sticking" on the one hand, and too sudden seizure on the other. $\alpha$ usually ranges from 7 1/2 to 13 deg. Proportions may be taken as follows (see Fig. 22*a*): $B = 2\,D + 1$; $C = 1.5\,D$; $F = 1.8\,D$; $H = 0.5\,D$; $E = 0.4\,D + 0.4$; $G = 0.3\,D + 0.3$; $t = 0.2\,D + 0.1$; $L = 2\,D$.

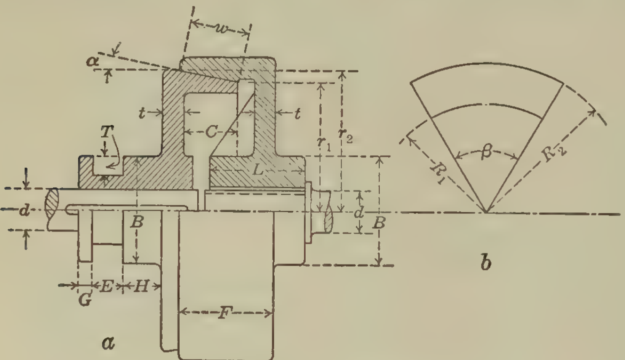


Fig. 22. Conical Friction Clutch

**Formulas for Cone Clutches** (Fig. 22).—Let $M$ = moment clutch is capable of transmitting, in.-lb.; Hp. = horsepower transmitted; $N$ = rev. per min.; $r_1$ = small radius of friction cone, in.; $r_2$ = large radius of friction cone, in.; $r$ = mean radius of friction cone, in.; $V$ = velocity of a point at distance $r$ from center, ft. per min.; $T$ = total tangential force acting at radius $r$, lb.; $P_n$ = total normal pressure between cone surfaces, lb.; $p$ = allowable pressure per sq. in. of contact surface; $P_s$ = total axial or spring pressure, lb.; $\alpha$ = angle of clutch surface with axis of shaft; $\mu$ = coefficient of friction; then

$$r = {}^1\!/_2\,(r_1 + r_2);\ M = T\,r;\ T = \mu P_n;\ V = {}^1\!/_{12}\,(2\,\pi\,r\,N);\ W = P_n/(2\,\pi\,r\,p).$$

For engagement without slip,

$$\text{Hp.} = P_n \mu\, r\, N/63{,}030;\ P_s = (T/\mu)(\sin\alpha + \mu\cos\alpha) = \text{Hp.} \times 63{,}030\{(\sin\alpha + \mu\cos\alpha)/\mu\, r\, N\}$$

and $$P_n = P_s/\sin\alpha, \quad \text{and} \quad P_s = (\text{Hp.} \times 63{,}030\,\sin\alpha)/\mu\, r\, N.$$

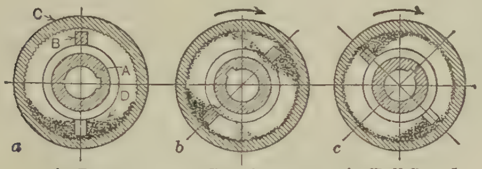
If the cone is to be leather covered (see Fig. 22*b*), then

$$R_1 = r_1/\sin\alpha;\ R_2 = r_2/\sin\alpha;\ R_2 - R_1 = w;\ \beta = \sin\alpha \times 360.$$

The allowable normal pressure $p$, lb. per sq. in., may be taken as 7 or 8 lb. for leather bands, according to *Machinery's Handbook;* Smith and Marx give values of 30 to 40 lb. for leather or cork, and 35 to 45 lb. for maple wood. For values of $\mu$, see Table 1, p. 8–24.

**THE RAMSEY-PULVIS CLUTCH** is a unique form of clutch designed to start a load without shock. The principle is shown in Fig. 23. The rotor $A$, keyed to the driving shaft, has two diametrically opposite blades or paddles $B$ that extend almost to the serrated inner circumference of shell $C$ which is attached to the driven shaft. Hardened steel shot is introduced into the annular space between the rotor and shell. When rotor $A$ is started, the paddles *B* lift the shot as shown in Fig. 23 *b*. As the rotor increases in speed, centrifugal force throws a portion of the shot outward where it is trapped between the serrations in the shell and the ends of the blade causing the shell to start turning. Still further increase in speed causes additional shot to be thrown outward, the several layers interlocking with each other and with the serrations in the shell until they form practically two solid masses as shown in Fig. 23 *c*. The shell and rotor then are in synchronism. The action of the shot as the shell comes up to speed is virtually that of conversion from a fluid to a solid condition.

This clutch is made in several forms for attachment to different types of mechanism. It may be used as a pulley mounted directly on a motor shaft, the shell forming the pulley face. It also may be used as a coupling between two shafts in the same straight line, the shell driving through friction members or studs. Dimensions of the pulley type are given in Table 19 and horsepower ratings in Table 20. This clutch is made by the Ramsey Chain Co., Albany, N. Y.



At Rest Starting At Full Speed

FIG. 23. Action of Ramsey-Pulvis Clutch

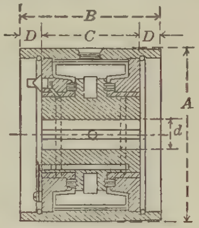


FIG. 24. Dimensions of Ramsey-Pulvis Clutch

**Table 19.—Dimensions of Ramsey-Pulvis Clutches**

| Dimensions, in. See Fig. 24 | Clutch Size | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 12 | 14 | 16 | 18 |
| *A* | 3 9/64 | 3 15/16 | 4 3/4 | 6 | 7 3/32 | 8 | 9 | 10 | 12 | 14 | 16 | 18 |
| *B* | 2 3/4 | 3 5/32 | 3 15/16 | 4 3/4 | 5 1/2 | 6 1/4 | 6 3/4 | 7 1/4 | 9 3/4 | 11 1/4 | 13 1/4 | 14 1/4 |
| *C* | 1 31/32 | 2 9/32 | 2 11/16 | 3 5/16 | 3 29/32 | 4 13/32 | 4 5/8 | 5 | 7 1/4 | 8 1/4 | 9 3/4 | 10 1/4 |
| *D* | 3/8 | 7/16 | 5/8 | 23/32 | 13/16 | 15/16 | 1 1/16 | 1 1/8 | 1 1/4 | 1 1/2 | 1 3/4 | 2 |
| *d*, max. | 1 | 1 1/4 | 1 1/2 | 1 7/8 | 2 1/8 | 2 1/4 | 2 1/2 | 3 | 3 1/2 | 4 | 4 1/2 | 5 |
| Wt., lb. | 6 | 10 | 14 | 20 | 35 | 50 | 65 | 80 | 200 | 320 | 440 | 580 |

**Table 20.—Horsepower Ratings of Ramsey-Pulvis Clutches**

| Clutch, Size | | Revolutions per Minute | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 300 | 400 | 500 | 600 | 700 | 800 | 900 | 1000 | 1200 | 1400 | 1600 | 1800 | 2000 | 2400 |
| 3 | Min. Hp. | ...... | ...... | ...... | ...... | 0.1 | 0.2 | 0.3 | 0.5 | 0.7 | 0.9 | 1.0 | 1.6 | 2.0 | 2.0 |
| | Max. Hp. | ...... | ...... | 0.3 | 0.5 | 0.7 | 1.0 | 1.3 | 1.6 | 2.4 | 3.4 | 4.6 | 6.1 | 8.1 | 14.0 |
| 4 | Min. Hp. | ...... | ...... | 0.1 | 0.2 | 0.3 | 0.4 | 0.7 | 0.9 | 1.2 | 1.7 | 2.0 | 2.5 | 3.0 | 3.0 |
| | Max. Hp. | ...... | 0.4 | 0.6 | 0.8 | 1.1 | 1.3 | 1.6 | 2.2 | 3.0 | 4.2 | 5.5 | 7.2 | 9.3 | 14.5 |
| 5 | Min. Hp. | 0.1 | 0.2 | 0.3 | 0.4 | 0.6 | 0.8 | 1.0 | 1.6 | 2.0 | 2.5 | 3.0 | 4.0 | 5.0 | 5.0 |
| | Max. Hp. | 0.3 | 0.5 | 0.8 | 1.2 | 1.5 | 2.1 | 2.7 | 3.6 | 5.2 | 7.0 | 9.0 | 12.0 | 16.0 | 20.0 |
| 6 | Min. Hp. | 0.2 | 0.4 | 0.6 | 1.0 | 1.3 | 1.7 | 2.0 | 2.5 | 3.0 | 4.0 | 5.0 | 7.5 | 10.0 | ..... |
| | Max. Hp. | 0.4 | 0.8 | 1.5 | 2.4 | 3.3 | 4.2 | 5.4 | 6.6 | 9.5 | 13.0 | 16.0 | 21.0 | 26.0 | ..... |
| 7 | Min. Hp. | 0.2 | 0.5 | 0.8 | 1.0 | 1.7 | 2.0 | 2.8 | 3.8 | 5.0 | 7.0 | 9.0 | 12.0 | 15.0 | ..... |
| | Max. Hp. | 0.6 | 1.4 | 2.3 | 3.7 | 5.0 | 7.0 | 9.0 | 11.0 | 17.0 | 23.0 | 30.0 | 37.0 | 50.0 | ..... |
| 8 | Min. Hp. | 0.3 | 0.5 | 1.0 | 1.4 | 2.0 | 2.5 | 3.5 | 4.5 | 6.0 | 9.0 | 12.0 | 16.0 | ..... | ..... |
| | Max. Hp. | 1.0 | 1.8 | 3.2 | 4.6 | 6.5 | 9.0 | 11.0 | 15.0 | 21.0 | 30.0 | 41.0 | 52.0 | ..... | ..... |
| 9 | Min. Hp. | 0.4 | 0.8 | 1.4 | 2.0 | 3.0 | 3.5 | 5.0 | 7.0 | 9.0 | 13.0 | 18.0 | ..... | ..... | ..... |
| | Max. Hp. | 1.5 | 3.0 | 5.0 | 7.0 | 10.5 | 14.0 | 18.0 | 22.0 | 33.0 | 44.0 | 58.0 | ..... | ..... | ..... |
| 10 | Min. Hp. | 0.7 | 1.5 | 2.5 | 4.0 | 5.0 | 7.0 | 10.0 | 14.0 | 20.0 | 25.0 | ..... | ..... | ..... | ..... |
| | Max Hp. | 3.0 | 6.0 | 10.0 | 17.0 | 23.0 | 30.0 | 38.0 | 47.0 | 66.0 | 85.0 | ..... | ..... | ..... | ..... |
| 12 | Min. Hp. | 1.5 | 3.0 | 5.0 | 7.5 | 11.0 | 14.0 | 17.0 | 20.0 | 26.0 | ..... | ..... | ..... | ..... | ..... |
| | Max. Hp. | 5.0 | 12.0 | 19.0 | 31.0 | 45.0 | 64.0 | 73.0 | 110.0 | 139.0 | ..... | ..... | ..... | ..... | ..... |
| 14 | Min. Hp. | 2.0 | 5.0 | 7.0 | 12.0 | 15.0 | 20.0 | 26.0 | 30.0 | ..... | ..... | ..... | ..... | ..... | ..... |
| | Max. Hp. | 10.0 | 20.0 | 36.0 | 61.0 | 92.0 | 131.0 | 165.0 | 210.0 | ..... | ..... | ..... | ..... | ..... | ..... |
| 16 | Min. Hp. | 2.5 | 6.0 | 10.0 | 15.0 | 22.0 | 33.0 | 44.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | Max. Hp. | 19.0 | 42.0 | 75.0 | 128.0 | 190.0 | 225.0 | 255.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 18 | Min. Hp. | 4.0 | 8.0 | 14.0 | 20.0 | 32.0 | 45.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| | Max. Hp. | 37.0 | 80.0 | 150.0 | 225.0 | 300.0 | 380.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |

**Table 21.—Dimensions and Capacities of Magnetic Clutches**

(Cutler-Hammer, Inc., Milwaukee, Wis.)

| Hp. per 100 r.p.m., max. | Torque, lb.-ft., max. | Safe r.p.m., max. | Direct Current amperes | | Dimensions, in. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 230 Volts | 115 Volts | d, max. | D | L | A | B | C | H | J | K |
| **Type L Clutch. See Fig. 26** | | | | | | | | | | | | | |
| 0.3 | 16 | 3540 | 0.2 | 0.4 | 1 3/8* | 7 | 7 1/4 | 1 3/4 | 3/4 | 4 3/4 | 1 3/4 | 4 1/8 | 1 3/8 |
| 1.7 | 90 | 2840 | .5 | 1.0 | 1 3/8* | 10 | 8 7/16 | 1 5/8 | 7/8 | 3 15/16 | 2 13/16 | 4 1/4 | 1 3/8 |
| 3.0 | 160 | 2380 | .5 | 1.0 | 1 7/8* | 12 | 8 5/8 | 3 3/4 | 5/8 | 4 | 3 1/16 | 4 1/8 | 1 7/16 |
| 7.0 | 365 | 2040 | .7 | 1.4 | 2 7/8 | 14 | 12 1/16 | 3 | 3 15/16 | 5 1/8 | 4 3/8 | 5 3/16 | 2 1/2 |
| 15.0 | 780 | 1790 | .8 | 1.5 | 3 1/2 | 16 | 13 7/8 | 3 9/16 | 5 3/16 | 5 1/8 | 4 11/32 | 5 5/8 | 3 29/32 |
| 17.5 | 930 | 1420 | .8 | 1.5 | 3 1/2 | 20 | 13 7/8 | 3 9/16 | 5 3/16 | 5 1/8 | 4 11/32 | 5 5/8 | 3 29/32 |
| 20.0 | 1,050 | 1190 | .8 | 1.5 | 3 1/2 | 24 | 13 7/8 | 3 9/16 | 5 3/16 | 5 1/8 | 4 11/32 | 5 5/8 | 3 29/32 |
| 30.0 | 1,575 | 1420 | .9 | 1.8 | 4 5/16 | 20 | 15 1/16 | 5 1/4 | 4 9/16 | 5 1/4 | 5 3/8 | 6 1/32 | 3 21/32 |
| 33.0 | 1,750 | 1190 | .9 | 1.8 | 4 5/16 | 24 | 15 1/16 | 5 1/4 | 4 9/16 | 5 1/4 | 5 3/8 | 6 1/32 | 3 21/32 |
| 35.0 | 1,950 | 1030 | .9 | 1.8 | 4 5/16 | 28 | 15 1/16 | 5 1/4 | 4 9/16 | 5 1/4 | 5 3/8 | 6 1/32 | 3 21/32 |
| 50.0 | 2,620 | 1190 | 1.2 | 2.3 | 5 1/2 | 24 | 16 9/16 | 6 | 4 9/16 | 6 | 6 | 6 1/2 | 4 1/16 |
| 53.5 | 2,800 | 1030 | 1.2 | 2.3 | 5 1/2 | 28 | 16 9/16 | 6 | 4 9/16 | 6 | 6 | 6 1/2 | 4 1/16 |
| 58.0 | 3,050 | 900 | 1.2 | 2.3 | 5 1/2 | 32 | 16 9/16 | 6 | 4 9/16 | 6 | 6 | 6 1/2 | 4 1/16 |
| 90.0 | 4,720 | 1030 | 1.4 | 2.8 | 5 1/2 | 28 | 16 3/4 | 6 | 4 3/4 | 6 | 6 | 6 11/16 | 4 1/16 |
| 97.0 | 5,100 | 900 | 1.4 | 2.8 | 5 1/2 | 32 | 16 3/4 | 6 | 4 3/4 | 6 | 6 | 6 11/16 | 4 1/16 |
| 105.0 | 5,550 | 800 | 1.4 | 2.8 | 5 1/2 | 36 | 16 3/4 | 6 | 4 3/4 | 6 | 6 | 6 11/16 | 4 1/16 |
| 130.0 | 6,800 | 900 | 1.8 | 3.6 | 6 5/8 | 32 | 18 1/2 | 6 3/4 | 5 | 6 3/4 | 6 3/4 | 6 11/16 | 5 1/16 |
| 141.0 | 7,400 | 800 | 1.8 | 3.6 | 6 5/8 | 36 | 18 1/2 | 6 3/4 | 5 | 6 3/4 | 6 3/4 | 6 11/16 | 5 1/16 |
| 153.0 | 8,000 | 710 | 1.8 | 3.6 | 6 5/8 | 40 | 18 1/2 | 6 3/4 | 5 | 6 3/4 | 6 3/4 | 6 11/16 | 5 1/16 |
| **Type DL Clutch. See Fig. 27** | | | | | | | | | | | | | |
| 17.2 | 900 | 1420 | 0.8 | 1.5 | 3 1/2† | 19 | 18 1/4 | 5 1/2 | 4 1/8 | 5 | 3 5/8 | 7 7/8 | 6 3/4 |
| 21.3 | 1,112 | 1190 | .8 | 1.5 | 3 1/2† | 24 | 18 1/4 | 5 1/2 | 4 1/8 | 5 | 3 5/8 | 7 7/8 | 6 3/4 |
| 24.2 | 1,270 | 1030 | .8 | 1.5 | 3 1/2† | 28 | 18 1/4 | 5 1/2 | 4 1/8 | 5 | 3 5/8 | 7 7/8 | 6 3/4 |
| 45.0 | 2,400 | 1190 | .9 | 1.8 | 4 1/8† | 24 | 19 9/16 | 5 1/2 | 3 5/8 | 5 3/16 | 4 1/2 | 7 5/8 | 7 1/2 |
| 51.0 | 2,700 | 1030 | .9 | 1.8 | 4 1/8† | 28 | 19 9/16 | 5 1/2 | 3 5/8 | 5 3/16 | 4 1/2 | 7 5/8 | 7 1/2 |
| 57.0 | 3,000 | 900 | .9 | 1.8 | 4 1/8† | 32 | 19 9/16 | 5 1/2 | 3 5/8 | 5 3/16 | 4 1/2 | 7 5/8 | 7 1/2 |
| 80.0 | 4,200 | 1030 | 1.2 | 2.3 | 5 1/2 | 24 | 21 25/32 | 6 1/4 | 4 11/32 | 5 3/4 | 5 | 8 5/16 | 8 |
| 85.0 | 4,450 | 900 | 1.2 | 2.3 | 5 1/2 | 32 | 21 25/32 | 6 1/4 | 4 11/32 | 5 3/4 | 5 | 8 5/16 | 8 |
| 93.0 | 4,900 | 800 | 1.2 | 2.3 | 5 1/2 | 36 | 21 25/32 | 6 1/4 | 4 11/32 | 5 3/4 | 5 | 8 5/16 | 8 |
| 127.0 | 6,700 | 900 | 1.4 | 2.8 | 5 1/2 | 32 | 21 25/32 | 6 1/4 | 4 11/32 | 5 3/4 | 4 7/16 | 8 1/4 | 8 |
| 138.0 | 7,200 | 800 | 1.4 | 2.8 | 5 1/2 | 36 | 21 25/32 | 6 1/4 | 4 11/32 | 5 3/4 | 4 7/16 | 8 1/4 | 8 |
| 148.0 | 7,800 | 710 | 1.4 | 2.8 | 5 1/2 | 40 | 21 25/32 | 6 1/4 | 4 11/32 | 5 3/4 | 4 7/16 | 8 1/4 | 8 |
| 190.0 | 10,000 | 800 | 1.8 | 3.6 | 6 5/8 | 36 | 23 7/16 | 6 3/4 | 5 | 7 | 4 11/16 | 8 7/8 | 8 5/16 |
| 198.0 | 10,400 | 710 | 1.8 | 3.6 | 6 5/8 | 40 | 23 7/16 | 6 3/4 | 5 | 7 | 4 11/16 | 8 7/8 | 8 5/16 |
| 220.0 | 11,600 | 650 | 1.8 | 3.6 | 6 5/8 | 44 1/4 | 23 7/16 | 6 3/4 | 5 | 7 | 4 11/16 | 8 7/8 | 8 5/16 |
| 265.0 | 14,000 | 710 | 2.1 | 4.1 | 6 5/8 | 40 | 23 9/16 | 6 3/4 | 5 | 7 | 4 7/8 | 8 15/16 | 9 3/16 |
| 287.0 | 15,100 | 650 | 2.1 | 4.1 | 6 5/8 | 44 1/4 | 23 9/16 | 6 3/4 | 5 | 7 | 4 7/8 | 8 15/16 | 9 3/16 |
| 300.0 | 16,000 | 600 | 2.1 | 4.1 | 6 5/8 | 48 1/4 | 23 9/16 | 6 3/4 | 5 | 7 | 4 7/8 | 8 15/16 | 9 3/16 |
| 369.0 | 19,400 | 650 | 2.6 | 5.2 | 6 5/8 | 44 1/4 | 23 9/16 | 6 3/4 | 5 | 7 | 4 3/4 | 9 3/8 | 9 3/16 |
| 391.0 | 20,500 | 600 | 2.6 | 5.2 | 6 5/8 | 48 1/4 | 23 9/16 | 6 3/4 | 5 | 7 | 4 3/4 | 9 3/8 | 9 3/16 |
| 412.0 | 21,600 | 530 | 2.6 | 5.2 | 6 5/8 | 52 1/4 | 23 9/16 | 6 3/4 | 5 | 7 | 4 3/4 | 9 3/8 | 9 3/16 |
| **Type DL Clutch. See Fig. 28** | | | | | | | | | | | | | |
| 490.0 | 25,700 | 600 | 2.6 | 5.3 | 7 | 48 | 27 5/8 | 8 1/2 | 10 5/8 | 8 1/2 | 8 1/4 | 10 1/8 | 9 1/4 |
| 515.0 | 27,000 | 530 | 2.6 | 5.3 | 7 | 52 | 27 5/8 | 8 1/2 | 10 5/8 | 8 1/2 | 8 1/4 | 10 1/8 | 9 1/4 |
| 560.0 | 29,500 | 480 | 2.6 | 5.3 | 7 | 58 | 27 5/8 | 8 1/2 | 10 5/8 | 8 1/2 | 8 1/4 | 10 1/8 | 9 1/4 |
| 630.0 | 33,300 | 530 | 2.9 | 5.8 | 7 | 52 3/4 | 28 1/8 | 8 1/2 | 11 1/8 | 8 1/2 | 8 1/4 | 10 5/8 | 9 1/4 |
| 700.0 | 37,000 | 480 | 2.9 | 5.8 | 7 | 58 3/4 | 28 1/8 | 8 1/2 | 11 1/8 | 8 1/2 | 8 1/4 | 10 5/8 | 9 1/4 |
| 740.0 | 39,000 | 425 | 2.9 | 5.8 | 7 | 64 3/4 | 28 1/8 | 8 1/2 | 11 1/8 | 8 1/2 | 8 1/4 | 10 5/8 | 9 1/4 |
| 900.0 | 47,500 | 480 | 3.2 | 6.5 | 7 | 60 | 28 3/4 | 8 1/4 | 12 1/8 | 8 3/8 | 8 1/8 | 11 3/8 | 9 1/4 |
| 1000.0 | 52,500 | 425 | 3.2 | 6.5 | 7 | 66 | 28 3/4 | 8 1/4 | 12 1/8 | 8 3/8 | 8 1/8 | 11 3/8 | 9 1/4 |
| 1070.0 | 56,200 | 395 | 3.2 | 6.5 | 7 | 72 | 28 3/4 | 8 1/4 | 12 1/8 | 8 3/8 | 8 1/8 | 11 3/8 | 9 1/4 |
| 1250.0 | 65,600 | 425 | 3.9 | 7.8 | 8 1/2 | 65 | 33 5/16 | 10 1/2 | 12 5/16 | 10 1/2 | 10 1/4 | 11 9/16 | 11 1/2 |
| 1265.0 | 66,350 | 395 | 3.9 | 7.8 | 8 1/2 | 71 | 33 5/16 | 10 1/2 | 12 5/16 | 10 1/2 | 10 1/4 | 11 9/16 | 11 1/2 |
| 1440.0 | 75,500 | 365 | 3.9 | 7.8 | 8 1/2 | 77 | 33 5/16 | 10 1/2 | 12 5/16 | 10 1/2 | 10 1/4 | 11 9/16 | 11 1/2 |
| 1660.0 | 87,200 | 395 | 4.2 | 8.3 | 9 1/2 | 72 | 36 13/16 | 11 3/4 | 13 5/16 | 11 3/4 | 11 1/2 | 12 | 13 5/16 |
| 1730.0 | 90,800 | 365 | 4.2 | 8.3 | 9 1/2 | 78 | 36 13/16 | 11 3/4 | 13 5/16 | 11 3/4 | 11 1/2 | 12 | 13 5/16 |
| 1835.0 | 96,350 | 345 | 4.2 | 8.3 | 9 1/2 | 84 | 36 13/16 | 11 3/4 | 13 5/16 | 11 3/4 | 11 1/2 | 12 | 13 5/16 |
| 2155.0 | 112,800 | 365 | 4.7 | 9.5 | 9 1/2 | 78 | 39 7/16 | 12 3/4 | 13 15/16 | 12 3/4 | 12 1/2 | 12 11/16 | 14 1/4 |
| 2165.0 | 113,500 | 345 | 4.7 | 9.5 | 9 1/2 | 84 | 39 7/16 | 12 3/4 | 13 15/16 | 12 3/4 | 12 1/2 | 12 11/16 | 14 1/4 |

* Maximum bore of field hub = 2 in. † Maximum bore of field hub = 4 3/8 in.

**MAGNETIC CLUTCHES** are a form of friction clutch in which the friction surfaces are brought into contact by an electromagnet instead of by mechanical means. They may be operated at higher rotative speeds than mechanical clutches, and are recommended for heavy duty where the latter are impractical, for high-speed drives, and where the motor cannot start a heavy load. The principle of the clutch is shown in Fig. 25. A magnet coil *A*, carried by one member of the clutch, attracts to itself the armature *B* carried by a spring plate *C*, which flexes when the magnet is energized. A friction ring *D* carried by the armature is faced with friction lining *F*. When the coil *A* is energized the friction lining is drawn into contact with field friction ring *E*. The friction between *E* and *F* transmits the torque from one clutch member to the other. For heavy duty, two friction linings often are used. See Figs. 27 and 28.

The clutch is engaged by energizing the magnet coil, and disengaged by de-energizing it. The spring plate then pulls the friction lining away from the friction ring until there is an idling clearance of about $^1/_{16}$ in. between them. The current in the magnet coil is varied by a manually-operated controller, which permits the torque to be built up gradually. Heavy loads thus can be started without shock.

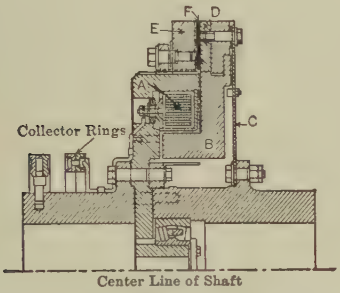


FIG. 25. Magnetic Clutch

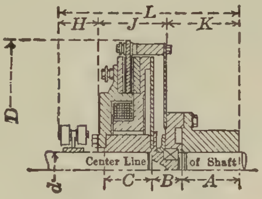


FIG. 27

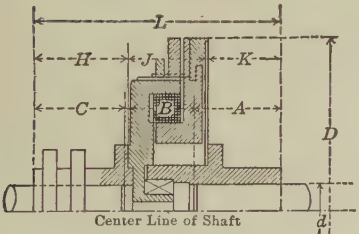


FIG. 26.

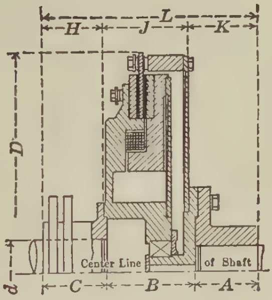


FIG. 28

Dimensions and capacities of magnetic clutches are given in Table 21.

**SLIP RING STARTER.**—A device known as a slip ring starter, developed by the Bethlehem Steel Co., Bethlehem, Pa., may be used to start slowly machinery with heavy inertia load, driven by constant-speed motors. It also may be used as a flexible coupling. The starter depends on the action of centrifugal force to produce pressure between the friction surfaces of the driving and driven members. The construction is shown in Fig. 29. The driving member *A* is keyed to the motor shaft. The driven member *B* is keyed to the machine shaft. Pins *C* project from the driving member *A*, and transmit motion to the interrupted friction rings *F* through the balls *D*. Attached to the friction rings *F* at regular intervals are the weights *E*. Two rings are used in order to balance the drive, and the pins *C* are at opposite edges of a diametral plane. The construction shown in Fig. 29 provides for rotation in one direction only. For rotation in both directions, another set of pins *C* would be provided, which would engage the other ends of the friction rings *F*. A cover *G*, held in place by a spring wire retaining ring *H*, encloses the mechanism.

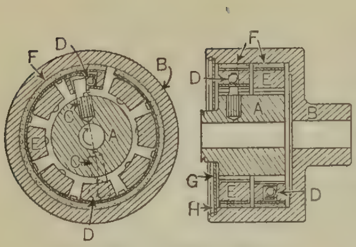


FIG. 29. Slip Ring Starter

In operation, the starting of the motor causes the friction rings to be rotated by the pins *C*. As the speed increases, centrifugal force expands the friction rings until they are in contact with the sleeve *B*. They slip on the sleeve until the pressure due to centrifugal force is sufficient to develop the maximum torque of the friction material, after which the drive operates without slip. The drive normally is designed to transmit 125% of the rated horsepower of the motor, but can be built to transmit any required torque at a specified speed by variation in the weights *E*. The drive will slip under an overload greater than that for which it is designed, but operates as a positive drive as soon as normal conditions are resumed. The torque transmitted at a given speed is not affected to any material extent by variation in the coefficient of friction.

**SHIFTING GEAR FOR CLUTCH COUPLINGS.**—The usual means of engaging or disengaging clutch couplings is a forked lever, whose prongs fit the grooves in the sliding part of the coupling. (See Fig. 30.) Table 22, condensed from *Machinery's Handbook* gives standard dimensions.

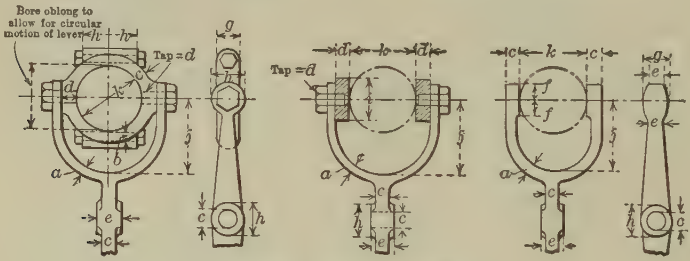


FIG. 30. Clutch Shift Gear

**Table 22.—Dimensions of Clutch-shifting Gear**

| Diam. of Shaft, in. | Dimensions, in. See Fig. 30 | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *a* | *b* | *c* | *d* | *e* | *f* | *g* | *h* | *i* | *j* | *k* |
| 1 | 1/4 | 5/16 | 5/16 | 3/8 | 7/16 | 5/8 | 3/4 | 7/8 | 1 1/2 | 2 3/8 | 1 15/16 |
| 1 1/4 | 5/16 | 5/16 | 3/8 | 7/16 | 1/2 | 3/4 | 7/8 | 1 | 1 5/8 | 2 3/4 | 2 3/16 |
| 1 1/2 | 3/8 | 3/8 | 7/16 | 1/2 | 5/8 | 3/4 | 7/8 | 1 1/8 | 1 7/8 | 3 | 2 1/2 |
| 1 3/4 | 3/8 | 1/2 | 1/2 | 9/16 | 5/8 | 7/8 | 1 | 1 1/4 | 2 | 3 1/2 | 2 7/8 |
| 2 | 3/8 | 1/2 | 5/8 | 5/8 | 3/4 | 7/8 | 1 1/8 | 1 3/8 | 2 1/4 | 3 3/4 | 3 1/4 |
| 2 1/4 | 7/16 | 1/2 | 5/8 | 3/4 | 3/4 | 1 | 1 1/4 | 1 1/2 | 2 1/2 | 4 | 3 5/8 |
| 2 1/2 | 7/16 | 9/16 | 5/8 | 3/4 | 7/8 | 1 1/8 | 1 3/8 | 1 5/8 | 2 3/4 | 4 1/2 | 4 |
| 2 3/4 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 3/8 | 1 3/4 | 2 7/8 | 4 3/4 | 4 3/8 |
| 3 | 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/4 | 1 1/2 | 1 7/8 | 3 1/8 | 5 | 4 3/4 |
| 3 1/4 | 1/2 | 3/4 | 3/4 | 1 | 1 | 1 3/8 | 1 5/8 | 2 | 3 1/4 | 5 1/2 | 5 1/8 |
| 3 1/2 | 5/8 | 3/4 | 7/8 | 1 | 1 1/8 | 1 1/2 | 1 3/4 | 2 1/8 | 3 1/2 | 5 3/4 | 5 1/2 |
| 3 3/4 | 5/8 | 3/4 | 7/8 | 1 1/8 | 1 1/4 | 1 1/2 | 1 3/4 | 2 1/4 | 3 3/4 | 6 | 5 3/4 |
| 4 | 5/8 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 5/8 | 1 7/8 | 2 3/8 | 4 | 6 1/2 | 6 1/4 |
| 4 1/4 | 3/4 | 7/8 | 1 | 1 1/4 | 1 3/8 | 1 3/4 | 2 | 2 1/2 | 4 | 6 3/4 | 6 1/2 |
| 4 1/2 | 3/4 | 7/8 | 1 | 1 1/4 | 1 3/8 | 1 3/4 | 2 | 2 5/8 | 4 1/2 | 7 | 7 |
| 4 3/4 | 3/4 | 7/8 | 1 1/8 | 1 1/4 | 1 1/2 | 1 7/8 | 2 1/4 | 2 3/4 | 4 1/2 | 7 1/2 | 7 1/4 |
| 5 | 3/4 | 1 | 1 1/8 | 1 1/4 | 1 1/2 | 2 | 2 1/4 | 2 7/8 | 4 1/2 | 7 3/4 | 7 5/8 |
| 5 1/2 | 7/8 | 1 | 1 1/4 | 1 1/2 | 1 5/8 | 2 1/8 | 2 1/2 | 3 1/8 | 5 | 8 1/2 | 8 1/2 |
| 6 | 7/8 | 1 | 1 3/8 | 1 1/2 | 1 7/8 | 2 1/4 | 2 3/4 | 3 3/8 | 5 1/2 | 9 | 9 |
| 6 1/2 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/2 | 3 | 3 5/8 | 6 | 9 3/4 | 10 |
| 7 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 1/8 | 2 5/8 | 3 | 3 7/8 | 6 1/2 | 10 1/2 | 10 1/2 |

# SPEED CHANGING MECHANISMS

Speed changing mechanisms are of two general classes: 1. Those in which the ratio of speeds of driving and driven members is fixed, generally known as *speed reducers;* 2. Those in which the ratio of speeds of driving and driven members can be varied at will, generally known as *variable speed drives.* The belt and cone pulley also is, strictly speaking, a speed changing device. See p. 24–12.

## 1. SPEED REDUCERS

**SPEED REDUCERS** comprise, generally, either one or more pairs of spur or herringbone gears, enclosed in a suitable case, or a worm and worm wheel. Another type of speed reducer consists of two sprockets connected by either roller or silent chain.

**Geared Speed Reducers** usually are made with herringbone gears to eliminate all end thrust. The single reducers employ one pair of gears, the pinion being on the input or driving shaft. The *double reduction unit* employs two pairs of gears, the pinion on the input shaft meshing with a gear on an intermediate shaft. The other end of this shaft

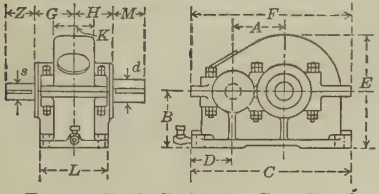


FIG. 1. Single Reduction Gear Speed Reducer

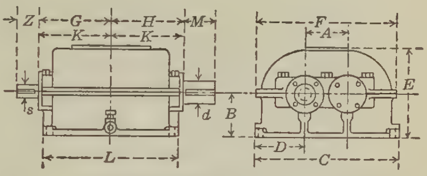


FIG. 2. Double Reduction Gear Speed Reducer

**Table 1.—Dimensions of Herringbone Gear Speed Reducers**

(Link-Belt Co., Chicago)

| Size | Dimensions, in. | | | | | | | | | | Low-speed Shaft | | High-speed Shaft | | Bolts, diam. | Pull † |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *K* | *L* | *M* | *d* | *Z* | *s* * | | |
| **Single Reduction Gears. (Fig. 1)** | | | | | | | | | | | | | | | | |
| S-4 | 4 | 5 | 2 5/8 | 3 1/4 | 9 3/16 | 12 3/8 | 4 1/8 | 4 1/8 | 3 1/16 | 7 | 3 1/4 | 1 5/8 | 2 7/8 | 15/16 | 5/8 | 650 |
| S-5 | 5 | 6 | 3 | 4 1/2 | 11 5/16 | 16 1/4 | 5 1/4 | 5 1/4 | 4 1/4 | 7 3/4 | 4 | 2 | 3 1/8 | 1 1/4 | 5/8 | 1,300 |
| S-6 | 6 | 7 | 3 3/8 | 4 5/8 | 13 5/8 | 19 | 5 3/8 | 5 3/8 | 4 5/8 | 8 1/2 | 5 | 2 1/2 | 3 3/4 | 1 3/8 | 3/4 | 1,600 |
| S-8 | 8 | 9 | 3 7/8 | 5 7/8 | 17 5/8 | 24 | 6 | 6 1/8 | 5 1/2 | 10 | 6 | 3 | 4 1/8 | 1 3/4 | 7/8 | 2,550 |
| S-10 | 10 | 11 | 5 | 6 | 21 3/4 | 28 1/2 | 7 3/8 | 7 3/8 | 6 5/8 | 12 1/2 | 7 | 3 1/2 | 4 3/8 | 2 1/4 | 1 | 4,650 |
| S-12 | 12 | 13 | 5 1/2 | 6 3/4 | 25 3/4 | 33 1/2 | 8 1/8 | 8 3/8 | 7 5/8 | 13 1/2 | 8 | 4 | 5 | 2 1/2 | 1 1/8 | 5,200 |
| S-14 | 14 | 15 | 6 1/2 | 7 7/8 | 29 7/8 | 39 | 8 7/8 | 9 5/8 | 10 | 15 3/4 | 9 | 4 1/2 | 5 1/8 | 3 | 1 1/4 | 6,150 |
| S-16 | 16 | 18 | 7 7/8 | 7 7/8 | 35 1/4 | 45 1/4 | 9 5/8 | 10 7/8 | 11 3/8 | 19 | 10 | 5 | 5 1/4 | 3 3/8 | 1 1/4 | 6,850 |
| S-18 | 18 | 19 1/2 | 9 | 9 1/2 | 38 7/8 | 50 3/4 | 10 3/8 | 12 1/8 | 12 5/8 | 22 | 11 | 5 1/2 | 5 3/4 | 4 | 1 1/2 | 9,550 |
| S-20 | 20 | 21 1/2 | 10 | 11 1/8 | 42 7/8 | 57 3/4 | 11 3/4 | 13 3/8 | 14 | 24 | 11 1/2 | 5 3/4 | 6 1/2 | 4 1/2 | 1 1/2 | 10,000 |
| S-25 | 25 | 26 1/2 | 11 | 13 3/8 | 52 1/4 | 70 7/8 | 12 1/2 | 14 1/2 | 15 1/2 | 26 1/2 | 14 | 7 | 6 1/2 | 4 1/2 | 1 3/4 | 11,250 |
| **Double Reduction Gears (Fig. 2)** | | | | | | | | | | | | | | | | |
| DV-4 | 4 | 9 | 10 1/2 | ...... | 14 5/8 | 5 1/8 | 4 3/8 | 8 3/8 | 4 1/4 | 11 1/4 | 3 | 1 3/4 | 2 | 1 1/16 | 5/8 | 1,000 |
| DL-6 | 6 | 7 | 21 1/2 | 7 3/4 | 14 3/8 | 21 1/2 | 7 5/8 | 8 1/8 | 5 9/16 | 13 1/2 | 4 | 2 3/8 | 2 7/8 | 1 | 3/4 | 1,200 |
| D-5 | 5 | 6 | 18 1/4 | 6 5/8 | 12 1/2 | 18 1/2 | 10 1/8 | 10 3/4 | 7 1/4 | 17 1/4 | 4 1/4 | 2 1/2 | 3 1/8 | 1 1/8 | 3/4 | 1,600 |
| D-6 | 6 | 7 | 22 | 8 | 14 1/2 | 22 | 11 5/8 | 12 5/8 | 8 7/8 | 21 1/4 | 5 1/4 | 3 1/4 | 3 3/4 | 1 1/2 | 7/8 | 4,600 |
| D-8 | 8 | 9 | 28 3/8 | 10 1/4 | 18 5/8 | 28 | 13 1/4 | 14 1/2 | 10 1/8 | 24 3/8 | 5 3/4 | 4 | 4 1/8 | 1 3/4 | 1 | 6,000 |
| D-10 | 10 | 11 1/2 | 34 1/2 | 12 1/4 | 23 | 34 1/2 | 15 3/4 | 17 1/8 | 12 | 29 1/2 | 5 1/2 | 3 5/8 | 4 3/8 | 2 1/8 | 1 1/4 | 9,500 |
| D-12 | 12 | 13 | 40 | 14 | 26 5/8 | 40 3/4 | 17 3/4 | 19 | 13 3/8 | 33 1/4 | 5 3/4 | 4 1/8 | 5 1/8 | 2 3/8 | 1 1/2 | 14,000 |
| D-14 | 28 | 17 | 55 1/4 | 9 3/8 | 33 1/4 | 55 1/4 | 16 9/16 | 18 1/16 | 12 | 33 | 9 | 7 | 5 1/8 | 2 5/8 | 1 1/2 | 14,000 |
| D-18 | 35.4 | 25 | 54 5/8 | 5 7/8 | 49 1/2 | 73 3/8 | 14 7/8 | 17 5/8 | 10 3/8 | 31 | 14 | 7 | 5 1/8 | 3 | 1 5/8 | 16,000 |
| D-20 | 44 1/4 | 30 | 65 5/8 | 6 | 59 1/2 | 86 7/8 | 15 1/2 | 18 1/4 | 11 | 33 1/4 | 14 | 7 1/2 | 5 3/4 | 3 1/2 | 1 3/4 | 20,000 |

* Diameter smaller for the higher ratios; manufacturer should be consulted.
† On slow-speed shaft without outboard bearing.

carries a pinion which meshes with a gear on the driven or output shaft. The *triple reduction unit* employs three pairs of gears, similarly arranged. Gears usually are of steel forgings or steel castings and pinions of alloy steel cut integral with the shaft. Teeth generally are of involute form, 20-deg. pressure angle, and helix angle ranging from

**Table 2.—Horsepower Transmitted by Single Gear Speed Reducers**

(Link-Belt Co., Chicago)

| Size | R.p.m. High Speed Shaft | Horsepower | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| S–4 | Ratio | 2.84 | 3.36 | 3.80 | 4.33 | 4.94 | 5.40 | 5.86 | 6.38 | 6.92 | 7.53 | 8.14 | 8.85 | 9.67 |
| | 580 | 5.3 | 4.8 | 4.4 | 4.1 | 3.7 | 3.5 | 3.0 | 2.6 | 2.3 | 2.0 | 1.8 | 1.6 | 1.4 |
| | 690 | 6.1 | 5.5 | 5.1 | 4.7 | 4.3 | 4.1 | 3.5 | 3.1 | 2.7 | 2.4 | 2.1 | 1.8 | 1.6 |
| | 860 | 7.1 | 6.5 | 6.1 | 5.6 | 5.2 | 4.9 | 4.3 | 3.8 | 3.3 | 2.9 | 2.6 | 2.2 | 1.9 |
| | 1160 | 8.8 | 8.1 | 7.6 | 7.0 | 6.5 | 6.2 | 5.7 | 5.0 | 4.4 | 3.9 | 3.4 | 3.0 | 2.6 |
| | 1750 | 11.3 | 10.5 | 9.9 | 9.3 | 8.7 | 8.3 | 7.9 | 7.4 | 6.4 | 5.7 | 5.0 | 4.4 | 3.8 |
| S–5 | Ratio | 2.91 | 3.29 | 3.74 | 4.29 | 4.93 | 5.43 | 5.92 | 6.42 | 6.77 | 7.33 | 8.23 | 8.92 | 9.91 |
| | 580 | 11.3 | 10.5 | 9.6 | 9.0 | 8.0 | 7.0 | 6.1 | 5.5 | 5.0 | 4.3 | 3.7 | 3.2 | 2.7 |
| | 690 | 12.8 | 12.1 | 11.2 | 10.4 | 9.3 | 8.1 | 7.2 | 6.5 | 5.8 | 5.0 | 4.3 | 3.7 | 3.2 |
| | 860 | 15.0 | 14.1 | 13.2 | 12.2 | 11.2 | 10.0 | 8.7 | 7.9 | 7.1 | 6.1 | 5.2 | 4.5 | 3.9 |
| | 1160 | 18.1 | 17.2 | 16.2 | 15.1 | 14.0 | 13.0 | 11.5 | 10.4 | 9.4 | 8.1 | 6.9 | 6.0 | 5.1 |
| | 1750 | ..... | ..... | ..... | 18.5 | 18.3 | 17.5 | 16.6 | 15.0 | 13.5 | 11.7 | 10.0 | 8.7 | 7.4 |
| S–6 | Ratio | 2.84 | 3.36 | 3.80 | 4.33 | 4.94 | 5.40 | 5.86 | 6.38 | 6.92 | 7.31 | 7.92 | 8.92 | 9.91 |
| | 580 | 17.5 | 16.1 | 15.0 | 13.9 | 12.8 | 11.6 | 10.2 | 9.1 | 8.0 | 7.2 | 6.5 | 5.5 | 4.6 |
| | 690 | 19.8 | 18.2 | 17.1 | 15.8 | 14.6 | 13.6 | 12.0 | 10.6 | 9.3 | 8.4 | 7.6 | 6.4 | 5.3 |
| | 860 | 22.8 | 21.1 | 19.9 | 18.5 | 17.2 | 16.3 | 14.6 | 13.0 | 11.4 | 10.3 | 9.2 | 7.8 | 6.5 |
| | 1160 | 27.3 | 25.5 | 24.2 | 22.7 | 21.2 | 20.2 | 19.1 | 17.0 | 15.0 | 13.6 | 12.2 | 10.4 | 8.6 |
| | 1750 | ..... | ..... | ..... | 28.8 | 27.2 | 26.2 | 25.1 | 23.8 | 21.5 | 19.5 | 17.5 | 14.9 | 12.4 |
| S–8 | Ratio | 2.84 | 3.36 | 3.80 | 4.33 | 4.94 | 5.40 | 5.86 | 6.38 | 6.92 | 7.62 | 8.33 | 8.85 | 9.67 |
| | 580 | 35.3 | 32.4 | 30.3 | 28.3 | 26.1 | 24.3 | 21.4 | 19.0 | 16.8 | 14.6 | 12.5 | 11.1 | 9.7 |
| | 690 | 39.4 | 36.3 | 34.2 | 31.9 | 29.5 | 28.1 | 24.9 | 22.2 | 19.6 | 17.0 | 14.6 | 13.0 | 11.4 |
| | 860 | 44.8 | 41.6 | 39.4 | 36.9 | 34.5 | 32.8 | 30.4 | 27.0 | 23.8 | 20.8 | 17.9 | 15.8 | 13.9 |
| | 1160 | ..... | 49.2 | 46.8 | 44.3 | 41.6 | 39.9 | 38.1 | 35.3 | 31.2 | 27.2 | 23.4 | 20.7 | 18.3 |
| | 1750 | ..... | ..... | ..... | ..... | 48.4 | 48.4 | 48.4 | 46.0 | 42.7 | 36.2 | 33.5 | 29.7 | 26.2 |
| S–10 | Ratio | 2.81 | 3.21 | 3.71 | 4.33 | 4.71 | 5.15 | 5.67 | 6.14 | 6.69 | 7.33 | 8.23 | 8.92 | 9.91 |
| | 580 | 74.6 | 70.3 | 65.8 | 61.0 | 54.5 | 48.3 | 42.1 | 37.4 | 32.8 | 28.2 | 23.7 | 20.6 | 17.6 |
| | 690 | 82.5 | 78.2 | 73.5 | 68.2 | 63.8 | 56.4 | 49.1 | 43.6 | 38.2 | 33.0 | 27.8 | 24.2 | 20.6 |
| | 860 | ..... | 80.0 | 80.0 | 78.0 | 75.0 | 68.3 | 59.6 | 53.0 | 46.5 | 40.0 | 33.8 | 29.5 | 25.2 |
| | 1160 | ..... | ..... | ..... | ..... | 80.0 | 80.0 | 70.0 | 65.0 | 60.5 | 52.3 | 44.1 | 38.5 | 32.9 |
| | 1750 | ..... | ..... | ..... | ..... | ..... | ..... | 72.0 | 72.0 | 72.0 | 65.3 | 57.5 | 53.3 | 47.0 |
| S–12 | Ratio | 2.84 | 3.36 | 3.80 | 4.33 | 4.94 | 5.40 | 5.86 | 6.38 | 6.92 | 7.57 | 8.23 | 8.92 | 9.91 |
| | 475 | 90.0 | 83.7 | 78.8 | 73.5 | 68.1 | 64.7 | 55.5 | 49.0 | 43.0 | 37.9 | 32.8 | 28.6 | 24.4 |
| | 580 | 102.0 | 94.5 | 89.5 | 83.8 | 76.4 | 72.2 | 66.4 | 58.9 | 51.5 | 45.4 | 39.3 | 34.2 | 29.1 |
| | 690 | 111.0 | 104.0 | 99.0 | 93.2 | 87.4 | 83.5 | 77.5 | 68.7 | 60.0 | 53.0 | 46.0 | 40.1 | 34.3 |
| | 860 | ..... | 107.0 | 107.0 | 106.0 | 100.0 | 95.3 | 92.0 | 83.3 | 73.0 | 64.5 | 56.0 | 48.8 | 41.6 |
| | 1160 | ..... | ..... | ..... | ..... | 108.0 | 108.0 | 108.0 | 103.0 | 94.5 | 83.6 | 72.8 | 63.4 | 50.4 |
| S–14 | Ratio | 2.86 | 3.31 | 3.87 | 4.33 | 4.90 | 5.53 | 5.94 | 6.47 | 6.93 | 7.62 | 8.33 | 8.93 | 9.78 |
| | 475 | 125.0 | 125.0 | 125.0 | 125.0 | 119.0 | 98.0 | 87.8 | 77.8 | 69.7 | 60.6 | 52.3 | 47.0 | 40.7 |
| | 580 | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 116.0 | 105.0 | 92.7 | 83.4 | 72.3 | 62.7 | 56.4 | 48.7 |
| | 690 | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 122.0 | 108.0 | 97.1 | 84.5 | 72.9 | 65.7 | 57.1 |
| | 860 | ..... | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 118.0 | 103.0 | 88.7 | 79.9 | 69.3 |
| | 1160 | ..... | ..... | ..... | ..... | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 125.0 | 115.0 | 104.0 | 90.1 |
| S–16 | Ratio | 2.84 | 3.36 | 3.80 | 4.33 | 4.94 | 5.40 | 5.86 | 6.38 | 6.92 | 7.62 | 8.33 | 8.85 | 9.67 |
| | 475 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 143.0 | 127.0 | 114.0 | 100.0 | 86.3 | 74.1 | 65.9 | 59.2 |
| | 580 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 152.0 | 136.0 | 119.0 | 103.0 | 89.4 | 81.1 | 70.2 |
| | 690 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 158.0 | 138.0 | 119.0 | 104.3 | 93.8 | 82.3 |
| | 860 | ..... | ..... | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 160.0 | 145.0 | 125.0 | 114.0 | 100.0 |
| S–18 | Ratio | 2.86 | 3.50 | 3.91 | 4.40 | 4.94 | 5.35 | 5.75 | 6.20 | 6.71 | 7.31 | 8.08 | 8.77 | 9.58 |
| | 475 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 177.0 | 160.0 | 141.0 | 124.0 | 106.0 | 93.9 | 81.0 |
| | 580 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 169.0 | 148.0 | 126.0 | 112.0 | 96.7 |
| | 690 | ..... | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 172.0 | 147.0 | 130.0 | 112.0 |
| | 860 | ..... | ..... | ..... | ..... | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 185.0 | 178.0 | 160.0 | 136.0 |
| S–20 | Ratio | 2.85 | 3.35 | 3.76 | 4.26 | 4.88 | 5.25 | 5.67 | 6.14 | 6.77 | 7.57 | 8.23 | 8.92 | 9.77 |
| | 475 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 211.0 | 182.0 | 154.0 | 135.0 | 119.0 | 103.0 |
| | 580 | ..... | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 193.0 | 161.0 | 144.0 | 123.0 |
| | 690 | ..... | ..... | ..... | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 212.0 | 185.0 | 165.0 | 143.0 |
| | 860 | ..... | ..... | ..... | ..... | ..... | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 215.0 | 200.0 | 173.0 |
| S–25 | Ratio | 2.91 | 3.31 | 3.81 | 4.21 | 4.68 | 5.25 | 5.89 | 6.35 | 6.81 | 7.44 | 8.25 | 8.93 | 9.71 |
| | 390 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 254.0 | 224.0 | 188.0 | 168.0 | 145.0 |
| | 475 | ..... | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 265.0 | 227.0 | 200.0 | 173.0 |
| | 580 | ..... | ..... | ..... | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 270.0 | 238.0 | 206.0 |
| | 690 | ..... | ..... | ..... | ..... | ..... | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 240.0 |
| | 860 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 | 275.0 |

23 to 30 deg. Splash lubrication generally is used for gears and bearings. Table 1 and Figs. 1 and 2 give dimensions of a typical line of herringbone gear reducers. Tables 2 and 3 give capacities.

### Table 3.—Horsepower Transmitted by Double Gear Speed Reducers

(Link-Belt Co., Chicago)

| Size | R.p.m. High Speed Shaft | Horsepower | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| DV–4 | Ratio | ...... | 15.00 | 19.77 | 24.46 | 30.00 | 35.64 | 40.98 | 46.03 | 55.10 | ...... | ..... | ..... |
| | 580 | ...... | 3.0 | 2.4 | 1.9 | 1.5 | 1.2 | 1.0 | 0.9 | 0.6 | ...... | ..... | ..... |
| | 690 | ...... | 3.6 | 2.8 | 2.2 | 1.7 | 1.4 | 1.2 | 1.1 | 0.8 | ...... | ..... | ..... |
| | 860 | ...... | 4.4 | 3.4 | 2.8 | 2.5 | 1.8 | 1.5 | 1.4 | 1.0 | ...... | ..... | ..... |
| | 1160 | ...... | 5.6 | 4.3 | 3.7 | 2.8 | 2.3 | 2.0 | 1.8 | 1.3 | ...... | ..... | ..... |
| | 1750 | ...... | 7.4 | 5.8 | 4.6 | 4.1 | 3.2 | 3.0 | 2.6 | 1.9 | ...... | ..... | ..... |
| DL–6 | Ratio | 10.49 | 15.47 | 19.93 | 24.84 | 29.37 | 36.35 | 42.00 | 49.40 | 52.55 | 57.46 | 64.34 | 70.12 |
| | 580 | 7.3 | 5.7 | 4.8 | 3.9 | 3.3 | 2.5 | 2.1 | 1.6 | 1.5 | 1.3 | 1.2 | 1.1 |
| | 690 | 8.5 | 6.6 | 5.4 | 4.5 | 3.7 | 3.1 | 2.5 | 2.0 | 1.8 | 1.6 | 1.4 | 1.3 |
| | 860 | 9.8 | 7.3 | 6.2 | 5.1 | 4.3 | 3.4 | 2.9 | 2.4 | 2.2 | 2.0 | 1.8 | 1.6 |
| | 1160 | 13.3 | 9.8 | 8.4 | 7.1 | 6.1 | 4.9 | 4.2 | 3.3 | 3.0 | 2.6 | 2.4 | 2.2 |
| | 1750 | 16.3 | 13.6 | 12.1 | 10.5 | 9.2 | 7.6 | 6.6 | 5.6 | 5.2 | 4.7 | 4.2 | 3.9 |
| D–5 | Ratio | 10.29 | 15.49 | 21.41 | 26.00 | 29.33 | 35.44 | 43.50 | ...... | ...... | ...... | ..... | ..... |
| | 580 | 12.9 | 7.8 | 5.9 | 4.6 | 4.1 | 3.4 | 2.7 | ...... | ...... | ...... | ..... | ..... |
| | 690 | 14.6 | 9.2 | 6.9 | 5.4 | 4.8 | 3.9 | 3.3 | ...... | ...... | ...... | ..... | ..... |
| | 860 | 17.5 | 11.2 | 8.6 | 6.7 | 6.0 | 4.9 | 4.1 | ...... | ...... | ...... | ..... | ..... |
| | 1160 | 21.5 | 14.7 | 11.3 | 8.9 | 8.0 | 6.5 | 5.4 | ...... | ...... | ...... | ..... | ..... |
| | 1750 | 27.4 | 19.0 | 15.2 | 13.0 | 11.6 | 9.5 | 8.0 | ...... | ...... | ...... | ..... | ..... |
| D–6 | Ratio | 10.5 | 15.11 | 21.18 | 26.86 | 30.29 | 36.00 | 40.09 | 44.89 | 49.00 | ...... | ..... | ..... |
| | 580 | 21.9 | 15.5 | 11.5 | 9.5 | 8.3 | 6.8 | 6.1 | 5.4 | 4.8 | ...... | ..... | ..... |
| | 690 | 24.9 | 18.5 | 13.6 | 11.2 | 9.6 | 7.9 | 7.3 | 6.3 | 5.6 | ...... | ..... | ..... |
| | 860 | 28.9 | 21.8 | 16.7 | 13.7 | 11.4 | 9.5 | 9.0 | 7.6 | 6.8 | ...... | ..... | ..... |
| | 1160 | 34.9 | 26.8 | 22.0 | 17.1 | 14.4 | 12.0 | 11.9 | 9.6 | 8.7 | ...... | ..... | ..... |
| | 1750 | 43.7 | 34.3 | 27.1 | 22.6 | 19.2 | 16.2 | 16.2 | 13.1 | 11.8 | ...... | ..... | ..... |
| D–8 | Ratio | 10.71 | 15.59 | 21.48 | 25.40 | 30.33 | 36.11 | 41.89 | 43.67 | 47.67 | 59.42 | ..... | ..... |
| | 580 | 42.2 | 31.4 | 23.5 | 19.5 | 16.1 | 13.6 | 11.7 | 10.5 | 9.5 | 7.8 | ..... | ..... |
| | 690 | 50.6 | 36.0 | 28.3 | 22.4 | 19.2 | 16.1 | 14.0 | 12.2 | 10.9 | 9.0 | ..... | ..... |
| | 860 | 58.1 | 41.3 | 31.3 | 27.5 | 23.2 | 19.8 | 16.1 | 14.5 | 13.0 | 10.9 | ..... | ..... |
| | 1160 | 68.1 | 50.2 | 38.1 | 35.9 | 30.7 | 25.6 | 20.2 | 18.2 | 17.1 | 13.7 | ..... | ..... |
| | 1750 | ...... | 62.7 | 48.5 | 48.5 | 42.6 | 33.4 | 26.8 | 24.2 | 22.0 | 18.5 | ..... | ..... |
| D–10 | Ratio | 10.12 | 15.49 | 20.49 | 24.54 | 30.29 | 36.09 | 39.46 | 47.21 | 51.56 | 59.61 | ..... | ..... |
| | 580 | 63.0 | 41.2 | 34.3 | 28.6 | 23.2 | 19.5 | 17.8 | 14.9 | 13.6 | 11.1 | ..... | ..... |
| | 690 | 75.0 | 49.0 | 40.8 | 34.1 | 27.6 | 23.2 | 21.2 | 17.7 | 16.2 | 13.2 | ..... | ..... |
| | 860 | 93.6 | 61.1 | 50.9 | 42.5 | 34.4 | 28.8 | 26.4 | 22.0 | 20.2 | 16.5 | ..... | ..... |
| | 1160 | 126.0 | 82.4 | 68.6 | 57.2 | 46.4 | 39.0 | 35.6 | 29.8 | 27.2 | 22.2 | ..... | ..... |
| | 1750 | ...... | ...... | 103.0 | 86.5 | 70.0 | 58.7 | 53.7 | 45.0 | 41.0 | 33.6 | ..... | ..... |
| D–12 | Ratio | 10.5 | 15.27 | 21.17 | 25.31 | 30.29 | 36.00 | 40.85 | 45.73 | 50.40 | 59.61 | 71.00 | ..... |
| | 580 | 94.0 | 64.5 | 51.0 | 42.9 | 35.8 | 30.2 | 26.5 | 23.7 | 21.5 | 17.4 | 14.6 | ..... |
| | 690 | 112.0 | 77.0 | 61.0 | 51.0 | 42.6 | 35.8 | 31.6 | 28.2 | 25.6 | 20.8 | 17.4 | ..... |
| | 860 | 139.0 | 96.0 | 76.0 | 63.5 | 53.1 | 44.7 | 39.4 | 35.2 | 32.0 | 25.8 | 21.7 | ..... |
| | 1160 | 188.0 | 129.0 | 102.0 | 85.8 | 71.6 | 60.4 | 53.0 | 47.4 | 43.1 | 34.8 | 29.2 | ..... |
| | 1750 | ...... | ...... | ...... | 129.0 | 108.0 | 91.0 | 80.0 | 71.6 | 65.0 | 52.6 | 44.2 | ..... |
| D–14 | Ratio | 10.11 | 14.94 | 19.79 | 25.39 | 30.12 | 36.10 | 39.64 | 46.36 | 51.01 | 62.29 | 71.11 | ..... |
| | 580 | 108.0 | 92.0 | 75.6 | 63.0 | 53.0 | 49.0 | 43.6 | 38.8 | 34.2 | 28.2 | 24.6 | ..... |
| | 690 | 129.0 | 107.0 | 89.4 | 75.0 | 64.0 | 58.3 | 52.0 | 45.0 | 40.8 | 33.6 | 29.4 | ..... |
| | 860 | 143.0 | 129.0 | 110.0 | 92.5 | 78.5 | 72.5 | 64.6 | 56.0 | 50.7 | 41.7 | 36.4 | ..... |
| | 1160 | ...... | 149.0 | 133.0 | 114.0 | 97.5 | 86.8 | 79.8 | 69.0 | 62.7 | 51.5 | 45.0 | ..... |
| D–18 | Ratio | 11.52 | 13.62 | 15.87 | 19.52 | 24.37 | 28.79 | 35.00 | 42.87 | 50.00 | 58.79 | 70.78 | ..... |
| | 580 | 172.0 | 159.0 | 146.0 | 124.0 | 102.0 | 86.0 | 71.0 | 58.0 | 50.0 | 43.0 | 36.0 | ..... |
| | 690 | 187.0 | 175.0 | 161.0 | 140.0 | 118.0 | 101.0 | 80.0 | 69.0 | 60.0 | 51.0 | 43.0 | ..... |
| | 860 | 206.0 | 194.0 | 182.0 | 161.0 | 140.0 | 123.0 | 103.0 | 86.0 | 73.0 | 63.0 | 55.0 | ..... |
| | 1160 | 225.0 | 213.0 | 201.0 | 183.0 | 162.0 | 144.0 | 123.0 | 102.0 | 87.0 | 74.0 | 64.0 | ..... |
| D–20 | Ratio | 11.52 | 13.62 | 15.87 | 19.52 | 24.37 | 28.79 | 35.00 | 42.87 | 50.00 | 58.79 | 70.78 | ..... |
| | 580 | 243.0 | 228.0 | 212.0 | 185.0 | 160.0 | 141.0 | 120.0 | 98.0 | 84.0 | 71.0 | 56.0 | ..... |
| | 690 | 264.0 | 252.0 | 235.0 | 210.0 | 181.0 | 161.0 | 142.0 | 115.0 | 100.0 | 86.0 | 71.0 | ..... |
| | 860 | 288.0 | 276.0 | 262.0 | 240.0 | 212.0 | 191.0 | 165.0 | 139.0 | 123.0 | 106.0 | 90.0 | ..... |

**Table 4.—Dimensions of Worm Gear Speed Reducers**

| No. | Dimensions, in. (Fig. 3) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *K* | *L* | *M* | *d* | *w* |
| 1 | 7 1/2 | 6 | 3 | 4 1/4 | 9 | 4 1/8 | 5 | 2 | 6 | 3 1/2 | 2 1/2 | 1 3/16 | 3/4 |
| 2 | 14 | 11 1/2 | 6 | 8 1/2 | 17 3/4 | 8 | 9 1/2 | 2 3/4 | 12 1/2 | 7 1/4 | 3 1/2 | 2 | 1 5/16 |
| 3 | 30 | 20 | 12 1/2 | 17 1/2 | 35 1/2 | 15 | 16 1/2 | 3 3/4 | 20 1/2 | 11 1/2 | 6 | 3 7/16 | 2 5/16 |
| 4 | 64 | 36 | 27 | 38 | 74 | 29 | 31 | 8 | 34 | 18 1/2 | 10 | 6 5/8 | 4 1/2 |

**Table 5.—Service Factors for Worm Gear Reducers***

| Class of Service | Drive | | |
|---|---|---|---|
| | Motor; Turbine | Steam Engine; 4-cyl. Gas or Oil Engine | 1- or 2-cyl. Gas or Oil Engine |
| Steady load, as centrifugal blowers, centrifugal pumps, forced draft fans | 1 | 1.5 | 2 |
| Slightly fluctuating load, as liquid agitators, induced draft fans, rotary pumps and blowers, line-shaft drives, freely starting conveyors | 1.25 | 1.75 | 2.25 |
| Widely fluctuating load, as plastic agitators, paper mill beaters, bucket elevators, machine tool drives, rubber calenders, conveyors starting under load, stokers | 1.5 | 2. | 2.5 |
| Vibrating load, as rotary kilns, rotary dryers, skip hoists, cranes, punch presses | 1.75 | 2.25 | 2.75 |
| Shock loads, as rolling mills, ball and bar mills, crushers, single cylinder pumps, air compressors, screw presses | 2 | 2.25 | 3 |

* Occasional overloads up to 40 and 80% should take service factors one and two classes larger than that used for the same service without overloads. For high-starting torque loads, the drives should have a capacity of not less than 50% of the starting peak load.

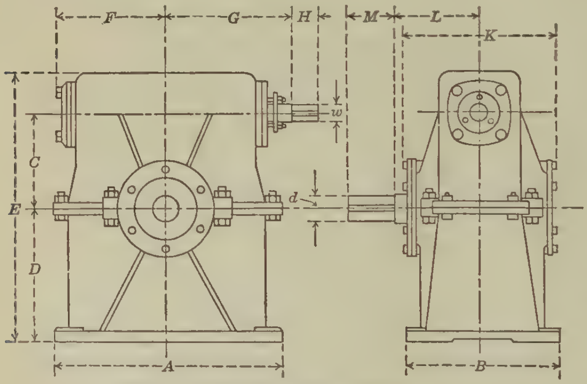


FIG. 3. Worm Gear Speed Reducer

**Worm Gear Reducers** are made in various forms. In general, they comprise an alloy steel worm mating with a phosphor-bronze worm wheel, which may be integral, or in larger sizes may be mounted on an iron or steel center. Splash lubrication generally is employed. As a rule, the reducer is connected to the motor and driven mechanism by means of flexible couplings. Table 4 and Fig. 3 give dimensions of a typical worm gear reducer. Table 6 gives capacities. Where the load is other than a steady one, the service factors in Table 5 should be applied.

## 2. VARIABLE SPEED DRIVES

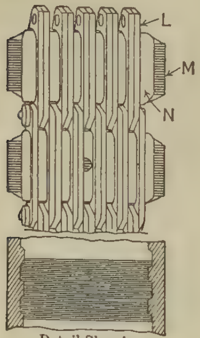
The earliest type of variable speed drive consisted of a pair of similar cones, set with their axes parallel, the large diameter of one cone being alongside the small diameter of the other. A friction ring of flexible material encircled one cone and engaged the surface of the other. One cone being the driver, the speed of the other cone could be varied by moving the friction ring axially. This principle of driving contact with different diameters on two conical surfaces forms the basis of the more modern variable speed drives, some of which are described below.

**THE P. I. V. GEAR** uses a positive chain drive to transmit power. The gear is shown diagrammatically in Fig. 5. It consists of the two pairs of conical discs *A* with radial teeth cut on the conical face, mounted on the splined shafts *B* on which they are free to slide axially. The bearings *C* are carried on levers *D*, which are pivoted at *E* midway

**Table 6.—Capacities of Worm Gear Speed Reducers**

| Size | Ratio | Rev. per Min. of Worm | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 100 | 300 | 600 | 750 | 900 | 1200 | 1500 | 1800 |
| | | Horsepower | | | | | | | |
| 1 | 3.63 | 0.76 | 1.6 | 2.4 | 2.7 | 3.0 | 3.4 | 3.8 | 4.0 |
| | 4.63 | .71 | 1.5 | 2.2 | 2.4 | 2.7 | 3.1 | 3.4 | 3.7 |
| | 6.16 | .43 | 1.1 | 1.8 | 2.1 | 2.4 | 2.8 | 3.1 | 3.4 |
| | 7.25 | .37 | 0.95 | 1.5 | 1.8 | 2.0 | 2.3 | 2.6 | 2.8 |
| | 9.25 | .26 | .76 | 1.3 | 1.5 | 1.8 | 2.1 | 2.3 | 2.5 |
| | 11.75 | .22 | .62 | 1.1 | 1.3 | 1.4 | 1.7 | 1.9 | 2.1 |
| | 14.5 | .16 | .50 | 0.90 | 1.0 | 1.2 | 1.4 | 1.6 | 1.7 |
| | 18.5 | .15 | .41 | .80 | 0.90 | 1.0 | 1.2 | 1.4 | 1.5 |
| | 23.5 | .11 | .33 | .62 | .71 | 0.82 | 0.96 | 1.1 | 1.2 |
| | 29 | .09 | .26 | .49 | .58 | .67 | .76 | 0.90 | 1.0 |
| | 37 | .07 | .20 | .40 | .47 | .51 | .63 | .71 | 0.80 |
| | 47 | .05 | .16 | .30 | .37 | .45 | .54 | .62 | .70 |
| | 52 | .04 | .13 | .26 | .29 | .40 | .50 | .58 | .62 |
| | 58 | .03 | .11 | .22 | .25 | .35 | .42 | .49 | .54 |
| 2 | 3.63 | 6.0 | 11.0 | 17.0 | 20.0 | 22.0 | 23.0 | 24.0 | 25 |
| | 4.50 | 5 | 9 | 15 | 17 | 19 | 21 | 23 | 24 |
| | 5.88 | 3 | 8 | 14 | 15 | 17 | 18 | 20 | 21 |
| | 7.25 | 2.2 | 6.5 | 13 | 14 | 16 | 17 | 19 | 20 |
| | 9 | 1.8 | 5.5 | 10 | 12 | 14 | 16 | 17 | 18 |
| | 11.75 | 1.6 | 5.0 | 9 | 10 | 11 | 13 | 14 | 15 |
| | 14.50 | 1.5 | 4.0 | 7.5 | 9 | 10 | 11 | 12 | 13 |
| | 17.25 | 1.3 | 3.8 | 6.5 | 7.5 | 9 | 9.5 | 10.5 | 11.5 |
| | 18 | 1.2 | 3.6 | 6.0 | 7.0 | 8.5 | 9.0 | 10.0 | 11.0 |
| | 23.50 | 1.0 | 3.0 | 5.0 | 5.5 | 6.0 | 6.5 | 7.5 | 8.0 |
| | 28.50 | 0.80 | 2.7 | 4.5 | 5.0 | 5.5 | 6.0 | 6.3 | 6.5 |
| | 36 | .65 | 2.2 | 3.6 | 4.0 | 4.5 | 5.2 | 5.6 | 6.0 |
| | 47 | .56 | 1.5 | 3.0 | 3.5 | 3.9 | 4.3 | 4.8 | 5.2 |
| | 57 | .38 | 1.2 | 2.2 | 2.6 | 3.1 | 3.5 | 4.0 | 4.4 |
| | 69 | .25 | 0.85 | 1.6 | 2.0 | 2.2 | 2.7 | 3.1 | 3.4 |
| | 77 | .20 | .65 | 1.1 | 1.5 | 1.7 | 2.1 | 2.5 | 2.8 |
| | 87 | .12 | .35 | 0.64 | 0.82 | 0.96 | 1.5 | 1.7 | 1.8 |
| | 98 | .05 | .12 | .37 | .48 | .57 | 0.82 | 0.89 | 0.95 |
| 3 | 3.63 | 33 | 62 | 82 | 88 | 92 | 99 | 106 | 113 |
| | 4.38 | 29 | 52 | 70 | 75 | 80 | 85 | 92 | 98 |
| | 5.50 | 25 | 47 | 65 | 71 | 74 | 80 | 86 | 92 |
| | 7.25 | 20 | 42 | 60 | 66 | 69 | 75 | 81 | 87 |
| | 8.75 | 17 | 39 | 57 | 61 | 66 | 70 | 75 | 80 |
| | 10.75 | 13 | 34 | 52 | 57 | 60 | 65 | 69 | 73 |
| | 13.25 | 11.5 | 28 | 45 | 50 | 53 | 57 | 60 | 63 |
| | 14.50 | 10.0 | 26 | 41 | 45 | 49 | 53 | 56 | 58 |
| | 17.50 | 7.5 | 21 | 33 | 37 | 40 | 46 | 48 | 50 |
| | 21.50 | 6.0 | 19 | 30 | 33 | 36 | 39 | 41 | 42 |
| | 26.50 | 5.0 | 17 | 28 | 32 | 34 | 37 | 39 | 40 |
| | 29 | 4.7 | 16 | 26 | 30 | 32 | 34 | 36 | 37 |
| | 35 | 4.2 | 14 | 23 | 27 | 28 | 30 | 32 | 33 |
| | 43 | 3.2 | 10 | 16 | 18 | 20 | 22 | 23 | 24 |
| | 53 | 2.2 | 8.5 | 12.5 | 14.5 | 16 | 18 | 19 | 20 |
| | 64 | 1.6 | 6.4 | 9.5 | 11.5 | 12.5 | 15 | 15.5 | 16 |
| | 73 | 1.3 | 5.0 | 8.5 | 9.5 | 11.0 | 12.5 | 13.0 | 14 |
| | 87 | 0.92 | 3.0 | 6.0 | 7.0 | 8.0 | 9.0 | 9.5 | 10 |
| | 100 | .37 | 1.0 | 3.5 | 4.0 | 4.7 | 5.3 | 5.6 | 6 |
| 4 | 5.83 | 130 | 253 | 362 | 387 | 400 | 430 | 440 | 450 |
| | 7.13 | 103 | 234 | 352 | 375 | 382 | 418 | 432 | 445 |
| | 9.13 | 73 | 210 | 318 | 345 | 375 | 400 | 415 | 425 |
| | 11.38 | 64 | 193 | 300 | 334 | 357 | 390 | 400 | 410 |
| | 14.25 | 59 | 176 | 282 | 312 | 333 | 358 | 368 | 376 |
| | 18.25 | 47 | 162 | 232 | 273 | 300 | 305 | 310 | 315 |
| | 22.75 | 44 | 151 | 214 | 236 | 253 | 275 | 280 | 287 |
| | 28.50 | 41 | 132 | 191 | 216 | 230 | 238 | 245 | 252 |
| | 36.50 | 31 | 92 | 143 | 156 | 168 | 181 | 186 | 191 |
| | 45.50 | 25 | 75 | 121 | 130 | 138 | 145 | 148 | 151 |
| | 57 | 22 | 65 | 87 | 103 | 111 | 118 | 119 | 121 |
| | 73 | 16 | 44 | 77 | 84 | 90 | 100 | 103 | 106 |
| | 88 | 12 | 34 | 55 | 62 | 68 | 77 | 81 | 84 |
| | 102 | 6 | 20 | 40 | 47 | 52 | 58 | 63 | 66 |

between the bearings of the two conical wheels. A right- and left-hand screw *F* connects the two levers *D*. Rotation of this screw causes the discs *A* on one shaft to approach each other while those on the other shaft recede by a corresponding amount. The two wheels are connected by means of the drive chain *G*, details of which are shown in Fig. 4. The chain is made up of a series of overlapping steel links *L* with hardened steel pin joints. Attached to the wheel side of the links are containers *N*, which hold packs of hardened steel laminations *M* at right angles to the link, projecting about $^1/_8$ in. from either side of the chain. These laminations are free to slide from side to side individually, and adjust themselves to engagement with the radial teeth on the discs *A* over the full range of diameter, as shown in detail in Fig. 5. The teeth are staggered on the opposing faces of each pair of discs.



Detail Showing Action of Links

Fig. 4. P.I.V. Gear Drive Chain

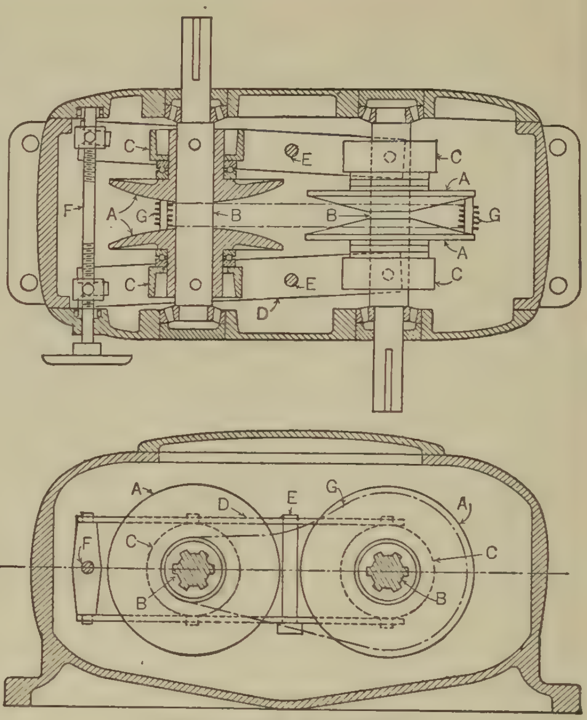


Fig. 5. P.I.V. Gear Principle

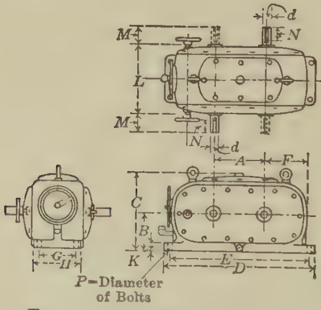


Fig. 6. P.I.V. Horizontal Gear

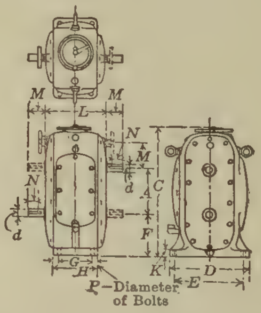


Fig. 7. P.I.V. Vertical Gear

As the distance between the faces of a wheel is increased the links engage with them on a smaller working diameter. The working diameter of the mating wheel is correspondingly increased, thus providing speed variations between the two shafts in any ratio desired up to the limiting ratio of the particular drive in use.

These drives are built both in horizontal and vertical types, for driving by roller or silent chain on the constant-speed shaft. They also are built with a motor integral with the drive and geared to the constant-speed shaft. Another form that is regularly built is provided with gear reducers on either the constant or variable speed shafts, or on both. This gear is built by the Link-Belt Co., Chicago. Dimensions and horsepower capacities are shown in Tables 7 and 8 and Figs. 6 and 7.

**Table 7.—Dimensions of P. I. V. Gears**

(Link-Belt Co., Chicago)

All dimensions in inches. See Figs. 6 and 7

| Size | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *K* | *L* | *M* | *N* | *P* | *d* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Horizontal Type (Fig. 6)** | | | | | | | | | | | | | | |
| 1 | 6 1/2 | 4 1/2 | 9 7/16 | 19 1/2 | 18 | 5 3/8 | 4 1/2 | 6 | 1 | 8 3/8 | 2 1/4 | 1 5/8 | 1/2 | 15/1 |
| 2 | 7 1/2 | 5 1/2 | 11 3/8 | 22 1/4 | 20 1/2 | 6 3/16 | 5 1/2 | 7 1/4 | 1 1/8 | 10 1/8 | 2 3/4 | 2 | 5/8 | 1 1/8 |
| 3 | 9 | 5 1/2 | 13 9/16 | 26 3/4 | 24 3/4 | 7 9/16 | 7 | 9 | 1 3/8 | 12 3/8 | 3 1/2 | 2 3/4 | 3/4 | 1 5/16 |
| 4 | 12 | 8 | 16 11/16 | 33 | 30 3/4 | 8 15/16 | 9 | 11 1/4 | 1 3/4 | 15 1/8 | 4 | 3 1/4 | 7/8 | 1 9/16 |
| 5 | 14 | 9 1/2 | 19 11/16 | 39 | 36 1/2 | 10 3/4 | 11 | 13 1/2 | 2 | 18 5/8 | 4 1/2 | 3 3/4 | 1 | 1 3/4 |
| **Vertical Type (Fig. 7)** | | | | | | | | | | | | | | |
| 1 | 6 1/2 | ..... | 18 1/2 | 11 | 9 1/2 | 6 | 4 1/2 | 6 | 1 | 8 3/8 | 2 1/4 | 1 5/8 | 1/2 | 15/16 |
| 2 | 7 1/2 | ..... | 21 3/8 | 13 1/4 | 11 1/2 | 7 | 5 1/2 | 7 1/4 | 1 1/8 | 10 1/8 | 2 3/4 | 2 | 5/8 | 1 1/8 |
| 3 | 9 | ..... | 25 7/8 | 16 1/2 | 14 1/2 | 8 1/2 | 7 | 9 | 1 3/8 | 12 3/8 | 3 1/2 | 2 3/4 | 3/4 | 1 5/16 |
| 4 | 12 | ..... | 31 7/8 | 20 1/4 | 18 | 10 | 9 | 11 1/4 | 1 3/4 | 15 1/8 | 4 | 3 1/4 | 7/8 | 1 9/16 |
| 5 | 14 | ..... | 37 11/16 | 24 | 21 1/2 | 12 | 11 | 13 1/2 | 2 | 18 5/8 | 4 1/2 | 3 3/4 | 1 | 1 3/4 |

**Table 8.—Speed and Horsepower Ratings of P. I. V. Gears**

(Link-Belt Co., Chicago)

| Size | Ratio | Rating at Normal r.p.m. of Constant-speed Shaft | | | | | | Rating at 100 r.p.m. of Constant-speed Shaft | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Variable-speed shaft maximums | | | Variable-speed shaft maximums | | | Variable-speed shaft maximums | | Variable-speed shaft minimums | |
| | | R.p.m. | Hp. | Torque, in.-lb. | R.p.m. | Hp. | Torque, in.-lb. | R.p.m. | Hp. | R.p.m. | Hp. |
| 3-Hp. Motor. Constant-speed shaft r.p.m., 860 | | | | | | | | | | | |
| 1 | 2 : 1 | 1216 | 2.7 | 140 | 608 | 1.9 | 200 | 141 | 0.314 | 71 | 0.221 |
| | 3 : 1 | 1491 | 2.7 | 115 | 497 | 1.6 | 200 | 173 | .314 | 58 | .186 |
| | 4 : 1 | 1720 | 2.7 | 100 | 430 | 1.4 | 200 | 200 | .314 | 50 | .163 |
| | 5 : 1 | 1925 | 2.7 | 90 | 385 | 1.2 | 200 | 224 | .314 | 45 | .140 |
| | 6 : 1 | 2106 | 2.7 | 80 | 351 | 1.1 | 200 | 245 | .314 | 41 | .128 |
| 5-Hp. Motor. Constant-speed shaft r.p.m., 860 | | | | | | | | | | | |
| 2 | 2 : 1 | 1216 | 4.5 | 235 | 608 | 3.2 | 330 | 141 | 0.523 | 71 | 0.372 |
| | 3 : 1 | 1491 | 4.5 | 190 | 497 | 2.6 | 330 | 173 | .523 | 58 | .302 |
| | 4 : 1 | 1720 | 4.5 | 165 | 430 | 2.3 | 330 | 200 | .523 | 50 | .267 |
| | 5 : 1 | 1925 | 4.5 | 145 | 385 | 2.0 | 330 | 224 | .523 | 45 | .233 |
| | 6 : 1 | 2106 | 4.5 | 135 | 351 | 1.8 | 330 | 245 | .523 | 41 | .210 |
| 7.5-Hp. Motor. Constant-speed shaft r.p.m., 860 | | | | | | | | | | | |
| 3 | 2 : 1 | 1216 | 6.75 | 350 | 608 | 4.8 | 495 | 141 | 0.785 | 71 | 0.558 |
| | 3 : 1 | 1491 | 6.75 | 285 | 497 | 3.9 | 495 | 173 | .785 | 58 | .453 |
| | 4 : 1 | 1720 | 6.75 | 250 | 430 | 3.4 | 495 | 200 | .785 | 50 | .395 |
| | 5 : 1 | 1925 | 6.75 | 220 | 385 | 3.0 | 495 | 224 | .785 | 45 | .349 |
| | 6 : 1 | 2106 | 6.75 | 200 | 351 | 2.8 | 495 | 245 | .785 | 41 | .326 |
| 10-Hp. Motor. Constant-speed shaft r.p.m., 690 | | | | | | | | | | | |
| 4 | 2 : 1 | 976 | 9.0 | 580 | 488 | 6.4 | 820 | 141 | 1.304 | 71 | 0.928 |
| | 3 : 1 | 1194 | 9.0 | 475 | 398 | 5.2 | 820 | 173 | 1.304 | 58 | .754 |
| | 4 : 1 | 1380 | 9.0 | 410 | 345 | 4.5 | 820 | 200 | 1.304 | 50 | .652 |
| | 5 : 1 | 1545 | 9.0 | 365 | 309 | 4.0 | 820 | 224 | 1.304 | 45 | .580 |
| | 6 : 1 | 1692 | 9.0 | 335 | 282 | 3.7 | 820 | 245 | 1.304 | 41 | .536 |
| 15-Hp. Motor. Constant-speed shaft r.p.m., 580 | | | | | | | | | | | |
| 5 | 2 : 1 | 820 | 13.5 | 1035 | 410 | 9.5 | 1470 | 141 | 2.328 | 71 | 1.638 |
| | 3 : 1 | 1005 | 13.5 | 845 | 335 | 7.8 | 1470 | 173 | 2.328 | 58 | 1.345 |
| | 4 : 1 | 1160 | 13.5 | 735 | 290 | 6.8 | 1470 | 200 | 2.328 | 50 | 1.172 |
| | 5 : 1 | 1295 | 13.5 | 655 | 259 | 6.0 | 1470 | 224 | 2.328 | 45 | 1.035 |
| | 6 : 1 | 1422 | 13.5 | 600 | 237 | 5.5 | 1470 | 245 | 2.328 | 41 | 0.948 |

**THE REEVES VARIABLE-SPEED TRANSMISSION** (Figs. 8 and 10) consists of two pairs of cone-shaped discs mounted on spline shafts. The discs are carried in bearings on the levers *V*, pivoted at *P*. The hand wheel *W* actuates the right- and left-hand screw *S* causing the levers to rotate about *P*, and thus causing the discs of one pair to approach and of the other pair to recede from each other as desired. One shaft is connected to the driving motor, or other prime mover. The other is connected to the driven mechanism. A rubber belt, with blocks of hard wood bolted to each face, connects the two pairs of discs. See Fig. 9. The ends of the blocks are finished at the same angle as that of the discs. The driving force is the friction between the blocks and the disc

**Table 9.—Sizes and Capacities of Reeves Variable Speed Transmissions**

| H.P. Required | Class | V | S | R | A | B | C | D | E | F | G |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Variation | 16:1 | 14:1 | 12:1 | 10:1 | 8:1 | 6:1 | 5:1 | 4:1 | 3:1 | 2:1 |
| | Factor $M$ | 4.00 | 3.74 | 3.46 | 3.16 | 2.83 | 2.45 | 2.24 | 2.00 | 1.73 | 1.41 |
| 1/8 to 1/4 | Size | .... | .... | .... | .... | .... | 0000 | 0000 | 0000 | 0000 | 0000 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | .... | 700 | 700 | 750 | 750 | 750 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | .... | 1715 | 1568 | 1500 | 1298 | 1058 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | .... | 286 | 313 | 375 | 433 | 529 |
| 1/2 to 1 | Size | .... | .... | .... | .... | .... | 000 | 000 | 000 | 000 | 000 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | .... | 600 | 600 | 600 | 600 | 600 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | .... | 1470 | 1345 | 1200 | 1038 | 846 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | .... | 245 | 269 | 300 | 346 | 423 |
| 1 1/2 | Size | .... | .... | .... | .... | 000 | 000 | 000 | 000 | 000 | 000 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 550 | 625 | 625 | 625 | 625 | 625 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 1557 | 1530 | 1400 | 1250 | 1082 | 880 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 195 | 255 | 280 | 312 | 394 | 440 |
| 2 | Size | .... | .... | .... | 00 | 00 | 000 | 000 | 000 | 000 | 000 |
| | R.p.m. constant speed shaft | .... | .... | .... | 550 | 550 | 650 | 650 | 650 | 650 | 650 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | 1738 | 1557 | 1592 | 1456 | 1300 | 1125 | 916 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | 174 | 195 | 265 | 290 | 325 | 375 | 458 |
| 3 | Size | .... | .... | 1 | 1 | 0 | 00 | 00 | 00 | 00 | 00 |
| | R.p.m. constant speed shaft | .... | .... | 450 | 450 | 500 | 600 | 600 | 600 | 600 | 600 |
| | Max. r.p.m. variable speed shaft | .... | .... | 1557 | 1422 | 1415 | 1470 | 1345 | 1200 | 1038 | 846 |
| | Min. r.p.m. variable speed shaft | .... | .... | 130 | 142 | 177 | 245 | 269 | 300 | 346 | 423 |
| 5 | Size | .... | .... | 2 | 2 | 1 | 0 | 0 | 0 | 0 | 0 |
| | R.p.m. constant speed shaft | .... | .... | 425 | 425 | 480 | 550 | 550 | 550 | 550 | 550 |
| | Max. r.p.m. variable speed shaft | .... | .... | 1471 | 1343 | 1358 | 1348 | 1232 | 1100 | 952 | 775 |
| | Min. r.p.m. variable speed shaft | .... | .... | 123 | 134 | 170 | 224 | 246 | 275 | 317 | 387 |
| 7 1/2 | Size | .... | .... | 3 | 3 | 2 | 1 | 1 | 1 | 1 | 1 |
| | R.p.m. constant speed shaft | .... | .... | 400 | 400 | 450 | 500 | 500 | 500 | 500 | 500 |
| | Max. r.p.m. variable speed shaft | .... | .... | 1384 | 1264 | 1274 | 1225 | 1120 | 1000 | 865 | 705 |
| | Min. r.p.m. variable speed shaft | .... | .... | 115 | 126 | 159 | 204 | 224 | 250 | 288 | 352 |
| 10 | Size | .... | .... | 4 | 4 | 3 | 2 | 2 | 2 | 2 | 2 |
| | R.p.m. constant speed shaft | .... | .... | 375 | 375 | 425 | 480 | 480 | 480 | 480 | 480 |
| | Max. r.p.m. variable speed shaft | .... | .... | 1297 | 1185 | 1202 | 1176 | 1075 | 960 | 830 | 677 |
| | Min. r.p.m. variable speed shaft | .... | .... | 108 | 119 | 150 | 196 | 215 | 240 | 277 | 339 |
| 15 | Size | 5 | 5 | 5 | 5 | 4 | 4 | 4 | 3 | 3 | 3 |
| | R.p.m. constant speed shaft | 300 | 300 | 350 | 350 | 400 | 400 | 400 | 450 | 450 | 450 |
| | Max. r.p.m. variable speed shaft | 1200 | 1122 | 1211 | 1106 | 1132 | 980 | 896 | 900 | 778 | 635 |
| | Min. r.p.m. variable speed shaft | 75 | 80 | 101 | 111 | 141 | 163 | 179 | 225 | 259 | 317 |
| 20 | Size | 6 | 6 | 6 | 6 | 5 | 5 | 5 | 4 | 4 | 4 |
| | R.p.m. constant speed shaft | 275 | 275 | 300 | 300 | 375 | 375 | 375 | 425 | 425 | 425 |
| | Max. r.p.m. variable speed shaft | 1100 | 1028 | 1038 | 948 | 1061 | 919 | 840 | 850 | 735 | 599 |
| | Min. r.p.m. variable speed shaft | 69 | 74 | 87 | 95 | 132 | 153 | 168 | 213 | 245 | 299 |
| 25 | Size | .... | .... | .... | .... | 6 | 6 | 6 | 5 | 5 | 5 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 300 | 300 | 330 | 400 | 400 | 400 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 849 | 735 | 740 | 800 | 692 | 564 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 106 | 123 | 148 | 200 | 231 | 284 |
| 30 | Size | .... | .... | .... | .... | 6 1/2 | 6 1/2 | 6 1/2 | 6 1/2 | 6 | 6 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 250 | 250 | 300 | 300 | 350 | 350 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 707 | 612 | 672 | 600 | 605 | 493 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 88 | 102 | 134 | 150 | 202 | 246 |
| 40 | Size | .... | .... | .... | .... | 7 | 7 | 7 | 6 1/2 | 6 1/2 | 6 1/2 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 250 | 250 | 270 | 300 | 330 | 330 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 707 | 612 | 605 | 600 | 570 | 465 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 88 | 102 | 121 | 150 | 190 | 232 |
| 50 | Size | .... | .... | .... | .... | 8 | 8 | 8 | 7 | 7 | 7 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 200 | 200 | 220 | 270 | 290 | 290 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 566 | 490 | 494 | 540 | 502 | 409 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 71 | 81 | 99 | 135 | 167 | 204 |
| 60 | Size | .... | .... | .... | .... | 9 | 9 | 8 | 8 | 8 | 8 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 180 | 180 | 220 | 220 | 250 | 250 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 509 | 441 | 494 | 440 | 432 | 353 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 64 | 73 | 99 | 110 | 144 | 176 |
| 75 | Size | .... | .... | .... | .... | 10 | 10 | 9 | 9 | 8 | 8 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | 160 | 160 | 200 | 200 | 250 | 250 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | 453 | 392 | 448 | 400 | 432 | 353 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | 57 | 65 | 89 | 100 | 144 | 176 |
| 100 | Size | .... | .... | .... | .... | .... | 10 | 10 | 10 | 9 | 9 |
| | R.p.m. constant speed shaft | .... | .... | .... | .... | .... | 160 | 180 | 180 | 220 | 220 |
| | Max. r.p.m. variable speed shaft | .... | .... | .... | .... | .... | 392 | 403 | 360 | 380 | 310 |
| | Min. r.p.m. variable speed shaft | .... | .... | .... | .... | .... | 65 | 80 | 90 | 126 | 155 |

faces. When the discs of one pair are widely separated, the diameter of the circle of contact with the blocks is small, while it is correspondingly large on the other pair. By varying the distance between the discs any desired speed ratio between the driving and driven shafts can be obtained, within the range of the transmission.

These transmissions are built by the Reeves Pulley Co., Columbus, Ind., in both horizontal and vertical types, and both open and enclosed. Capacities range up to 100 Hp. It should be remembered that the horsepower transmitted varies with the speed of the driven shaft. A transmission that will deliver the requisite horsepower at the maximum speed, may be unable to deliver it at the lowest speed, and a larger drive is indicated. Table 9 gives data for the selection of the proper size of transmission for a

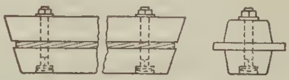


FIG. 9. Reeves Variable Speed Transmission Belt

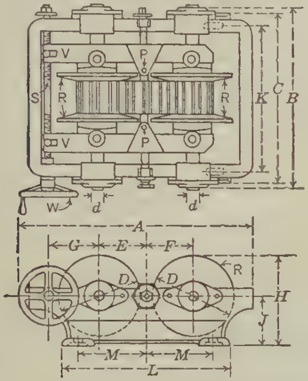


FIG. 8. Reeves Variable Speed Transmission, Horizontal Type

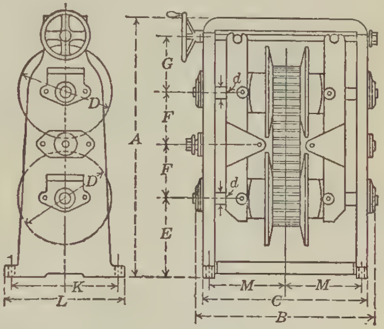


FIG. 10. Reeves Variable Speed Transmission Vertical Type

**Table 10.—Dimensions of Open-type Reeves Variable-speed Transmissions**

(Reeves Pulley Co., Columbus, Ind.)

| Size | Dimensions, in. | | | | | | | | | | | | Shaft, d | Bolts |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | A | B | C | D | E | F | G | H | J | K | L | M | | |
| **Horizontal Type (Fig. 8)** | | | | | | | | | | | | | | |
| 0000 | 13 1/2 | 11 3/4 | 10 1/2 | 4 1/2 | 4 | 2 3/4 | 2 7/8 | 5 1/8 | 2 15/16 | 9 | 9 3/4 | 3 7/8 | 11/16 | 5/16 |
| 000 | 17 15/16 | 14 1/2 | 14 | 6 7/8 | 4 13/16 | 3 3/4 | 4 1/16 | 7 3/8 | 3 15/16 | 12 | 13 1/4 | 5 1/16 | 1 3/16 | 3/8 |
| 00 | 22 13/16 | 15 1/8 | 14 3/4 | 8 7/8 | 6 | 5 | 5 1/16 | 9 5/8 | 5 3/16 | 12 1/2 | 16 1/4 | 6 1/2 | 1 3/16 | 3/8 |
| 0 | 30 1/2 | 22 1/4 | 21 1/4 | 11 1/8 | 8 1/16 | 6 1/2 | 6 3/4 | 12 1/2 | 6 15/16 | 18 1/4 | 19 1/2 | 7 13/16 | 1 7/16 | 1/2 |
| 1 | 34 11/16 | 24 1/2 | 23 1/2 | 13 1/4 | 9 3/16 | 7 1/2 | 7 13/16 | 14 9/16 | 7 15/16 | 20 1/2 | 21 1/2 | 8 13/16 | 1 7/16 | 1/2 |
| 2 | 42 1/4 | 31 1/8 | 30 1/2 | 16 3/16 | 11 1/4 | 9 | 9 1/2 | 18 1/8 | 10 | 26 1/2 | 27 5/8 | 11 1/16 | 1 15/16 | 5/8 |
| 3 | 49 5/8 | 32 | 31 7/8 | 19 11/16 | 13 | 11 | 11 1/8 | 21 3/4 | 11 7/8 | 27 7/8 | 32 1/2 | 13 1/4 | 1 15/16 | 5/8 |
| 4 | 56 1/16 | 37 5/8 | 37 5/8 | 22 3/4 | 14 9/16 | 12 1/2 | 12 3/4 | 25 3/8 | 14 | 32 3/8 | 35 1/4 | 14 3/8 | 2 3/16 | 3/4 |
| 5 | 62 3/4 | 41 7/8 | 42 3/8 | 25 5/8 | 16 3/8 | 14 | 14 3/8 | 28 11/16 | 15 7/8 | 36 3/8 | 38 1/4 | 15 7/8 | 2 3/16 | 7/8 |
| 6 | 79 1/4 | 51 1/8 | 52 1/8 | 32 1/2 | 20 1/2 | 18 | 18 | 36 1/4 | 20 | 45 1/8 | 48 1/2 | 20 1/2 | 2 3/4 | 7/8 |
| 6 1/2 | 79 1/4 | 51 1/8 | 52 1/8 | 32 1/2 | 20 1/2 | 18 | 18 | 36 1/4 | 20 | 45 1/8 | 48 1/2 | 20 1/2 | 2 3/4 | 7/8 |
| 7 | 98 1/4 | 59 1/4 | 59 1/2 | 39 | 26 1/8 | 23 | 21 | 42 1/2 | 23 | 52 | 61 | 25 1/2 | 3 3/16 | 1 |
| 8 | 111 15/16 | 82 1/4 | 68 | 42 | 23 3/4 | 28 | 27 3/4 | 50 | 29 | 60 | 77 | 10 1/2 | 3 7/16 | 7/8 |
| 9 | 129 | 96 3/4 | 80 1/8 | 50 | 29 | 31 | 33 1/2 | 58 | 33 | 70 1/8 | 87 | 12 1/2 | 3 15/16 | 1 |
| 10 | 151 1/4 | 113 1/2 | 94 1/2 | 60 | 34 | 36 | 39 5/8 | 68 | 38 | 82 1/2 | 102 | 15 | 4 7/16 | 1 |
| **Vertical Type (Fig. 10)** | | | | | | | | | | | | | | |
| 0000 | 13 9/16 | 11 3/4 | 9 3/4 | 4 1/2 | 4 1/16 | 2 3/4 | 2 7/8 | ........ | ........ | 5 7/8 | 7 | 4 17/32 | 11/16 | 5/16 |
| 000 | 19 1/8 | 14 1/2 | 13 | 6 7/8 | 5 11/16 | 3 3/4 | 4 1/16 | ........ | ........ | 8 | 9 5/8 | 6 | 1 3/16 | 3/8 |
| 00 | 24 1/8 | 15 1/8 | 14 | 8 3/4 | 6 7/8 | 5 | 5 1/16 | ........ | ........ | 11 | 13 9/16 | 6 1/4 | 1 3/16 | 3/8 |
| 0 | 31 11/16 | 22 1/4 | 20 1/4 | 11 1/8 | 9 9/16 | 6 1/2 | 6 3/4 | ........ | ........ | 13 | 14 1/2 | 9 1/8 | 1 7/16 | 1/2 |
| 1 | 35 11/16 | 24 1/2 | 22 1/4 | 13 1/4 | 10 3/8 | 7 1/2 | 7 13/16 | ........ | ........ | 15 | 17 | 10 1/4 | 1 7/16 | 1/2 |
| 2 | 43 3/4 | 31 1/8 | 28 3/4 | 16 3/16 | 13 1/8 | 9 | 9 1/2 | ........ | ........ | 19 | 21 | 13 1/4 | 1 15/16 | 5/8 |
| 3 | 51 | 32 | 29 7/8 | 19 11/16 | 14 3/4 | 11 | 11 1/8 | ........ | ........ | 22 3/4 | 25 | 13 11/16 | 1 15/16 | 5/8 |
| 4 | 58 1/8 | 37 5/8 | 34 7/8 | 22 3/4 | 16 5/8 | 12 1/2 | 12 3/4 | ........ | ........ | 27 1/2 | 30 1/2 | 16 3/16 | 2 3/16 | 3/4 |
| 5 | 65 1/8 | 41 7/8 | 39 1/8 | 25 5/8 | 19 | 14 | 14 3/8 | ........ | ........ | 31 1/2 | 34 | 18 3/16 | 2 3/16 | 7/8 |
| 6 | 81 1/4 | 51 1/8 | 48 1/8 | 32 1/2 | 23 | 18 | 18 | ........ | ........ | 39 | 42 | 22 9/16 | 2 3/4 | 1 |
| 6 1/2 | 81 1/4 | 51 1/8 | 48 1/8 | 32 1/2 | 23 | 18 | 18 | ........ | ........ | 39 | 42 | 22 9/16 | 2 3/4 | 1 |

given set of conditions. The table is entered opposite the horsepower that is to be transmitted. In the column under the desired speed ratio is the size of the drive recommended, and also the speed of the constant-speed shaft, and the maximum and minimum speeds of the variable speed shaft. The factor $M$ is used to determine the maximum and minimum speeds of the variable speed shaft, when the constant-speed shaft runs at other than the recommended speed. If $N$ = r.p.m. of constant-speed shaft, and $n_h$, $n_l$ = respectively the maximum and minimum speeds of the variable speed shaft, $n_h = N \times M$; $n_l = N/M$.

Table 10 gives dimensions of open-type transmissions.

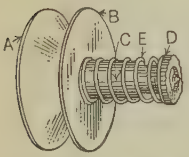


Fig. 11. Reeves Vari-speed Pulley

**The Reeves Vari-speed Pulley** is designed to replace the ordinary flat pulley on motors up to 7 1/2 Hp., to vary the speed of the driven mechanism. It operates on the same principle as the Reeves variable speed drive (see p. 24–75). The pulley (Fig. 11) consists of two discs $A$ and $B$ mounted on a shaft $C$, disc $B$ being free to move axially on the shaft. Its distance from disc $A$ can be regulated by the hand nut $D$. The spring $E$ holds it against the belt with a constant tension. The edges of the belt, which are beveled to the same angle as the discs, bear against the discs. The flat side of the belt bears on face of the driven pulley. As the distance between the discs is increased the working diameter of the belt on the discs is decreased, thus altering the speed ratio. Table 11 gives dimensions and capacities.

**THE GRAHAM VARIABLE SPEED TRANSMISSION** is based on the principle of a ring of fixed diameter in contact with conical rollers, the speed ratio depending on the ratio of the diameter of the ring to the diameter of the rollers at the point of contact. Fig.12 is a diagrammatic arrangement of the drive. Three conical rollers $A$

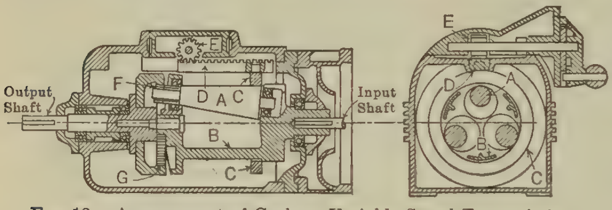


Fig. 12. Arrangement of Graham Variable Speed Transmission

**Table 11.—Horsepower and Dimensions of Reeves Vari-speed Pulley**

(Reeves Pulley Co., Columbus, Ind.)

| Motor r.p.m. | 1750 | | 1750 | | 1750 | | 1160 | | 1160 | | 1160 | 800 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Hp. | 1/8 | 1/4 | 1/3 | 3/4 | 1 | 1 1/2 | 1 | 1 1/2 | 2 | 3 | 5 | 7 1/2 |
| Min. pulley diam. | 5 | 6 | 6 | 8 | 10 | 12 | 8 | 10 | 10 | 12 | 14 | 16 |
| Ratio | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 3 : 1 | 2 3/4 : 1 | 2 3/4 : 1 |
| Max. bore | 1/2 | 1/2 | 3/4 | 3/4 | 1 | 1 | 1 | 1 | 1 1/8 | 1 1/8 | 1 7/16 | 1 5/8 |
| Disc working diam. | | | | | | | | | | | | |
| Max. | 5 5/8 | 5 5/8 | 7 1/8 | 7 1/8 | 7 3/8 | 7 3/8 | 10 | 10 | 11 1/4 | 11 1/4 | 11 3/4 | 13 3/4 |
| Min. | 1 7/8 | 1 7/8 | 2 3/8 | 2 3/8 | 2 7/16 | 2 7/16 | 3 1/4 | 3 1/4 | 3 3/4 | 3 3/4 | 4 1/4 | 5 |

**Table 12.—Speeds and Capacities of Graham Variable Speed Transmissions**

(Graham Transmissions, Springfield, Vt.)

| Size | Output Shaft | 1800 r.p.m. Motor | | | 1200 r.p.m. Motor | | | Max. Torque, in.-lb. |
|---|---|---|---|---|---|---|---|---|
| | | Max. Speed | Mean Speed * | Min. Speed | Max. Speed | Mean Speed * | Min. Speed | |
| 81E | R.p.m. | 680 | 340 | 60 ‡ | 450 | 220 | 40 ‡ | 150 |
| | Hp. | 1.5 | 1 | ......... | 1 | 0.6 | ......... | 150 |
| 81N | R.p.m. | 830 | 500 | 200 | 550 | 340 | 130 | 150 |
| | Hp. | 2 † | 1.4 | 0.7 | 1.5 | 1 | 0.5 | 150 |
| 83E | R.p.m. | 680 | 340 | 60 ‡ | 450 | 220 | 40 ‡ | 450 |
| | Hp. | 4 | 2.5 | ......... | 2.5 | 1.5 | ......... | 450 |
| 83N | R.p.m. | 830 | 500 | 200 | 550 | 340 | 130 | 450 |
| | Hp. | 5 † | 3 | 1.5 | 3.5 | 2 | 1 | 450 |
| 85E | R.p.m. | 680 | 340 | 60 ‡ | 450 | 220 | 40 ‡ | 750 |
| | Hp. | 6 | 4 | ......... | 4 | 2.5 | ......... | 750 |
| 85N | R.p.m. | 830 | 500 | 200 | 550 | 340 | 130 | 750 |
| | Hp. | 7.5 † | 5 | 2.3 | 5 | 3 | 1.5 | 750 |

* Mean speed is a single typical speed between maximum and minimum.

† Indicates the largest size of squirrel-cage motor within the rating of the unit.

‡ In reverse.

are carried in the carrier $B$, spaced 120 deg. apart. The rollers $A$ bear with fixed pressure against the contact ring $C$ which surrounds the carrier. The position of $C$ can be varied axially by means of the rack $D$ and the pinion $E$. The carrier $B$ is attached to the drive or input shaft. The rollers $A$ carry, at their small ends, pinions $F$ which mesh with an internal gear $G$, attached to the driven or output shaft. In operation, as the carrier $B$ is rotated, it causes the rollers $A$ to revolve around the center-line of the carrier. The friction between the rollers and the contact ring $C$ causes the rollers to rotate about their own axes, and in turn to drive the output shaft through the pinions and gear $F$ and $G$. The rate of rotation of the rollers, and consequently the speed of the output shaft, depends on the position of the contact ring $C$. The speed of the driven shaft is lowest when the ring is in contact with the smallest diameter of the rollers, and highest when it is in contact with the largest diameter. An indicator on the front of the case shows the position of the contact ring, and the speed ratio at which the drive is operating.

Table 12 gives the capacity of drives up to 7 1/2 Hp. and the maximum and minimum speeds. In the Type $E$ drive, the speed variation is from maximum through zero and into reverse. In the type $N$ drive, the speed variation range is about 4 : 1. The drive may be operated by a built-in motor, or motor coupled to the input shaft. It also is made with a built-in geared head on the output end which reduces the speed of the output shaft to 1/7 of the values given in Table 12, and increases the torque 7 times. This drive is made by Graham Transmissions, Springfield, Vt. Table 13 and Fig. 13 give dimensions.

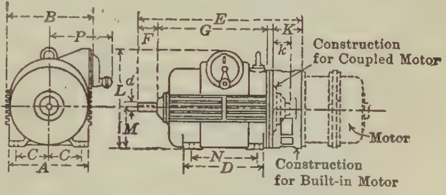

B
P
E
G
K
F
k
d
L
H
M
C
C
A
N
D
Construction for Coupled Motor
Motor
Construction for Built-in Motor

Fig. 13. Dimensions of Graham Variable Speed Transmissions

**Table 13.—Dimensions of Graham Variable Speed Transmissions**

(Graham Transmissions, Springfield, Vt.)

| Size | Dimensions, in. (Fig. 13.) | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *K* | *k* | *L* | *M* | *N* | *P* | *d* |
| 81 | 9 1/4 | 9 7/8 | 4 | 9 1/2 | 18 | 2 3/8 | 12 1/2 | 1/2 | 2 5/8 | 2 1/4 | 11 5/16 | 5 | 7 1/2 | 7 1/4 | 1 |
| 83 | 10 3/4 | 11 1/2 | 4 1/2 | 12 | 23 1/2 | 2 5/8 | 16 7/16 | 7/8 | 3 9/16 | 2 1/2 | 14 1/4 | 5 1/2 | 9 1/2 | 8 15/16 | 1 1/8 |
| 85 | 12 1/4 | 13 | 5 | 13 | 25 1/16 | 3 1/4 | 17 1/4 | 7/8 | 4 | 2 1/2 | 15 3/8 | 6 1/4 | 10 1/2 | 9 1/2 | 1 1/4 |

## 3. HYDRAULIC DRIVES

**THE HYDRAULIC COUPLING**, built by the Hydraulic Coupling Corporation, Detroit, is a development of the Vulcan Sinclair fluid transmission. It is used as a variable speed transmission between centrifugal fans and the driving motor. The principle of operation is shown in Fig. 14. The vaned impeller $A$ is attached to the driving shaft of the motor. The runner $B$, similarly vaned, surrounded by an inner casing $C$, is attached to the driven shaft, which may be the fan shaft or the shaft of any other mechanism. The passages $D$ in runner and impeller correspond, and form a continuous circuit or vortex ring between the two elements. The system is filled with oil, which, under the influence of centrifugal force, is discharged from the impeller against the runner, which causes the runner to rotate. The speed of the runner, when the system is filled with oil, is from 1% to 3% less than that of the impeller. This slip sets up a vortex in the vortex ring, pumping the oil around as shown by the arrows, and thus maintains the torque.

Oil leaking from between the runner and impeller flows into the passage between the runner and the inner casing, and is discharged through calibrated ports $E$ into the space between the inner casing and the outer casing $K$. This oil is picked up by the scoop tube $G$ and returned through ports into the vortex ring as shown by the arrows. Speed variation is effected by varying the amount of oil in the system. This is accomplished by the pump $H$, which removes a portion of the oil as it passes the pump connection, and stores it in tank $K$. Maximum speed

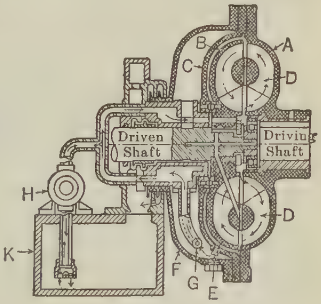


Fig. 14. Principle of Hydraulic Coupling

is obtained with the system full of oil. With all oil removed from the system and stored in the tank a minimum of power is transmitted, the active fluid then being air contained in the vortex ring. To increase the speed of the driven shaft, the pump *H* is reversed and oil is restored to the system. Speed may be infinitely varied from maximum to minimum by varying the amount of oil in the vortex. Instead of the storage tank *K*, in which the supply and removal of oil is effected by a pump, an elevated tank is used for certain types of service. Oil picked up by the scoop tube is led to the elevated tank. It is returned to the coupling by gravity through a calibrated weir, which can be raised or lowered manually or automatically in accordance with the speed control desired.

The heat generated inside the coupling under moderate speed, and load conditions usually is dissipated by radiation from the surface of the coupling. Under heavy loads at high speeds, or when $S\,(V/1000)^3 > 600$, an oil cooler is necessary. $S$ = slip, and $V$ = peripheral velocity.

The efficiency with 3% slip is about 96.5%, which includes windage and heat loss. For average conditions of operation, a slip of about 2% usually is most satisfactory. Fig. 15 shows the brake horsepower output of various sizes of couplings at various motor speeds, and also at various tip speeds based on inside of diameter of coupling. This chart is based on a coupling filled with oil.

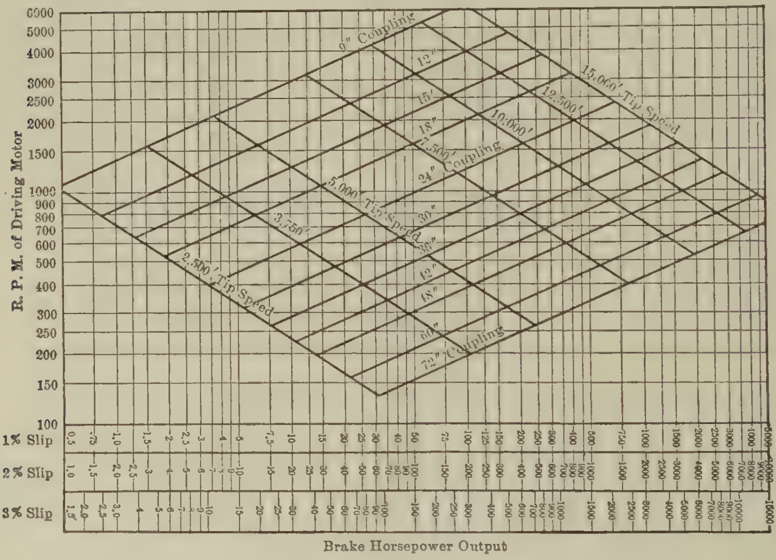


FIG. 15. Brake Horsepower Output of Hydraulic Couplings

To select a coupling, enter Fig. 15 with the required horsepower at the assumed slip. At the intersection of this ordinate with the abscissa representing motor speed, find the size coupling required. The coupling is selected for the maximum rating of the fan. For other ratings, the procedure is as follows: Calculate the constants $K_c$, $K_r$ and $K_p$ representing, respectively, cu. ft. per min. delivered by the fan, rev. per min. of the fan, and resistance pressure at full motor speed.

$K_c = Q/(100-S)$; $K_r = N/(100-S)$; $K_p = P/(100-S)^2$, where $Q$ = cu. ft. per min. at full motor speed; $S$ = slip; $N$ = r.p.m. of fan; $P$ = resistance pressure, in. of water. Then if $Q_1$, $N_1$ and $P_1$ = respectively, the cu. ft. per min. rev. per min., and resistance pressure at any speed other than full speed, and $n$ = fan speed, expressed as a percentage of full speed.

$$Q_1 = K_c \times n;\ \ N_1 = K_r \times n;\ \ P_1 = K_p \times n^2.$$

Motor Hp. = Fan B.Hp. × *f*, in which *f* is a factor whose value is as follows:

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Fan speed, % of constant motor speed = | 100–S | 90 | 80 | 70 | 60 | 50 | 40 | 30 | 20 |
| *f* = | 1.04 | .885 | .70 | .545 | .405 | .282 | .182 | .105 | .052 |

**Example.**—Select a coupling for a fan delivering 140,000 cu. ft. per min. at 14 in. resistance pressure; fan speed 859 r.p.m.; brake horsepower required 478. Determine the fan r.p.m., resistance pressure, and B.Hp. at 90%, 70%, 50% and 30% of maximum fan speed.

*Solution.*—From Fig. 15, a No. 36 coupling will be satisfactory with a slip of 1.2%. Then

$$K_c = 140{,}000/(100 - 1.2) = 142{,}500$$
$$K_r = 859/(100 - 1.2) = 875 = \text{motor speed}$$
$$K_p = 14/(100 - 1.2)^2 = 14.5$$

| | | | | |
|---|---|---|---|---|
| $n =$ | 0.90 | 0.70 | 0.50 | 0.30 |
| $f =$ | .885 | .545 | .282 | .105 |
| $Q_1 = K_c \times n =$ | 128,000 | 100,000 | 71,400 | 42,800 |
| $N_1 = K_r \times n =$ | 787 | 612 | 437 | 262 |
| $P_1 = K_p \times n^2 =$ | 11.8 | 7.1 | 3.6 | 1.3 |
| $\text{B.Hp.}_1 = \text{B.Hp.}_{\text{max.}} \times f =$ | 423 | 261 | 135 | 50 |

Dimensions of standard hydraulic couplings are given in Table 14 and Fig. 16.

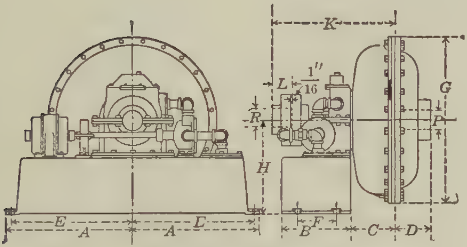


Fig. 16. Dimensions of Hydraulic Couplings

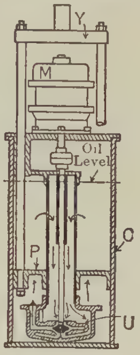


Fig. 17. Thrustor

**Table 14.—Dimensions of Hydraulic Couplings**

(Hydraulic Coupling Corp., Detroit)

| Size, in. | Dimensions, in. See Fig. 16 | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *K* | *L* | *P* | *R* |
| 9 | 10 | 5 1/2 | 3 1/2 | 3 1/2 | 9 3/8 | 3 1/2 | 13 | 7 | 12 1/16 | 2 1/16 | 1 5/8 | 1 1/2 |
| 12 | 12 1/4 | 7 5/8 | 4 3/4 | 4 | 11 5/8 | 5 1/8 | 16 1/2 | 9 | 14 1/16 | 2 7/16 | 2 | 1 15/16 |
| 15 | 17 | 9 3/4 | 5 3/4 | 4 1/2 | 16 1/8 | 6 | 20 1/4 | 11 | 16 11/16 | 2 11/16 | 2 5/8 | 2 3/16 |
| 18 | 18 1/2 | 10 1/4 | 6 3/8 | 5 | 17 5/8 | 6 1/4 | 23 | 12 1/2 | 19 3/16 | 3 9/16 | 2 7/8 | 2 15/16 |
| 24 | 25 1/2 | 14 1/2 | 8 1/2 | 6 1/2 | 24 5/8 | 9 | 30 | 17 | 23 1/16 | 3 9/16 | 2 7/8 | 2 15/16 |
| 30 | 28 | 16 1/4 | 10 | 8 | 26 7/8 | 9 1/4 | 37 3/8 | 21 | 27 9/16 | 4 7/16 | 4 3/8 | 3 15/16 |
| 36 | 38 1/4 | 21 1/2 | 12 1/2 | 9 1/2 | 37 1/8 | 13 | 44 | 25 | 31 7/16 | 4 7/16 | 4 3/8 | 3 15/16 |
| 42 | 44 3/4 | 25 | 14 1/2 | 10 3/4 | 43 1/2 | 15 | 51 | 29 1/4 | 36 13/16 | 5 13/16 | 4 15/16 | 4 15/16 |
| 48 | 52 | 28 3/4 | 16 3/4 | 12 1/4 | 50 3/4 | 17 1/4 | 58 | 33 1/2 | 40 13/16 | 5 13/16 | 4 15/16 | 4 15/16 |

**THE THRUSTOR,** while not a speed changing device, is a hydraulic mechanism to produce thrust in a straight-line direction. The principle of the thrustor is shown in Fig. 17. A piston *P* contained in a cylinder *C* filled with oil, is connected by two piston rods to the yoke *Y*. A motor *M* drives the centrifugal pump *U* at the bottom of the cylinder, setting up a pressure on the under side of the piston and causing it to rise. Oil is drawn by the pump from the upper part of the cylinder through the tube *C*. When the motor is stopped, the weight of the piston causes it to descend, the oil flowing back through the pump to the upper part of the cylinder.

The thrustor is made by the General Electric Co. It is used to control brakes and clutches on punch and drawing presses, to operate arbor presses, to apply pressure to the work in spot welders, to operate brakes on various types of equipment, to operate valves, to open and close doors on furnaces, elevators, etc. It is used wherever straight line motion is required, or pressure is to be applied. In connection with a pawl and ratchet it can be used to bar engines or to give practically continuous rotary motion. By means of appropriate linkwork, the direction of motion of the thrustor can be changed, transformed or multiplied as desired.

Capacities of thrustors range from 50 lb. thrust with a 2-in. stroke to 3200 lb. thrust with a 16-in. stroke. The pressure varies as the square of the speed of the motor. The speed of movement depends on the load.

# DYNAMOMETERS

Dynamometers are instruments used for measuring power. They are of several classes, as: 1. Traction dynamometers, used for determining the power required to pull a car or other vehicle, or a plow or harrow. 2. Brake or absorption dynamometers, in which the power of a rotating shaft or wheel is absorbed or converted into heat by the friction of a brake or into electrical energy, and 3. Transmission dynamometers, in which the power in a rotating shaft is measured during its transmission through a belt or other connection to another shaft, without being absorbed.

**TRACTION DYNAMOMETERS** generally contain two principal parts: 1. A spring or series of springs, through which the pull is exerted, the extension of the spring measuring the pulling force; 2. A paper-covered drum, rotated either at a uniform speed by clockwork, or at a speed proportional to the speed of the traction, through gearing, on which the extension of the spring is registered by a pencil. The average height of the diagram above the zero-line gives the average pulling force in pounds. This multiplied by the distance traversed, in feet, gives the work done, in foot-pounds. The product divided by the time in minutes and by 33,000 gives the horsepower.

**THE PRONY BRAKE** is the typical form of absorption dynamometer. Primarily this consists of a lever so connected to a revolving shaft or pulley that the friction induced between the surfaces in contact will tend to rotate the arm in the direction in which the shaft revolves. This rotation is counterbalanced by a weight-measuring apparatus, as a platform scale, one or more spring balances, or simply by weights hung in a pan at the end of the lever arm attached to the friction surfaces. See Fig. 1. The friction blocks *B* are of wood, and are confined by the strap which can be tightened on them, to increase the friction on the wheel, by means of the bolt *C*. The lever *D* is bolted to the band *A* and bears on the scale as shown. The heat generated by the friction is carried off by water introduced inside the rim, and removed by the outlet pipe which is so shaped as to scoop the water from the rotating wheel. The downward pressure of the lever arm is balanced by the weights *W* on the platform scale.

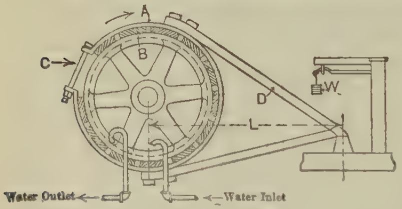


Fig. 1. Prony Brake

If $L$ = horizontal length of lever arm, ft., $N$ = rev. per min. of wheel, $W$ = weight required to balance the downward pressure of the lever arm, lb., $w$ = tare of brake, lb. = weight on scales with apparatus at rest, and Hp. = the horsepower of the shaft or mechanism to which the brake is attached, then

$$\text{Hp.} = 2\pi LN(W - w)/33{,}000 = (W - w)\,N/K, \quad . \quad . \quad . \quad . \quad [1]$$

where $K = 33{,}000/2\pi L$. If $L = 33/2\pi = 5\ {}^1/_4$ ft. (approx.), then Hp. = $1000WN$.

If the rubbing-surface be too small, the resulting friction will show great irregularity, probably because of insufficient lubrication, the blocks will seize the pulley and thus produce shocks and sudden vibrations of the lever-arm.

Soft woods, as bass, plane-tree, beech, poplar, or maple, are all to be preferred to the harder woods for brake-blocks. The blocks should be spaced at least $^1/_4$ in. apart. The rubbing surface should be well lubricated with a heavy grease.

Another form of Prony brake is a pair of blocks or jaws, gripping the shaft. The blocks should extend around the greater part of the circumference. A pair of bolts through the blocks on either side of the shaft serves to tighten the blocks on the shaft. A lever-arm of any desired length is attached to the upper block and bears on a platform scale or is supported by a spring balance. The horsepower is computed by formula [1].

**Rope Brake.**—For small amounts of power, the Prony brake may consist simply of a rope or strap wrapped around the brake wheel, one end being fastened to the floor or some other immovable part of the structure, through a spring balance and the other end attached to a second spring balance or weighted. See Fig. 2. If $W$ and $w$ are, respectively the net applied weight and the net spring balance reading, and $R$ is the radius of the brake wheel in feet, then

$$\text{Hp.} = 2\pi R N(w-W)/33{,}000 \quad . \quad . \quad . \quad . \quad [2]$$

Prony brakes are suitable only for the measurement of relatively small powers, say up to about 100 Hp. and for speeds of not over 1000 r.p.m. For greater horsepower and higher speeds a water brake or electric dynamometer should be used.

**CAPACITY OF FRICTION-BRAKES.**—W. W. Beaumont (*Proc. Inst. C. E.*, 1889) has deduced a formula by means of which the relative capacity of brakes can be compared, based on the amount of horsepower ascertained by their use.

Let $W$ = width of rubbing-surface on brake-wheel, in.; $V$ = velocity of point on circumference of wheel, ft. per min.; $K$ = coefficient; then

$$K = WV \div \text{Hp.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

Prof. Flather obtained the values of $K$ given in Table 1. Calculations for eleven brakes gave values of $K$ varying from 84.7 to 1385 for actual horsepowers determined, the average being $K$ = 655. Instead of assuming an average coefficient, Prof. Flather recommends the following:

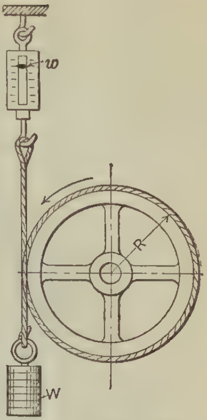


FIG. 2. Rope Brake

Water-cooled brake, non-compensating, $K$ = 400; $W$ = 400 Hp. ÷ $V$.
Water-cooled brake, compensating, $K$ = 750; $W$ = 750 Hp. ÷ $V$.
Non-cooling brake, with or without compensating device, $K$ = 900; $W$ = 900 Hp. ÷ $V$.

F. A. Halsey (*Am. Mach.*, April 4, 1912), advises a minimum brake-drum surface for water-cooled Prony brakes of 0.09 sq. ft. per brake Hp. to avoid the danger of the blocks taking fire.

A brake described in *Am. Mach.*, July 27, 1905, had an iron water-cooled drum, 30 in. diam., 20 in. face, with brake blocks of maple attached to an iron strap nearly surrounding the drum. At 250 r.p.m., or a circumferential speed of 1963 ft. per min., the limit of its capacity was about 140 Hp.; above that power the blocks took fire. At 140 Hp. the total surface passing under the brake blocks per minute was 3272 sq. ft., or 23.37 per Hp. This corresponds to a value of $K$ = 285.

**ABSORPTION DYNAMOMETERS.**—The two types of absorption dynamometers in general use are the water brake, exemplified by the Alden and Froude dynamometers, and the electric dynamometer.

**The Alden Absorption-Dynamometer** (G. I. Alden, *Trans.* A.S.M.E., xi, p. 958; also xii, p. 700 and xiii, p. 429.) is a friction-brake, capable in moderate sizes of absorbing large powers with unusual steadiness and complete regulation. A smooth cast-iron disc is keyed on the rotating shaft. This is enclosed in a cast-iron shell, formed of two discs and a ring at their circumference, which is free to revolve on the shaft. To the interior of each of the sides of the shell is fitted a copper plate, enclosing between itself and the side a water-tight space. Water under pressure is admitted into each of these spaces, forcing the copper plates against the central disc. The chamber enclosing the disc is filled with oil. To the outer shell is fixed a weighted arm, which resists the tendency of the shell to rotate with the shaft, caused by the friction of the plates against the central disc. See *Trans.* A.S.M.E., xi, p. 958; xii, p. 700; xiii, p. 429.

**Table 1.—Values of $K$ for Various Types of Friction Brakes.**

| Horse-power | R.p.m. Brake Pulley | Brake Pulley | | Length of Arm, in. | Design of Brake | Value of $K$ |
|---|---|---|---|---|---|---|
| | | Face, in. | Diam., ft. | | | |
| 21 | 150 | 7 | 5 | 33 | Royal Ag. Soc., compensating | 785 |
| 19 | 148.5 | 7 | 5 | 33.38 | McLaren, compensating | 858 |
| 20 | 146 | 7 | 5 | 32.19 | McLaren, water-cooled and compensating | 802 |
| 40 | 180 | 10.5 | 5 | 32 | Garrett, water-cooled and compensating | 741 |
| 33 | 150 | 10.5 | 5 | 32 | Garrett, water-cooled and compensating | 749 |
| 150 | 150 | 10 | 9 | ........ | Schoenheyder, water-cooled | 282 |
| 24 | 142 | 12 | 6 | 38.31 | Balk | 385 |
| 180 | 100 | 24 | 5 | 126.1 | Gately & Kletsch, water-cooled | 209 |
| 475 | 76.2 | 24 | 7 | 191 | Webber, water-cooled | 847 |
| 125<br>250 | 290<br>250 | 24 | 4 | 63 | Westinghouse, water-cooled | 465 |
| 40<br>125 | 322<br>290 | 13 | 4 | 27 3/4 | Westinghouse, water-cooled | 847 |

Four brakes of this type, 56 in. diam., were used in testing the experimental locomotive at Purdue University. Each was designed for a maximum moment of 10,500 ft.-lb. with a water-pressure of 40 lb. per sq. in. The area in effective contact with the copper plates on either side is represented by an annular surface with an outer radius of 28 in. and an inner radius equal to 10 in. The apparent coefficient of friction between the plates and the disc was 3 $^1/_2$%.

**The Froude Dynamometer,** made by C. H. Wheeler Mfg. Co., Philadelphia, is a water brake, consisting of a rotor keyed to the shaft and rotating in a casing. Both rotor and casing have pockets in their faces, separated by vanes. The friction of the water tends to cause the casing to rotate simultaneously with the impeller, which tendency is resisted by a lever arm fixed to the casing and bearing on a weight-measuring apparatus. The horsepower is calculated by formula [1]. These dynamometers are regularly built to measure up to 5000 Hp. at 250 r.p.m., and 300 Hp. at 4000 r.p.m.

In these dynamometers, the water should be supplied under a constant pressure of from 10 to 30 lb. per sq in., gage, preferably from an overhead tank in which a float valve maintains a constant water level. The quantity of water required depends on the temperature of the water. With water at 70° F. a minimum supply of about 3 gal. per brake Hp.-hour should be maintained, and about 4 gal. per brake Hp.-hour with water at 100° F. These figures are based on the water leaving the dynamometer at 180° F.

**The Electric Dynamometer** made by the Sprague Works of the General Electric Co. is a device for measuring dynamic torque, which may be used either as an absorption element or as a prime mover. In both cases the power shaft of the apparatus under test is coupled to the shaft of the internally revolving rotor of the dynamometer which transmits the full torque exerted by or on that shaft to the external floating frame by electro-magnetic interaction. The floating field is balanced in anti-friction bearings and is restrained from revolving by scales on which the torque is measured directly. The full torque developed by the dynamometer (as a motor) or delivered to it by a prime mover (as a brake) is recorded on the scales. These dynamometers are made in capacities of from 4 to 1000 Hp., with maximum speeds up to 5000 r.p.m.

The horsepower formula is, as before, Hp. = $2\pi NWL/33{,}000 = WN/K$. For a 15 $^3/_4$ in. arm $K = 4000$; for a 21 in. arm $K = 3000$. The adjustment of the torque or speed is accomplished by a dual control of current strength in the rotor and stator windings. This dual control permits the speed to be adjusted through successive increments of the closest reading speed indicator from zero to maximum speed and the torque to any intermediate value between maximum and a minimum of nearly zero.

**TRANSMISSION DYNAMOMETERS** are of various forms, as the Batchelder dynamometer, in which the power is transmitted through a "trained-arm" of bevel gearing, with its modifications, as the one described by William Kent in *Trans.* A.I.M.E., viii, p. 177, and the one described by Samuel Webber in *Trans.* A.S.M.E., x, p. 514; belt dynamometers, as the Tatham; the Van Winkle dynamometer, in which the power is transmitted from a revolving shaft to another in line with it, the two almost touching, through the medium of coiled springs fastened to arms or discs keyed to the shafts; the Brackett and the Webb cradle dynamometers, used for measuring the power required to run dynamo-electric machines. Descriptions of the four last named are given in Flather on Dynamometers.

The Kenerson transmission dynamometer is described in *Trans.* A.S.M.E. 1909. It has the form of a shaft coupling, one part of which contains a cavity filled with oil and covered by a flexible copper diaphragm. The other part, by means of bent levers and a thrust ball-bearing brings an axial pressure on the diaphragm and on the oil, and the pressure of the oil is measured by a gage.

Much information on various forms of dynamometers will be found in *Trans.* A.S.M.E., vols. vii to xv inclusive, indexed under Dynamometers.

## Section 25

# ELECTRIC MOTORS AND THEIR CONTROL

By M. N. Halberg

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## ELECTRIC MOTOR CONTROL

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# ELECTRIC MOTORS AND THEIR CONTROL

By M. N. Halberg

## DEFINITIONS

**ALTERNATING CURRENT.**—An electric current, the direction of which reverses at regularly recurring intervals. Unless distinctly otherwise specified, *alternating current* refers to a periodically varying current with successive half waves of the same shape and area. Abbreviation, A.C.

**APPARENT EFFICIENCY** of an alternating-current motor is the product of its efficiency and its power factor.

**APPARENT POWER OR VOLT-AMPERES** in an alternating-current circuit is the product (r.m.s. value * of voltage across the circuit) × (r.m.s. value * of current in the circuit). Apparent power also is expressed in kilo-volt-amperes (1000 volt-amperes). Abbreviation, kva.

**ARMATURE.**—The part of a dynamo-electric machine which includes the main current-carrying winding. In direct-current machines and in alternating-current commutator machines, the armature winding is connected to the commutator, and the armature is the rotating member. In alternating-current machines without commutators the armature may be either the rotating member or the stationary member.

**ASYNCHRONOUS MACHINE.**—A dynamo-electric machine in which the speed of operation is not exactly proportional to the frequency of the system to which it is connected.

**AUTOMATIC TRIP.**—The opening (other than manually) of an oil or air circuit breaker as a result of changes in current or voltage, or of other electrical conditions.

**AUTO-TRANSFORMER.**—A form of transformer that has one continuous winding, part of which is common to both primary and secondary circuits.

**BRANCH CIRCUIT.**—That portion of a wiring system extending beyond the final automatic overload protective device of the circuit.

**CAPACITOR (Static Condenser).**—A device whose primary purpose is to store electric charges. It commonly is used in A.C. circuits to change the relative time phase between voltage applied and current flowing in the circuit. The usual form consists of two conducting plates, usually of aluminum foil, separated by a thin layer of insulation (dielectric), the combination being hermetically sealed in a steel container.

**CIRCUIT BREAKER.**—A device designed to open, under abnormal conditions, a current-carrying circuit without injury to itself. The term as used in the National Electrical Code applies only to the automatic type designed to trip on a predetermined overload of current.

**CIRCULAR MIL.**—The area of a circle of 1 mil diameter, or $(\pi/4) \times (0.001)^2$ sq. in. $= 0.000\,000\,7854$ sq. in.

**CLASS A INSULATION** consists of cotton, silk, paper and similar organic materials when impregnated or immersed in oil; also enamel as applied to conductors.

**CLASS B INSULATION** consists of inorganic materials, such as mica and asbestos in built-up form, combined with binding substances. If Class A material is used in small quantities in conjunction with Class B, for structural purposes only, the combined material may be considered as Class B, provided the electrical and mechanical properties of the insulated winding are not impaired by the application of the temperature permitted for Class B material. (The word "impair" is here used in the sense of causing any change which could disqualify the insulating material for continuous service.)

**CLASS C INSULATION** consists of inorganic materials such as pure mica, porcelain, quartz, etc.

**CLASS O INSULATION** consists of cotton, silk, paper and similar organic materials when neither impregnated nor oil immersed.

**COMMUTATING POLE (Interpole).**—An auxiliary pole placed between the main poles of a commutating dynamo-electric machine. Its exciting winding carries a current proportional to the load current and produces a flux in such a direction and phase as to assist the reversal of current in the short-circuited coil.

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* R.m.s. value (root-mean-square) of a current or voltage is the value read by an ordinary ammeter or voltmeter. It numerically is equal to $\sqrt{\text{mean of (instantaneous values)}^2}$.

**COMMUTATOR.**—An annular ring or disc assembly of conducting members, individually insulated in a supporting structure, with an exposed surface for contact with current collecting brushes, and ready for mounting on an armature spider, shaft, or quill.

**CONDUCTOR.**—A metallic structure used to carry electric current. It usually is a wire or a combination of wires not insulated from one another, but bars or tubes also are used as conductors.

**CONTROLLER.**—A device, or group of devices, which governs, in some predetermined manner, the electric power delivered to the apparatus to which it is connected.

**CONVENTIONAL EFFICIENCY.**—The efficiency as computed by means of the component losses, most of which are accurately determinable, and the remainder of which are assigned conventional values; or all of the losses may be determined by conventional methods of test. The efficiency so determined is the ratio $O/(O + L)$ or the ratio $(I - L)/I$, where $O$ = output; $I$ = input; $L$ = losses.

**COUNTER ELECTROMOTIVE FORCE.**—The effective electromotive force, within a system, which opposes the passage of current in a specified direction.

**DIRECT CURRENT.**—A uni-directional current. As ordinarily used, the term designates a practically non-pulsating current. Abbreviation, D.C.

**EFFICIENCY** of an apparatus (machine or motor) is the ratio of the useful power output to the total power input.

**ELECTRICAL DEGREE.**—The 360th part of the angle subtended, at the axis of the machine, by two consecutive field poles of like polarity. One *mechanical degree* is thus equal to as many electrical degrees as there are pairs of poles in the machine.

**ELECTROMOTIVE FORCE.**—The property of a physical device which tends to produce an electric current in a circuit. Abbreviation, e.m.f.

**EXCITER.**—An auxiliary generator which supplies electric current to excite the field of another electrical machine.

**FREQUENCY, of an Alternating Current.**—The number of cycles through which the current passes per second; frequency is the reciprocal of the *period*.

**GENERATOR, Double Current.**—A machine which supplies both direct and alternating currents from the same armature.

**GENERATOR, Electric.**—A machine which transforms mechanical energy to electrical energy.

**GENERATOR, Induction.**—An induction machine (see below) driven above synchronous speed by an external source of mechanical power, and returning energy to the circuit.

**GENERATOR, Magneto-electric.**—An electric generator, the field poles of which are permanent magnets.

**GROUND.**—A conducting connection, whether intentional or accidental, between an electrical circuit and earth or any other conducting or partly conducting structure or material.

**INDUCTION MACHINE.**—An asynchronous alternating-current machine which comprises a magnetic circuit interlinked with two electric circuits, rotating with respect to each other and in which power is transferred from one circuit to another by electromagnetic induction. Examples of induction machines are induction motors, induction generators, and certain types of frequency converters and phase converters.

**KILOWATT.**—A unit of electrical power equal to 1000 watts or 1.34 horsepower.

**KILOWATT-HOUR.**—The work performed by one kilowatt acting for one hour.

**LOAD FACTOR.**—The ratio of the average load over a designated period to the peak load occurring in that period.

**MAGNETIC CONTROLLER.**—A controller all of whose basic functions are performed by electromagnets.

**MANUAL CONTROLLER.**—A controller all of whose basic functions are performed by hand.

**MOTOR, Adjustable Speed.**—See pp. 25–12, 25–32, 25–45.

**MOTOR, Capacitor.**—See p. 25–19.

**MOTOR, Capacitor-start.**—See p. 25–19.

**MOTOR, Compound-wound.**—A direct-current motor with two separate field windings connected, *a*, in parallel with the armature circuit, and, *b*, in series with the armature circuit. Winding *a* usually is the predominant field.

**MOTOR, Constant-speed.**—See p. 25–12.

**MOTOR, General Purpose.**—Any motor of 200 Hp. or less, and 450 r.p.m. or more having a continuous rating, and designed, listed or offered in standard ratings for use without restriction to a particular application.

**MOTOR, Induction.**—An induction machine that converts electrical energy delivered to the primary circuit into mechanical energy by the electromagnetic interaction between the primary and secondary circuits. The secondary circuit is short-circuited or closed through a suitable circuit.

**MOTOR, Large Power (Integral Horsepower).**—A motor built in a frame having a continuous rating of 1 Hp., open type, at 1700–1800 r.p.m., or in a larger frame.

**MOTOR, Non-excited Synchronous.**—A synchronous motor without a direct-current field winding.

**MOTOR, Reactor-start.**—A form of split-phase motor (see below) designed to start with a reactor in series with the main winding. The reactor is short-circuited, or otherwise made ineffective, and the auxiliary circuit is opened when the motor has attained a predetermined speed.

**MOTOR, Repulsion.**—A single-phase alternating-current motor with a stator winding arranged for connection to the source of supply, and a rotor winding connected to a commutator. Brushes on the commutator are short-circuited and so set that the line connecting the positive and negative brushes is at an angle to the neutral axis of the stator winding. This type of motor has a varying speed characteristic.

**MOTOR, Repulsion Induction.**—A form of repulsion motor which has a squirrel-cage winding in the rotor, in addition to the repulsion motor winding. This type of motor may have constant speed or varying speed characteristics. See p. 25–19.

**MOTOR, Repulsion-start Induction.**—A single-phase motor having the same windings as a repulsion motor, but in which, at a predetermined speed, the rotor winding is short-circuited or otherwise connected to give the equivalent of a squirrel-cage winding. This type of motor starts as a repulsion motor, but operates as an induction motor with constant speed characteristics. See p. 25–19.

**MOTOR, Resistance-start.**—A form of split-phase motor (see below) with a resistance connected in series with the auxiliary winding. The auxiliary circuit is opened when the motor has attained a predetermined speed.

**MOTOR, Series-wound.**—A commutator motor in which the field circuit and armature circuit are connected in series.

**MOTOR, Shaded Pole.**—See p. 25–19.

**MOTOR, Shunt-wound.**—A direct-current motor in which the field circuit and armature circuit are connected in parallel.

**MOTOR, Slip-Ring.**—See Motor, wound-rotor induction.

**MOTOR, Small Power (Fractional Horsepower).**—A motor built in a frame smaller than that having a continuous rating of 1 Hp., open type, at 1700–1800 r.p.m.

**MOTOR, Special Purpose.**—An industrial power motor specifically designated and listed for a particular power application, where load requirements and duty cycle are definitely known.

**MOTOR, Split-phase.**—A single-phase induction motor equipped with an auxiliary winding, which is displaced in magnetic position from, and connected in parallel with, the main winding. The auxiliary circuit is opened when the motor has attained a predetermined speed. See p. 25–19.

Note: The term split-phase motor, used without qualification describes a motor to be used without impedance, other than that offered by the motor windings themselves.

**MOTOR, Squirrel-cage Induction.**—An induction motor in which the secondary circuit consists of conductors suitably disposed in slots in the secondary core. See p. 25–28.

**MOTOR, Synchronous.**—A synchronous machine which transforms A.C. electrical power to mechanical power, and which has a speed strictly proportional to the frequency of the operating current. See p. 25–20.

**MOTOR, Synchronous Induction.**—A wound-rotor induction motor to which direct-current excitation is supplied when it approaches rated speed, enabling it to start as an induction motor and operate as a synchronous motor. See p. 25–40.

**MOTOR, Universal.**—A series-wound or compensated series-wound motor which may be operated either on direct current or single-phase alternating current (of approximately the same voltage and a frequency not greater than 60 cycles per second) at approximately the same speed and output. See p. 25–20.

**MOTOR, Wound-rotor Induction.**—An induction motor in which the secondary circuit consists of a polyphase winding, or coils whose terminals are short-circuited or closed through suitable circuits. (When provided with collector or slip rings a wound-rotor induction motor also is known as a slip-ring induction motor). See p. 25–30.

**OPEN MACHINE (or Motor).**—A self-ventilated machine having no restriction to ventilation, other than that necessitated by mechanical construction.

**POWER.**—The rate of transfer of energy. In the case of an alternating-current circuit the word power is generally used to denote the average value of the power over a cycle. The power in an electric circuit at any instant is equal to the product of the values of the current and voltage at that instant, and generally is called instantaneous power.

**POWER FACTOR.**—The ratio of the power to the apparent power, that is, it equals kw./kva. The maximum power factor is 1 or unity. It may be lagging or leading, depending upon whether the current wave lags or leads the voltage wave. See Kent's Mechanical Engineers' Handbook—Power, pp. 15–53 to 15–60, forming Volume 2 of this series.

**PROOF (as a suffix).**—A machine (or apparatus) is designated as splash-proof, moisture-proof, explosion-proof, etc., when so constructed, protected or treated that subjection to the specified material or condition will not interfere with its successful operation.

**RATING of a machine, apparatus or device.**—A designated limit of operating characteristics based on definite conditions. Such operating characteristics as load, voltage, frequency, etc., may be given in the rating.

**RATING, Continuous.**—The rating that defines the load which can be carried for an indefinitely long time.

**RATING, Short-time.**—The rating that defines the load which can be carried for a short and definitely specified time, the machine, apparatus or device being at approximately room temperature at the time load is applied.

**RESISTING (as a suffix).**—A machine (or apparatus) is designated as moisture-resisting, fume-resisting, etc., when so constructed, protected or treated that it will not be readily injured when subjected to the specified material.

**SEMI-MAGNETIC CONTROLLER.**—A controller having part of its basic functions performed by electromagnets and part by other means.

**SINGLE-PHASE CIRCUIT.**—A circuit energized by a single alternating electromotive force. (A single-phase circuit usually comprises two wires. The currents in these two wires, counted outwards from the source, differ in phase by 180 degrees or a half cycle.)

**SLIP (of an induction machine).**—The ratio, expressed as a percentage, of the difference between operating speed and synchronous speed to the synchronous speed.

**SWITCH, General Use.**—A switch intended for use as a switch in general distribution and branch circuits. It is rated in amperes and is capable of interrupting its rated current at its rated voltage.

**SWITCH, Isolating.**—A switch intended for isolating a circuit from its source of power. It is to be operated only when the circuit has been opened by some other means.

**SWITCH, Motor-circuit.**—A switch intended for use in a motor-branch-circuit. It is rated in horsepower and is capable of interrupting the maximum operating overload current of a motor of the same horsepower at the rated voltage. The maximum operating overload of A.C. motors is six times the rated current, and of D.C. motors is four times the rated current.

**SYNCHRONOUS MACHINE.**—A dynamo-electric machine in which the speed of normal operation is exactly proportional to the frequency of the current.

**THREE-PHASE CIRCUIT.**—A combination of circuits energized by alternating electromotive forces which differ in phase by one-third of a cycle, *i.e.*, 120 degrees.

**TIGHT (as a suffix).**—A machine is designated as water-tight, dust-tight, etc., when so constructed that the enclosing case will exclude the specified material.

**TORQUE.**—See p. 25–14.

**TWO-PHASE CIRCUIT.**—A combination of circuits energized by alternating electromotive forces which differ in phase by a quarter of a cycle; *i.e.*, 90 degrees.

**UNDER-VOLTAGE PROTECTION.**—The effect of a device, operative on the reduction or failure of voltage, to cause and maintain the interruption of power to the main circuit.

**UNDER-VOLTAGE PROTECTION, Time Delay.**—The effect of a device, operative on the reduction or failure of voltage for a definite period of time, to cause and maintain the interruption of power to the main circuit.

**UNDER-VOLTAGE RELEASE.**—The effect of a device, operative on the reduction or failure of voltage, to cause the interruption of power to the main circuit, but not to prevent the re-establishment of the main circuit on return of voltage.

## APPLICATION OF MOTORS AND CONTROL

**FACTORS AFFECTING MOTOR SELECTION.**—The various factors which may require consideration in selecting a motor and controller for a given application are summarized and discussed on pp. 25–07 to 25–10. These are broadly classified in accordance with their relation to: *a*. The power supply; *b*. The driven machine; *c*. The location.

**Power Supply**

- Characteristics
  - D.C.—voltage
  - A.C.—voltage, phases and frequency
- Power rates
  - Energy charge
  - Power-factor clause
  - Demand charge
- Voltage regulation
  - Starting current limitations
  - Allowable current pulsations
- Continuity of power
- Short-circuit current
- Voltage surges

**Driven Machine**

- Mechanical arrangement
  - Horizontal or vertical shaft
  - Tilting
  - Stationary or portable
  - Method of connecting motor — Direct-connected, built-in, belt, chain, or gear drive
  - Space available
- Speed, speed range, regulation
- Direction of rotation, reversing
- Horsepower, r.m.s., peak
- Torques
  - Starting, breakaway, pull-in
  - Running-torque pulsations
- $WR^2$ of rotating parts
- Thrust
- Frequency of starting
- Deceleration time
- Operating schedule
- Initiation of operating sequence

**Location**

- Ambient temperatures
- Altitude
- Moisture
- Dirt
- Corrosive atmosphere
- Explosive dust or gas
- Effect of noise
- Codes and ordinances

## Power Supply

**VOLTAGE** of a D.C. power supply, and the voltage, number of phases, and frequency of an A.C. power supply must be known in order to select a correspondingly rated motor and controller. In most cases these conditions are fixed and readily determined. However, in some cases a choice is available or a conversion from one set of conditions to another is desirable. For example, it often is necessary to convert a high A.C. voltage to a lower value by means of transformers for economical use of small motors. Again, in some applications, as those involving operation over a wide speed range or severe reversing duty, it is desirable to convert A.C. to D.C., by means of a motor-generator set or a rectifier, to allow the use of D.C. motors.

**POWER RATES** may affect the choice of motor and controller, since the efficiencies of various types of motor differ appreciably, which means a corresponding difference in the power used. This is particularly true, in the case of adjustable speed motors where wide variations in efficiency occur, which must be balanced against wide differences in cost. Many A.C. power-rate schedules incorporate a *power-factor clause*, by which the billing is adjusted in accordance with the power-factor of the current used. This may affect the choice of motor. For example, it may justify the use of a synchronous motor where an induction motor otherwise would be selected. Finally, most rate schedules embody a *demand charge*, that is, the maximum rate at which energy is used for a given period of time, as well as the total amount consumed, determines the power bill. This makes it desirable to maintain as uniform a load as possible at all times, which in some applications will justify the use of an adjustable speed or multi-speed motor instead of a constant speed machine. See Purchased Power, pp. 15–03 to 15–20, and Power Factor, pp. 15–53 to 15–60, Kent's Mechanical Engineers' Handbook—Power, forming Vol. 2 of this series.

**VOLTAGE REGULATION** of the power supply represents the drop in voltage which occurs with a certain increase in the current drawn. This must be considered, first, because the torques developed by motors vary with the impressed voltage, so that the actual voltage at the motor terminals under the various operating conditions must be known. In the case of A.C. motors, the starting torques vary as the square of the voltage. Since these motors draw a relatively high starting current, the effect of the voltage regulation of the power supply is particularly important. Secondly, if lights are supplied from the same power circuit, a drop in voltage of about 2% causes a noticeable flicker of these lights. Finally, a drop in voltage of 30% or more may cause other motors on the system to stall or be disconnected from the line by their under-voltage protective device.

Power companies usually establish limits to the starting current which can be drawn from the line by motors, in order to prevent excessive drop in line voltage. These factors often necessitate the use of reduced-voltage controllers, where full-voltage starting otherwise would be acceptable. In some cases, a different type of motor must be used, for example, a wound-rotor induction motor instead of a squirrel-cage induction motor, in order to obtain sufficient starting torque while keeping within the allowable starting current.

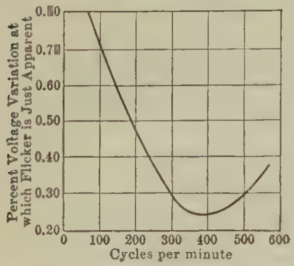

Fig. 1. Voltage Variation Causing Light Flicker at Various Temperatures

**Pulsating Current.**—Where the load is such that the motor draws a pulsating current from the line, corresponding variations in line voltage occur. If these have a frequency lying in a certain range (see Fig. 1), they can be of very small magnitude and yet produce a noticeable flickering of lights supplied from the circuit. The usual limit established for current pulsation is 66% of motor full-load current. In most cases this does not cause light flicker, but in particular installations the regulation of the system may be such as to require a much lower limit of pulsation such as 30 or 40%.

**CONTINUITY OF POWER SUPPLY** may affect the choice of motors and control. Thus, if momentary interruptions or *dips* in line voltage are apt to occur, the use of time-delay rather than instantaneous under-voltage protection may be justified. See p. 25–52. In special cases, control equipment may be required which automatically will transfer the motor from one source of power to another, in the event of failure of the first.

**SHORT-CIRCUIT CURRENT** of the power system may affect the choice of control equipment. If the line switch of the controller is to be called upon to interrupt short-circuit current, it may be necessary to use a circuit-breaker of suitable rating, or a high-interrupting-capacity contactor, to meet the requirements.

**VOLTAGE SURGES.**—When a motor is supplied from high-voltage overhead power lines, lightning disturbances in the surrounding area may produce dangerous voltage surges at the motor terminals. Special winding insulation and in some cases external protective equipment may be justified.

## Driven Machine

**MECHANICAL ARRANGEMENT** of the driven machine is the first consideration. Whether it is driven through a horizontal shaft or a vertical shaft determines which of these two types of motor should be selected. In some cases the motor normally operates in one position, but may, momentarily or for extended periods, be tilted in one or more directions. This condition may require special bearing construction to carry the thrust and prevent leakage of lubricant. Again, the motor may be required to operate continuously in any position. This usually means the use of grease-lubricated ball bearings.

Whether the driven machine is *stationary* or *portable* may affect the choice of motor. In the latter case, compactness and light weight are often essential. Operation in more than one position usually is involved.

**Method of Connecting Motor** to the driven machine must be considered. If the speed corresponds to that of an economical motor speed, a direct-connected motor normally would be used. Otherwise, speed transformation by belt, chain or gears must be employed. Maintenance, efficiency, and space requirements as well as initial cost enter into this choice. In some cases a direct-connected motor economically may be *built-in*. This usually involves the use of a special motor, supplied without shaft or bearings, and arranged to be mounted on the shaft of the driven machine. In some cases, the motor stator also is arranged to be supported from the driven machine. Compactness is one of the principal advantages of this construction. Where the motor drives through a flat belt, multiple-V belt, or chain, the dimensions of pulleys or sprockets and the center distance must be known, in order to determine the side pull on the motor shaft. In some cases, the motor must have an extended shaft and outboard bearing to limit bearing pressure and shaft deflection. A sliding base, or other means for adjusting the position of the motor, to maintain proper tension in the driving member is necessary with belt or chain drive.

**Space Available** for a motor may affect its choice. If space is limited, a direct-connected or built-in motor usually is most suitable. Where the speed of the driven machine prohibits this choice, a gear-motor often is the best solution. In some cases forced ventilation of a motor by an external blower may be used effectively to reduce the motor size. Similar results may be obtained by using a motor with Class B insulation, operating at a correspondingly higher temperature rise.

**SPEEDS.**—The operating speed, speeds, or speed range of the driven machine must be known in order to select a corresponding single-speed, multi-speed, or adjustable speed motor. If the machine operates at a single speed, any type of motor may be used, the choice being made on other factors. Where two, three or four fixed speeds, bearing the proper ratio to each other, are acceptable, a multi-speed squirrel-cage induction motor may meet the requirements. Where the speed range does not exceed 2:1, and a reasonable number of speed points is satisfactory, and where considerable variation in speed with load is not detrimental, the wound-rotor induction motor should be considered. For more severe requirements a D.C. motor with field or variable-voltage control, or an A.C. adjustable-speed, brush-shifting motor may be necessary.

**Speed Regulation.**—The allowable speed regulation (variation in speed with load) may determine the type of motor, particularly where adjustable speed operation is involved. For most single-speed applications, the squirrel-cage induction motor or shunt-wound D.C. motor is satisfactory, but in a few applications zero speed regulation is required. This necessitates the use of a synchronous motor. For adjustable speed service, the wound-rotor induction motor, and the D.C. motor with armature resistance control, often are eliminated because of their high speed regulation.

**DIRECTION OF ROTATION.**—Certain types of motors require adjustment or reconnection of windings to change the direction of rotation, which can be most conveniently done in the factory. Hence, the direction of rotation should be specified. Where the motor is to run in both directions, a motor suitable for reversing, and a reversing controller, must be provided.

**HORSEPOWER** required to drive the load determines the motor rating. Where the load is not constant but varies with time, the complete duty cycle (Hp. *vs.* time curve) should be obtained. From this curve, the peak horsepower, which determines the maximum torque of motor, and the root-mean-square (r.m.s.) horsepower can be calculated. The latter is the square root of the mean value (averaged against time) of the square of the horsepower needed. It is an approximate indication of the horsepower rating of the driving motor required from the standpoint of heating. However, where large variations in load occur, or where the motor is shut down, accelerating or decelerating an appreciable portion of the time, the r.m.s. horsepower does not necessarily give a true indication of the equivalent continuous load, and it is advisable to consult the motor manufacturer. If the load continues at a certain value for an extended period of time (15 minutes to 2 hours, depending on the size of the motor), the horsepower rating of the motor should be not less than this value, regardless of any subsequent reduction in load. Where the motor operates at more than one speed, the horsepower required at each speed must, of course, be known.

**TORQUES** necessary to start and drive a load deserve careful consideration. Of particular importance is the breakaway torque requirement, since this often is considerably higher than the running torque. Furthermore, reduced-voltage starting may be necessary to meet starting current limitations, which reduces the torque delivered by the motor. Where a synchronous motor is being considered, the torque required at full speed under starting conditions is equally important, since it determines the pull-in torque. Pull-in torque is not necessarily equal to full-load torque, since the machine may be unloaded for starting, or may be required to start when the load is temporarily higher than normal. It is often desirable or essential to know the load torque at all speeds from standstill to full speed (speed-torque curve). This affects the accelerating time and hence the heating of the motor and control equipment. In particular, squirrel-cage induction motors, synchronous motors, and starting resistors, reactors, and auto-transformers may have to be of special design if the accelerating time is unusually long. Where reduced-voltage starting is employed, the speed-torque curve of the load determines the speed to which the motor will accelerate on reduced voltage, and hence determines the current inrush when the motor is transferred to full voltage.

**Running Torque.**—The horsepower required and the speed of the driven machine determine the running torque. Certain driven machines, of which reciprocating compressors are typical, have a load torque which pulsates periodically. To prevent excessive pulsation in the current drawn by the motor proper, fly-wheel effect, $WR^2$, must be provided in the motor or driven machine. In the case of synchronous motor drives, the fly-wheel effect must be such that the natural frequency of the motor is not close to the frequency of any impulses in the load torque. The former can be calculated as follows: Let $F$ = natural frequency of motor, cycles per min.; $WR^2$ = (weight, lb.) × (radius of gyration, ft.)$^2$ of the rotating parts of motor and load; $N.$ = rev. per min. of motor;

$f$ = line frequency; $P_r$ = synchronizing power of motor, kilowatts per electrical radian. Then $F = (35{,}200/N)\sqrt{(P_r \times f)/WR^2}$

**$WR^2$ OF THE ROTATING PARTS** of the driven machine affects the accelerating time, and hence affects the heating during the starting period of motors and control equipment. Its value is particularly important where the motor is started frequently, or where reversing duty is involved. The $WR^2$ always should be known if a synchronous motor is to be employed, since the pull-in torque of the motor varies approximately as the square root of the total $WR^2$ of motor and load. If a rotating member of the driven machine operates at a different speed from that of the motor, its $WR^2$ can be converted to an equivalent value at the motor shaft by multiplying by

$$\{(\text{r.p.m. of rotating member})/(\text{r.p.m. of motor})\}^2$$

**THRUST.**—Some driven machines (certain types of pumps and fans are typical) impose a thrust on the motor in a direction parallel to the shaft. Horizontal motors with thrust bearings may be required. Vertical motors may need an over-size thrust bearing if considerable external thrust is imposed.

**FREQUENCY OF STARTING** may affect the choice of motor and controller, since each start results in additional heating of the motor windings, starting resistors, etc. This must be considered in connection with the load torques and load $WR^2$, which determine the accelerating time and hence the heating during each start.

In some applications, **rapid deceleration** of the motor and load is essential. Where induction motors are used, it may be necessary to provide *plugging control*, and with synchronous or D.C. motors, dynamic braking control (see p. 25–58.)

**Plugging** consists of reversing two leads of a motor while the motor is running, so as to reverse the phase rotation. The control thus is similar to that provided for reversing service. A relay, mounted on the motor shaft, usually is included to disconnect the motor from the line when it stops, and prevent it from starting in the reverse direction.

**THE OPERATING SCHEDULE,** *i.e.*, the total number of hours of operation per year, of a machine should be known. Where this figure is high, motors having maximum efficiency are more readily justified. Where the load or speed varies, the operating time under each condition should be considered to permit proper evaluation of the motor characteristics to be made.

**Initiating Operating Sequence.**—If it is acceptable for the operator to start and stop the motor at the controller, the latter may be of the manual type. Where remote control or automatic operation in response to temperature, pressure, liquid level, etc., is required, a magnetic controller must be used.

### Location

**THE AMBIENT TEMPERATURE,** if above 40° C., and the **altitude,** if greater than 3300 ft. above sea level may affect the choice of motor. See p. 25–11. If a motor is to operate in a very moist atmosphere special insulation or, under severe conditions, totally-enclosed or enclosed-separately-ventilated construction may be required. Extreme conditions of dirt or corrosive atmospheres also require the use of totally-enclosed or enclosed-separately-ventilated motors. If located where explosive dust (Class II Group G), or gases (Class I, Group D), may be present, a corresponding type of explosion-proof motor should be selected.

**Noise.**—Installations in public buildings, as schools, theatres, hospitals, stores, etc., and residences, unless isolated from occupied sections, may require quiet operating motors to prevent objectionable noise. Suitable resilient mountings to prevent transmission of vibration through the building structure are often necessary. Finally, local regulations such as codes and ordinances must be considered. Some require special enclosures on motors and control equipment to reduce the hazard to operators.

# ELECTRIC MOTORS

## 1. GENERAL

**POWER SUPPLY.**—Practically all public utility power systems distribute alternating-current (A.C.) power, although direct-current (D.C.) power is provided in the downtown areas of certain large cities. A.C. motors are, in general, less costly, require less maintenance than D.C. machines, and meet the requirements of the large majority of applications. However, for certain loads D.C. motors have more suitable characteristics, and are used, even though A.C. to D.C. conversion equipment is required. D.C. motors must be designed for the proper line voltage, and A.C. motors for the proper voltage, phase, and frequency.

**Standard Motor Voltages** are as follows: Direct-current, 32, 115, 230, 550 volts. Alternating-current, 110, 220, 440, 550, 2200, 4000, 6600, 11,000, 13,200 volts. Since alternating current can be transformed readily from one voltage to another, a choice of motor voltage often is available. In general, the lowest cost and best operating characteristics are obtained with a voltage increasing with the horsepower rating as indicated in Table 1. Certain types of motors are available with a *dual voltage* feature, *i.e.*, additional terminals are provided so the motors can be connected for either one of two voltages, as 110 and 220 volts, or 220 and 440 volts.

**Frequency.**—Alternating-current frequencies of 60 and 25 cycles are standard in the U. S. The latter, primarily because of the limitations it imposes on the selection of motor speeds and its inferiority for lighting, is rapidly (1937) being superseded by the former. Frequencies of 50, 40 and 30 cycles also are in use in certain sections of the U. S., but are being superseded by 60 cycles. In Europe and European dependencies, 50 cycles is the general standard. In Canada, except for parts of Ontario, 60 cycles is standard.

**Phase.**—Single-phase, three-phase, and, to a limited extent, two-phase power is distributed and used commercially in the U. S. Three-phase energy is most satisfactory for power purposes, and is the accepted standard. In general, single-phase motors cost more and have less satisfactory operating characteristics than three-phase machines. However, single-phase power is used for lighting, and in residential and rural districts often is the only power available. Hence, single-phase motors commonly are used on household appliances and other small equipment to be installed in such districts. Power companies place a limit on the size of motor which may be connected to single-phase lines. This limit usually falls between $^{1}/_{4}$ and $^{3}/_{4}$ Hp. for 110-volt circuits and between 3 and 10 Hp. for 220-volt circuits.

**HORSEPOWER RATING.**—Motors usually are rated in horsepower, the rating representing their power output. Very small motors ($^{1}/_{60}$ Hp. and below) usually are rated in watts output (746 watts = 1 Hp.). Standard ratings have been established for the various types of motors.

Most motors are rated on a continuous basis, that is, they will carry rated load continuously. However, many types, such as crane and hoist motors, are given short time ratings such as 15-min., $^{1}/_{2}$-hr., or 1-hr. ratings, indicating that they can carry their rated load for the designated period of time, after which they must be allowed to cool to room temperature. In some cases a motor is given several different ratings, each for a different period of time. Short-time-rated motors are used on intermittent loads, the time rating being chosen to give the motor heating equivalent to that produced by the actual duty cycle of the load.

**TEMPERATURE RISE.**—Motors are designated 40-deg., 50-deg., etc., indicating the temperature rise in deg. C. that will not be exceeded when the motor carries rated load continuously (or for a specified time in the case of short-time-rated motors). The temperature rise refers to the windings, core and other parts adjacent to the insulation. Some motors are given different temperature guarantees for different parts as armature, field, etc. Temperatures usually are determined by thermometer, although temperature detectors embedded in the winding are used on large motors. In some cases, the temperature rise of a winding is determined by measuring the change in its resistance.

Temperature rise guarantees usually are on the basis of normal conditions of ventilation, an altitude of not more than 1000 meters (3300 ft.), and an ambient temperature of 40 deg. C. Less effective dissipation of heat at higher altitudes (a result of reduced density of air) causes a higher temperature rise. It is assumed that the temperature rise as measured at altitudes below 3300 ft. is 1% less than the temperature rise at a higher altitude for each 330 ft. by which this altitude exceeds 3300 ft. Expressed as an equation this becomes,

$$t_h = t_s/\{1.1 - (h/33{,}000)\} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

where $t_h$ = temperature rise at altitude $h$, $t_s$ = temperature rise at sea level, $h$ = altitude, ft. Thus a motor having a temperature rise of 40° C. at sea level, will have at an altitude of 9900 ft. a temperature rise of

$$t_h = 40/\{1.1 - (9900/33{,}000)\} = 50^\circ \text{ C.}$$

The type of insulation used has an important bearing on the allowable temperature

**Table 1.—Recommended Voltage Ratings of A.C. Motors**

| Motor Horsepower | Recommended Voltage | Motor Horsepower | Recommended Voltage |
|---|---|---|---|
| 3/4 and smaller | 110*, 220* | 50–200 | 220, 440*, 550, 2200* |
| 1–5 | 110, 220*, 440*, 550 | 200–500 | 440*, 550, 2200* |
| 5–50 | 220*, 440*, 550 | 500–1000 | 2200*, 4000, 6600 |

* Preferred voltage ratings.

rise. For Class A insulation (see p. 25–03) the upper temperature limit is placed at 105° C. If the insulation is allowed to reach a higher temperature unduly rapid deterioration is expected. For Class B insulation the upper temperature limit is placed at 125° C. Class A insulation is standard for most types of motors, although Class B insulation usually can be obtained at increased cost. Since it is impractical to measure the temperature of a motor at the *hottest spot*, standard practice limits the temperature as measured to a lower value. Where the temperature is measured by thermometer, it is assumed that the thermometer reading will be 15° C. below the hottest spot temperature. (For totally enclosed motors the differential is taken as 10° C.) Where temperature detectors are used the differential is taken as 5° C.; with the resistance method of temperature determination the differential usually is taken as 10° C. On this basis the temperature limits and temperature rises (based on 40° C. ambient) are as indicated in Table 2. Where a motor is to operate in an ambient temperature in excess of 40° C. a motor should be chosen with a temperature rise correspondingly lower.

Many types of motors are rated on the basis of a temperature rise less than allowable for the type of insulation employed. For example, 40° C. rise by thermometer is a common rating for motors with Class A insulation, whereas in a 40° C. ambient temperature, a rise of 50° C. is allowable. Such motors are frequently guaranteed to have a certain overload capacity, which is not available in motors rated at the maximum allowable temperature. Temperatures and temperature rises usually are given in degrees Centigrade. For conversion of Centigrade to Fahrenheit and Fahrenheit to Centigrade, see p. 28–50.

**SPEED CHARACTERISTICS.**—All motors may be classified, according to their speed characteristics, as follows: *a.* Constant speed. *b.* Varying speed. *c.* Adjustable speed. *d.* Adjustable varying speed. *e.* Multi-speed.

**A Constant Speed Motor** is one in which the speed is practically constant regardless of load. A change of a few percent in speed between no load and full load is characteristic, except with synchronous motors. Induction motors with small slip, and D.C. shunt motors are constant speed machines.

**A Varying Speed Motor** is one in which speed varies with load, ordinarily decreasing as load increases. The series D.C. motor and the high-slip, squirrel-cage induction motor are typical.

**An Adjustable Speed Motor** is one the speed of which can be varied gradually over a considerable range, but when once adjusted, remains practically constant being unaffected by load. The D.C. shunt-wound motor with field resistance control, designed for a considerable range of speed adjustment, is an example of this type of motor.

**An Adjustable Varying Speed Motor** is one in which the speed can be varied gradually over a considerable range, but in which the speed, when once adjusted to a given load, will vary in considerable degree with change in load; for example, a wound-rotor induction motor with secondary resistance control.

**A Multi-speed Motor** is one which can be operated at any one of several definite speeds, each being practically independent of the load; an induction motor with a primary winding capable of various pole groupings is an example.

**Operating Speeds.**—A characteristic of all constant-speed A.C. motors is that their operating speed is equal to or slightly less than one of the *synchronous speeds* corresponding to the frequency of the power supply (see Table 3). Multi-speed A.C. motors operate at two or more of these speeds. Synchronous motors, as their name implies, operate at exact synchronous speed. Induction motors operate at synchronous speed under no load, but under load they operate at a slightly lower speed. The difference between synchronous speed and full-load speed is known as full-load slip; it varies between 1 and 5% of synchronous speed for constant speed induction motors. D.C. motors can be designed

**Table 2.—General Temperature Limits for Motors**

| | Class A Insulation | Class B Insulation |
|---|---|---|
| **Measured by thermometer** | | |
| Temperature limit—open motors | 90° C. | 110° C. |
| " " —totally enclosed motors | 95° C. | 115° C. |
| Temperature rise (40° C. ambient)—open motors | 50° C. | 70° C. |
| " " ( " " )—totally enclosed motors | 55° C. | 75° C. |
| **Measured by temperature detectors** | | |
| Temperature limit | 100° C. | 120° C. |
| Temperature rise (40° C. ambient) | 60° C. | 80° C. |
| **Measured by resistance method** | | |
| Temperature limit | 95° C. | 115° C. |
| Temperature rise (40° C. ambient) | 55° C. | 75° C. |

for any required speed, subject to mechanical limitations. However, for the purpose of standardization, constant-speed D.C. motors usually are designed for full-load speeds approximately equal to the full-load speeds of 60-cycle induction motors.

In some cases, the speed classification of a motor depends entirely upon the control used with it. For example, a wound-rotor induction motor may be constant speed, varying speed or adjustable varying speed with no structural change in the motor whatsoever. A constant-speed D.C. motor may become, with no change in the motor itself, a wide-range adjustable-speed motor by the provision of means for varying the voltage impressed on its armature.

**DIRECTION OF ROTATION.**—Facing the end of the machine opposite the drive, the standard direction of rotation for all non-reversing motors, with the exception of polyphase induction motors, is counter clockwise. Unless otherwise specified standard motors are designed or connected for the standard direction of rotation. In the case of polyphase A.C. motors the terminals must be connected to the power lines in accordance with their phase rotation marking. This rule does not apply to polyphase induction motors whose terminals are not marked for any particular direction of rotation. They must be connected to the power lines and if the direction of rotation is wrong, two leads are reversed. Most motors can be reversed either by a change in connections or by mechanical adjustments.

**EFFICIENCY.**—The efficiency of a motor is defined as the ratio of useful power output to the total power input. Since power input is usually given in kilowatts,

$$\text{Efficiency} = (0.746 \times \text{Hp. output})/(\text{kilowatts input}),$$
$$\text{Kilowatts input} = (0.746 \times \text{Hp. output})/(\text{efficiency})$$

Motors of the same type and rating usually differ but little in efficiency. There may be considerable difference in the efficiency of motors of different types; this is particularly true of adjustable speed motors operating at reduced speeds. The efficiency of motors also varies considerably with the load and usually is highest at $^3/_4$ or full load. At light loads the efficiency is quite low.

**POWER-FACTOR.**—The power-factor of a motor is defined as the ratio of the kilowatts input to the kilovolt-amperes input where

$$\begin{aligned} \text{kva.} &= (\text{volts} \times \text{amperes})/1000, \text{ for single-phase motors,} \\ &= (1.73 \times \text{volts} \times \text{amperes})/1000, \text{ for 3-phase motors,} \\ &= (2 \times \text{volts} \times \text{amperes})/1000, \text{ for 2-phase motors.} \end{aligned}$$

Thus the kva. input to a motor equals (kw. input/power factor).

Table 3.—Synchronous Speeds Possible at Various Frequencies

| No. of Poles | 60 Cycles | 50 Cycles | 40 Cycles | 25 Cycles | No. of Poles | 60 Cycles | 50 Cycles | 40 Cycles |
|---|---|---|---|---|---|---|---|---|
| 2 | 3600 | 3000 | 2400 | 1500 | 52 | 138.5 | 115.4 | 92.3 |
| 4 | 1800 | 1500 | 1200 | 750 | 54 | 133.3 | 111.1 | 88.9 |
| 6 | 1200 | 1000 | 800 | 500 | 56 | 128.6 | 107.2 | 85.7 |
| 8 | 900 | 750 | 600 | 375 | 58 | 124.1 | 103.5 | 82.8 |
| 10 | 720 | 600 | 480 | 300 | 60 | 120 | 100 | 80 |
| 12 | 600 | 500 | 400 | 250 | 62 | 116.1 | 96.8 | 77.4 |
| 14 | 514.2 | 428.6 | 343 | 214.3 | 64 | 112.5 | 93.7 | 75 |
| 16 | 450 | 375 | 300 | 187.5 | 66 | 109 | 90.8 | 72.7 |
| 18 | 400 | 333.3 | 266.6 | 166.6 | 68 | 105.9 | 88.2 | 70.6 |
| 20 | 360 | 300 | 240 | 150 | 70 | 102.8 | 85.7 | .... |
| 22 | 327.2 | 272.7 | 218.1 | 136.3 | 72 | 100 | 83.3 | .... |
| 24 | 300 | 250 | 200 | 125 | 74 | 97.3 | 81 | .... |
| 26 | 277 | 230.8 | 184.5 | 115.4 | 76 | 94.7 | 78.9 | .... |
| 28 | 257.1 | 214.2 | 171.5 | 107.1 | 78 | 92.3 | 76.9 | .... |
| 30 | 240 | 200 | 160 | 100 | 80 | 90 | 75 | .... |
| 32 | 225 | 187.5 | 150 | 93.7 | 82 | 87.8 | 73.2 | .... |
| 34 | 212 | 176.5 | 141.1 | 88.2 | 84 | 85.7 | 71.4 | .... |
| 36 | 200 | 166.6 | 133.3 | 83.3 | 86 | 83.7 | ..... | .... |
| 38 | 189.5 | 157.9 | 126.3 | 78.9 | 88 | 81.8 | ..... | .... |
| 40 | 180 | 150 | 120 | 75 | 90 | 80 | ..... | .... |
| 42 | 171.5 | 142.8 | 114.2 | 71.4 | 92 | 78.3 | ..... | .... |
| 44 | 163.5 | 136.3 | 109 | ..... | 94 | 76.6 | ..... | .... |
| 46 | 156.6 | 130.5 | 104.3 | ..... | 96 | 75 | Lower speeds are limited by cost rather than by design. | |
| 48 | 150 | 125 | 100 | ..... | 98 | 73.5 | | |
| 50 | 144 | 120 | 96 | ..... | 100 | 72 | | |

D.C. motors and certain types of A.C. motors have a power factor of unity (100%); that is, kw. input = kva. input. However, the more common types of A.C. motors, such as the induction motor, have a *lagging* power factor of less than unity. For the effect of low power factor, and the means of improving it, see Kent's Mechanical Engineers' Handbook—Power, p. 15–53, forming Vol. 2 of this series.

The product of the efficiency and the power factor of a motor is the *apparent efficiency.* Thus

$$\text{kva. input} = \text{kw./P. F.} = (0.746\ \text{Hp.})/(E \times \text{P. F.}) = 0.746\ \text{Hp.}/E_a$$

where $E$ = efficiency; $E_a$ = apparent efficiency; P. F. = power factor.

**FULL-LOAD CURRENT.**—The full-load current of a motor is the current drawn from the line when the motor is carrying full load, with rated voltage and frequency (A.C. motors) applied. It may be calculated as follows, $I_{\text{max.}}$ being full-load current:

**D.C. Motors**

$$I_{\text{max.}} = (746 \times \text{Hp.})/(\text{Efficiency} \times \text{Voltage})$$

**Single-phase A.C. Motors**

$$I_{\text{max.}} = (746 \times \text{Hp.})/(\text{Efficiency} \times \text{Voltage} \times \text{Power Factor})$$

**Three-phase A.C. Motors**

$$I_{\text{max.}} = (746 \times \text{Hp.})/(1.73 \times \text{Efficiency} \times \text{Voltage} \times \text{Power Factor})$$

**Two-phase A.C. Motors, 4-wire** *

$$I_{\text{max.}} = (746 \times \text{Hp.})/(2 \times \text{Efficiency} \times \text{Voltage} \times \text{Power Factor}).$$

**TORQUES.**—The full-load torque of a motor is fixed by its horsepower and speed rating.

$$\text{Full-load torque in lb.-ft.} = 5250 \times \text{Hp./r.p.m.}$$

This is the basic torque of a motor. Other torques (see below) are often expressed as a percentage of full-load torque.

Other motor torques which are of importance are: *a.* Starting torque; *b.* Pull-up torque (A.C. motors); *c.* Breakdown torque (A.C. motors); *d.* Pull-in torque (synchronous motors); *e.* Pull-out torque (synchronous motors).

**Starting Torque** of a motor is the torque developed at zero speed. In A.C. motors, it is the minimum torque for all angular positions of the rotor, with rated voltage and frequency applied to the motor.

**Pull-up Torque** is the minimum torque developed by a motor during the period of acceleration from rest to full speed, with rated voltage and frequency applied to the motor. In some applications, the torque developed by a motor at all speed points from zero to full speed is of importance. A curve of torque plotted against speed (speed-torque curve) then is necessary.

**Rated Breakdown Torque** of an A.C. motor is the maximum torque which the motor will carry, with rated voltage and frequency applied to the motor, without an abrupt drop in speed. It indicates the peak load which the motor can carry without stalling or decelerating to a low speed.

**Pull-in Torque** of a synchronous motor is the maximum constant torque under which the motor will pull its connected inertia load into synchronism, at rated voltage and frequency, when its field excitation is applied. The speed to which a motor will bring its load depends on the power required to drive it; and whether or not the motor can pull the load into step from this speed, depends on the inertia of the revolving parts. Hence, pull-in torque cannot be determined unless the $WR^2$ of the external load and the torque of the load are known.

**Pull-out Torque** of a synchronous motor is the maximum sustained torque which the motor will develop continuously at synchronous speed with rated voltage, frequency, and normal excitation applied to the motor.

**STARTING CURRENT.**—It is characteristic of most motors that if full line voltage is impressed on their terminals at start (rotor at standstill) a current considerably in excess of the normal running current will be drawn from the line. This current usually is designated *locked rotor current*, and is defined as the current taken from the line with the rotor locked and with rated voltage (and frequency in the case of A.C. motors) applied. The term *free rotor current* sometimes is used; it is defined as 75% of the locked-rotor value.

In the case of D.C. motors, starting current must be held to certain limits (usually 150 to 300% of full load current), to prevent excessive burning of the motor commutator.

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* Where 2-phase motors are supplied from a 3-wire circuit, the current in the common wire is 1.41 times the current in either other wire.

This prevents starting these motors on full voltage, except in sizes of 1/2 horsepower and below. In the larger sizes, the voltage must be reduced until the motor has accelerated. This usually is accomplished by means of a resistor temporarily inserted in series with the armature winding.

The full-voltage starting current of A.C. motors varies over a wide range for the different types of motors and may be as high as 10 times normal full-load current. Hence, in some cases reduced-voltage starting is necessary to prevent distortion of the windings and resultant failure of the insulation. However, with a few exceptions A.C. motors have the windings adequately braced, and full-voltage starting is permissible so far as the motor is concerned. But, in any case, reduced-voltage starting frequently is necessary to prevent undue voltage disturbance on the power lines. The starting current, flowing through transformers, transmission lines and feeders may cause a drop in voltage of sufficient magnitude to result in dimming of lights supplied from the circuit or cause other motors to stall or trip off the line.

Most power companies have established regulations governing the starting current which may be drawn by motors connected to their distribution systems. These regulations take a variety of forms, but two general types are common. The first type specifies the maximum starting current that may be drawn by any motor of a given horsepower rating, supplied with a given class of service. The rules of Table 4 are typical. The second type is based on the capacity to serve a given installation which, in general, is determined by the total electrical load being supplied. Usually the total current drawn by the motor in starting is not limited but if this total exceeds the value specified, it is necessary to employ a starter which will build the current up in steps, each step not exceeding the specified value, and with a short time interval (usually 1/2 second) between steps. Table 5 is typical of the second type of starting current regulation.

**Table 4.—Typical Motor Starting Current Regulations**

Based on Horsepower Rating of Motor

| Rating of Motor, Hp. | Maximum Allowable Locked Rotor Current, Amperes | | | | | | |
|---|---|---|---|---|---|---|---|
| | Direct Current | | 60-Cycle Alternating Current | | | | |
| | | | Single-phase | Three-phase | | | |
| | 230 Volts | 550 Volts | 220 Volts | 220 Volts | 440 Volts | 550 Volts | 2200 Volts |
| 1/2 and below * | 12 | 5 | 20 | 26.6 | ..... | ..... | .... |
| 3/4 and 1 | 12 | 5 | 27 | 26.6 | ..... | ..... | .... |
| 1 1/2 | 18 | 7.5 | 30 | 36.6 | ..... | ..... | .... |
| 2 | 24 | 10 | 40 | 46.6 | ..... | ..... | .... |
| 3 | 36 | 15 | 60 | 60 | 30 | 24 | .... |
| 5 | 45 | 20 | 100 | 86.6 | 43.3 | 34.6 | .... |
| 7 1/2 | 38 | 30 | 110 | 115 | 58 | 46 | 10 |
| 10 | 90 | 40 | 147 | 141 | 70.5 | 56 | 13.3 |
| 15 | 135 | 60 | ... | 197 | 98.5 | 78.6 | 20 |
| 20 | 180 | 80 | ... | 251 | 125 | 101 | 26.6 |
| 25 | 228 | 100 | ... | 304 | 152 | 122 | 33.4 |
| 30 | 270 | 120 | ... | 360 | 180 | 144 | 40 |
| 40 | 350 | 160 | ... | 380 | 190 | 152 | 40 |
| 50 | 450 | 200 | ... | 400 | 200 | 160 | 40 |
| 60 Hp. and above, amperes per Hp. | 9 | 4 | .... | 8 | 4 | 3.2 | 0.8 |

* If starting current does not exceed 30 amperes, these motors may be connected to 110-volt single-phase lines.

**Table 5.—Typical Motor Starting Current Regulations**

Based on Total Electrical Load *

| Type of Motor | Locked Rotor Starting Kva. Limitation Per Step |
|---|---|
| Single-phase, 110-volt | 2.2 kva. for motors up to and including 1/4 Hp.<br>3.5 kva. for motors over 1/4 Hp. |
| Single-phase, 220-volt | 2.5 kva. plus 0.25 kva. per installed kw. |
| 3-phase, 220- or 440-volt | 50 kva. plus 0.5 kva. per installed kw. |
| 3-phase, 2200-volt or above | 30 kva. plus 1.0 kva. per installed kw. |

* Notes:

1. The starting current limitations corresponding to the starting kva. limitations given above can be derived as follows: Current = (kva. × 1000)/($C$ × motor voltage); $C$ = a constant = 1 for single-phase motors, and 1.73 for 3-phase motors.
2. The installed kw. includes the motor under consideration. To convert Hp. to kw., multiply Hp. value by 0.746.
3. This specification does not limit the total kva. that may be taken by the motor in starting but may require that the total be built up gradually, or in steps which do not exceed the value determined from the table.

Where no power company regulation exists, the limits of motor starting current are determined by the voltage variation permissible, and the drop in voltage which results from a given increase in current. To avoid any perceptible light flicker it is necessary to prevent a sudden change in voltage in excess of approximately 2% of normal voltage; but in many cases a considerably greater voltage change, particularly if occurring at rare intervals, will not be objectionable. So far as motors are concerned, a 10 or 15% momentary drop in voltage usually is not serious, although in some special applications the resulting change in motor speed may cause difficulty. The drop in voltage caused by a given increase in current is determined by the rating and characteristics of the generating equipment, and by the resistance and reactance of transformers and feeders supplying the motor.

It is characteristic of most motors that if the starting current is reduced, for example, by means of a reduced voltage starter, the starting torque will also be reduced. Hence, the starting torque of the motor and the torque required by the load must be considered, as well as the starting current. A high ratio of starting torque to starting current (called *torque efficiency*) obviously is desirable, where severe starting current restrictions exist. This ratio varies over a wide range for the different types of motors and may vary with the type of control equipment employed.

**MECHANICAL FORMS.**—In its simplest form a motor may consist simply of the stator and rotor windings with their necessary magnetic structures. Certain types of small motors often are supplied in this form. The rotor then is mounted on a shaft of the driven machine which also is provided with a suitable support for the stator. However, a motor usually is furnished with shaft, two bearings, and mounting feet forming part of the stator frame. The bearings usually are carried by *end shields* which bolt to each end of the frame. On very large motors pedestal-type bearings often are used. In this case, a base may be provided which supports the stator and the bearing pedestals, or individual sole plates may be provided for each bearing pedestal and for each side of the stator frame.

Motors sometimes are supplied without mounting feet, being supported from the end shield on the drive end. This shield has a machined flange or face for bolting to the driven machine. This same construction may be used to wholly or partially support the driven machine by the motor. In this case the motor feet are retained.

Motors used with belt drives, usually are provided with a sliding base which permits adjustment of the motor position, in a direction perpendicular to the shaft, for tightening the belt. Up to certain sizes the standard, two-bearing motors are suitable for belt drive. In the larger sizes, an outboard bearing is provided.

All motors are available in the standard horizontal shaft construction. Most types also are available in the vertical shaft construction. In this case the motor is supported by feet or a ring base which forms part of, or is bolted to, the lower end shield. Two types of vertical motors are available. The *no-thrust type* is designed to carry no thrust load in addition to the weight of the rotor and the half coupling. The *thrust type* is designed to carry a specified amount of additional thrust load. Certain types of vertical motors can be obtained in *hollow-shaft construction.* These are used primarily for deep-well pumping, the hollow shaft allowing the pump shaft to be brought to the top of the motor for convenient coupling to the motor shaft, and for adjustment of its length.

The combination of a motor with an enclosed speed-reducing gear, built as an integral unit and designated *gear-motor* is offered by several manufacturers. The sizes available range from $^1/_8$ Hp. to approximately 75 Hp., with output shaft speeds of 13 to 600 r.p.m. An 1800 r.p.m. motor usually is employed. Various types of gear reduction units are used including the internal planetary with concentric output shaft, the worm and wheel type with right-angle shaft, and the straight helical gear reduction with parallel offset shaft. The gear-motor has a considerably lower cost than that of a conventional motor of corresponding output shaft speed.

Oil-lubricated sleeve-type bearings are standard for most types of horizontal motors. These may be waste-packed or oil-ring-lubricated. Ball and roller bearings are standard for certain types of motors, and are available for most types at slightly higher cost. These usually are grease-lubricated, although oil lubrication often is used on large high-speed motors, and for the thrust bearing of the larger vertical motors.

**ENCLOSING AND PROTECTIVE FEATURES.**—A motor normally requires the circulation of air through its interior for removing the heat resulting from the motor losses. If there is no restriction to the flow of ventilating air other than that necessitated by mechanical construction the motor is designated as *open type.* Although this type is most common and the lowest in cost, a number of other types are available, offering various degrees of enclosure.

**A Protected Motor** is one in which all ventilating openings in the frame are protected with a wire screen, expanded metal or perforated covers, the openings in which do not exceed 1/2 sq. in. area, and are of such shape as not to permit the passage of a rod larger than 1/2 in. diameter; if the distance of exposed live parts from the guard is more than 4 in., openings may be 3/4 in. diameter. This construction is designed to prevent accidental contact with rotating or live parts. Its use is confined largely to D.C. motors.

**A Semi-protected Motor** is one in which part of the ventilating openings in the frame, usually in the top half, are protected as in the case of a protected machine, the others being left open.

**A Drip-proof Motor** is one in which the ventilating openings are so constructed that drops of liquid, or solid particles, falling on the machine at any angle not greater than 15 deg. from the vertical, cannot enter the machine either directly, or by striking and running along a horizontal or inwardly inclined surface.

**A Splash-proof Motor** is one in which the ventilating openings are so constructed that drops of liquid or solid particles falling on the machine or coming towards it in a straight line at any angle not greater than 100 deg. from the vertical cannot enter the machine either directly, or by striking and running along a surface.

**A Totally-enclosed Motor** is one so enclosed as to prevent exchange of air between the inside and the outside of the case, but not sufficiently enclosed to be termed air-tight. It is used where necessary to protect the motor from dirt, moisture, chemical fumes or other harmful ingredients of the surrounding atmosphere.

**A Totally-enclosed, Fan-cooled Motor** is a totally-enclosed machine equipped for exterior cooling by means of a fan or fans, integral with the machine but external to the enclosing parts. It serves the same purpose as the totally-enclosed motor. The fan cooling feature is desirable on larger motors to dissipate the heat.

**An Enclosed, Self-ventilated Motor** is one having openings for the admission and discharge of ventilating air, which is circulated by means integral with the machine, the machine being otherwise totally enclosed. The openings are so arranged that inlet and outlet ducts or pipes may be connected to them. This construction serves the same purpose as the totally enclosed type where air free from harmful ingredients can be obtained within a reasonable distance. It also is used where it is necessary to prevent heat, resulting from the motor losses, from being discharged into the motor room.

**An Enclosed, Separately-ventilated Motor** is one having openings for the admission and discharge of the ventilating air, which is circulated by means external to and not a part of the machine, which otherwise is totally enclosed. These openings are so arranged that inlet and outlet ducts or pipes may be connected to them. This construction, rather than the enclosed self-ventilated construction, is employed on low-speed machines whose rotor fans do not develop sufficient pressure to circulate the ventilating air through the required duct length.

Both the enclosed self-ventilated and the enclosed separately ventilated type machines may be provided with a closed system of ventilation; that is, the hot air discharged from the machine may be cooled and then recirculated through the machine. Cooling usually is accomplished by water circulated through surface air coolers. The closed system of ventilation generally is restricted to rather large machines.

**An Explosion-proof Motor** is one in an enclosing case, which is designed and constructed to withstand an explosion of a specified gas or dust which may occur within it, and to prevent the ignition of the specified gas or dust surrounding the motor by sparks, flashes or explosions which may occur within the machine casing. The National Electrical Code (Article 32) classifies locations in which hazardous gas or dust may be present, in accordance with the degree of hazard involved as follows:

Class I: Group *A*, Atmospheres containing acetylene; Group *B*, Atmospheres containing hydrogen; Group *C*, Atmospheres containing ethyl ether vapor; Group *D*, Atmospheres containing gasoline, petroleum, naphtha, alcohols, acetone, lacquer solvent vapors, and natural gas.

Class II: Group *E*, Atmospheres containing metal dust; Group *F*, Atmospheres containing carbon black, coal, or coke dust; Group *G*, Atmospheres containing grain dust.

Since it is impractical to design a motor which is gas tight, motors for Class I locations are so constructed as to permit explosions within the motor without ignition of the surrounding atmosphere. This is accomplished by: 1. Making the enclosing parts sufficiently strong to withstand safely the maximum pressure caused by ignition of the most flammable mixture of the gas involved; 2. Providing long fits with close clearances at all joints, and at the shaft opening, to cool the flame to a point where ignition of the outside atmosphere is prevented. Practically no motors are available for the more hazardous locations of Groups *A*, *B* and *C*. On the other hand, rather complete lines of motors suitable for Class I, Group *D* conditions are available. It is impractical to build motors of this type in the very large sizes, the present (1937) upper limit being about 200 Hp.

Motors for Class II locations are built totally-enclosed (or totally-enclosed fan-cooled), with all joints, etc. sufficiently tight to exclude the specific dust. Fairly complete lines of motors are available for Class II Group *G* locations.

The Underwriters' Laboratories, Inc. established and maintained by the National Board of Fire Underwriters offer a testing and inspection service available to manufacturers of equipment for hazardous locations. Motors which have been tested and inspected by them are provided with a label so indicating. Since most inspection authorities look to the Underwriters' Laboratories for guidance as to the suitability of equipment for hazardous locations, motors provided with this label generally are acceptable for use in the specified atmospheres anywhere in the United States. In general, motors with the Underwriters' label are available only for Class I, Group *D* and Class II, Group *G* locations.

Motors for installation in mines come under the jurisdiction of the U. S. Bureau of Mines. This body does not issue approvals covering a motor or controls, but approves only a complete power-using assembly consisting of a machine, such as pump, coal cutter, conveyor, etc., combined with the motor which drives it and the complete control equipment and wiring. However, certain types of motors are available, which have passed the inspection and tests of the Bureau of Mines, indicating that they are eligible for ultimate approval, without further tests, as part of a complete power-using assembly.

## 2. SMALL POWER MOTORS

Small power or fractional Hp. motors are motors built on frames having a continuous rating of less than 1 Hp., open type, at 1700–1800 r.p.m. Standard horsepower and speed ratings of small power motors are given in Table 6. The various types in general use are as follows: *a.* Single-phase motors: 1. Shaded-pole; 2. Split-phase; 3. Repulsion-start induction (and repulsion-induction); 4. Capacitor start; 5. Capacitor; 6. Series-wound; 7. Synchronous (non-excited).

*b.* Polyphase motors: 1. Squirrel-cage induction; 2. Synchronous (non-excited).

*c.* Direct-current motors: 1. Shunt-wound; 2. Series-wound; 3. Compound-wound.

*d.* Universal motors (operate from either A.C. or D.C.): 1. Series-wound; 2. Compensated-series-wound.

**SINGLE-PHASE MOTORS** are available in a variety of types, but practically all operate essentially as squirrel-cage induction motors. Since a normal single-phase

**Table 6.—Standard Horsepower and Speed Ratings of Small Power Motors**

| Brake Hp. | Direct-current | | | Alternating Current | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | 60 Cycles | | | | | |
| | Approx. Full-load r.p.m. | | | Synchronous r.p.m. | Approx. Full-load r.p.m. | Synchronous r.p.m. | Approx. Full-load r.p.m. | Synchronous r.p.m. | Approx. Full-load r.p.m. |
| 1 | 3450 | .... | .... | 3600 | 3450 | .... | .... | .... | .... |
| 3/4 | 3450 | 1725 | .... | 3600 | 3450 | 1800 | 1725 | .... | .... |
| 1/2 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/3 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/4 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/6 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/8 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/12 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/20 | 3450 | 1725 | 1140 | 3600 | 3450 | 1800 | 1725 | 1200 | 1140 |
| 1/30 | .... | 1725 | 1140 | .... | .... | 1800 | 1725 | 1200 | 1140 |
| 1/40 | .... | 1725 | 1140 | .... | .... | 1800 | 1725 | 1200 | 1140 |
| 1/50 | .... | 1725 | 1140 | .... | .... | 1800 | 1725 | 1200 | 1140 |
| | | | | 50 Cycles | | | | 25 Cycles | |
| 1 | .... | .... | .... | 3000 | 2875 | .... | .... | .... | .... |
| 3/4 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | .... | .... |
| 1/2 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/3 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/4 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/6 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/8 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/12 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/20 | .... | .... | .... | 3000 | 2875 | 1500 | 1425 | 1500 | 1425 |
| 1/30 | .... | .... | .... | .... | .... | 1500 | 1425 | 1500 | 1425 |
| 1/40 | .... | .... | .... | .... | .... | 1500 | 1425 | 1500 | 1425 |
| 1/50 | .... | .... | .... | .... | .... | 1500 | 1425 | 1500 | 1425 |

Smaller sizes usually are rated in watts output (746 watts = 1 Hp.).

induction motor does not develop any starting torque, it must be provided with an auxiliary starting winding or phase-shifting device. The essential difference between the various types of single-phase motors lies in the type of starting winding or phase-shifting device employed.

**Shaded-pole Single-phase Motors** have a squirrel-cage rotor winding and a concentrically-wound single-phase stator winding. A portion of each stator pole is "shaded" by a single turn of heavy copper wire. Current, which is out of phase with the current in the main winding, is induced in the shading coil. This results in a revolving magnetic field similar to that of a polyphase induction motor, and hence starting torque is developed. Motors of this type are inefficient, and have rather low starting torque. However, in the small sizes they are the most economical design. They are used in ratings up to 10 or 15 watts ($^1/_{75}$ to $^1/_{50}$ Hp.) for such applications as small fans, timing devices, instruments, relays, toys, sign flashers, small motion picture projectors, and miscellaneous control devices. They are designed for a particular direction of rotation which cannot be changed after the motor is built.

**Split-phase Motors** have squirrel-cage rotors and two-phase stators, with one of the phases wound for relatively high resistance. Since the ratios of resistance to reactance of the two phases are not equal, the currents in the phases are not of the same time phase, and approach the relations of a true two-phase motor. Hence, a revolving magnetic field is produced and starting torque is developed. As the losses in the high resistance winding are high, it must be disconnected when the motor gets up to speed. The disconnecting switch usually is operated by a centrifugal mechanism on the rotor. These motors are used in ratings from $^1/_{60}$ to $^1/_3$ Hp. on a variety of applications whose power requirements fall within this range; typical are advertising devices, fans, blowers, oil burners, kitchen appliances, machine tools, mechanical office equipment, pumps, textile and laboratory equipment, washing and ironing machines.

**Repulsion-start Induction and Repulsion-induction Motors** have a single-phase distributed winding on the stator, and a commutated winding, similar to that of a D.C. machine, on the rotor. The brushes bearing on the commutator are spaced 180 electrical degrees apart and are short circuited. Repulsion-start induction motors also have a centrifugal device which short-circuits all of the commutator segments when the motor has accelerated to a certain speed, converting the rotor winding to a squirrel-cage winding. Thus the motor starts as a repulsion motor, but runs as an induction motor. The centrifugal device sometimes is arranged to lift the brushes of the motor as well as to short circuit the commutator segments. Repulsion-induction motors, on the other hand, do not have any centrifugal device, but instead have a squirrel-cage winding on the rotor in addition to the commutated winding. The squirrel-cage winding is placed below the commutated winding and is arranged to have a high reactance. At low motor speeds (high slip frequency) very little current flows in the squirrel-cage winding, and the motor starts essentially as a repulsion motor. When the motor reaches full-speed, the frequency of the induced currents is low, so that considerable current flows in the squirrel-cage winding and the motor operates essentially as an induction motor. Repulsion-start induction or repulsion-induction motors generally are offered by manufacturers in ratings of $^1/_6$ Hp. and larger. High starting torque, low starting current, and relatively low cost are important features. Typical applications are pumps, compressors, machine tools, hoists, conveyors, garage and service station equipment, and woodworking machinery.

**The Capacitor-start Motor** is provided with a squirrel-cage rotor, and with a main winding and an auxiliary winding on the stator. A capacitor (usually mounted on top of the motor) is connected in series with the auxiliary winding to provide the necessary shift in time phase of the current flowing through it. Since the capacitor is intermittently rated it must be disconnected for normal operation. This usually is accomplished by a switch operated by a centrifugal mechanism on the rotor. Capacitor-start motors generally are available in ratings of from $^1/_8$ to $^1/_4$ Hp., and are used for such applications as oil burners, stokers, pumps, and household appliances.

**The Capacitor Motor** is provided with a continuously rated capacitor, and, therefore, the auxiliary winding with the capacitor in series may be left in the circuit during running as well as starting. The capacitor is selected to give best operation at full speed, *i.e.*, maximum efficiency and minimum noise and vibration. If the same capacitance is used during the starting and running periods the motor develops only 40 to 60% starting torque and is known as the *low-torque* capacitor motor. The *high-torque* capacitor motor, which develops 275 to 350% starting torque, is obtained by increasing the capacitance in series with the auxiliary winding during the starting period. This usually is accomplished either by: 1. Providing an additional intermittently-rated capacitor which is connected in parallel with the main capacitor during the starting period; 2. Energizing the capacitor through an auto-transformer, provided with two taps, which allows the use of a higher voltage ratio during the starting period. The change from starting to running connections on the capacitor is accomplished by a switch operated by a centrifugal mechanism on the rotor, or a relay is used, operated by the rise in voltage across the auxiliary winding as the motor comes up to speed. High power-factor and efficiency, and quiet operation are important features of capacitor motors. They generally are available in ratings of $^1/_8$ Hp. and larger. The low-torque capacitor motor can be used only on such applications as direct-connected fans which require little starting torque. The high-torque capacitor motor is a general purpose machine of increasing popularity. Typical applications include reciprocating compressors, pumps, fans, and blowers. The capacitor motor frequently is used for variable speed operation of fans, the speed change being accomplished by varying the voltage impressed on the motor terminals.

**The Series-wound Single-phase Motor** is similar in construction and principle of operation to the series-wound direct-current motor. It employs a stationary field winding and a commutated rotor (armature winding), the two being connected in series. Like the series D.C. motor, it is of the varying speed type, that is, its speed increases as the load decreases. Usual full-load speeds range from 1500 to 5000 r.p.m.; no-load speeds range from 150 to 200% of full-load speeds. The principal advantages of series motors are their extreme compactness and light weight for the power delivered. Hence, they are particularly suitable for application to portable devices, such as vacuum cleaners and portable tools of all types. Since they usually are " built-in " the power-using device, they frequently are supplied as " motor parts," that is, the bare stator and rotor, without bearings or feet, the bearings and stator support being incorporated in the driven machine.

**Synchronous Motors** in the small power sizes usually are built as *reluctance motors.* The stators are similar to those of single-phase induction motors, and may be of the shaded-pole, split-phase, or capacitor type. The rotor has a squirrel-cage winding, and the core is shaped to provide projections (salient poles) corresponding to the number of poles for which the stator is wound. The motor starts as an induction motor, but upon reaching a speed near synchronism, it " pulls into step," due to the effect of the salient poles, and operates at exactly synchronous speed. Unlike the large-power synchronous motor, which has a field winding on the rotor supplied with D.C. excitation, and which operates at unity or leading power-factor with high efficiency, the reluctance motor operates at a lagging power-factor and has a rather low efficiency. Hence, it is used only where exact synchronous speed is required. Typical applications include electric-clocks, time switches, relays, and recording meters.

Another type of synchronous motor available in the small-power sizes is the *hysteresis motor.* Its construction is similar to that of the reluctance motor except that the rotor is perfectly cylindrical and does not have a squirrel-cage winding. Its operation depends on the permanent magnetism induced in the rotor by the magnetic field of the stator. It develops a constant torque from zero to synchronous speed. It is used principally for phonograph turntable drive.

**POLYPHASE MOTORS.—Squirrel-cage Induction Type Motors** meet the requirements of practically all applications in the small power field. They develop a high starting torque with a starting current well within the limits of polyphase systems. Their efficiencies are slightly higher than those of single-phase motors. Polyphase motors are generally available in sizes of $^1/_6$ Hp. and larger at 110, 220, 440, and 550 volts.

**Synchronous** motors of the reluctance type are employed where exact synchronous speed is required. They are similar to the single-phase motors of this type except that no starting winding or phase-splitting device is necessary.

**High-frequency Polyphase Motors** often are used on portable tools, textile machinery, and other high speed motor applications where the number required justifies the installation of special generating or conversion equipment to supply the high-frequency power. A frequency of 180 cycles usually is employed, giving a synchronous speed (2-pole motors) of 10,800 r.p.m. The advantages are a saving in cost, size, and weight over 60-cycle induction motors. As compared to the series motor, the high-frequency polyphase motor requires less maintenance, since it has no brushes or commutator and has a higher efficiency and better speed regulation.

**DIRECT-CURRENT MOTORS** of the **shunt-wound** or **compound-wound** type are used for most applications. In the smaller sizes (approximately $^1/_{20}$ Hp. and below) shunt-wound motors are standard, while in the larger sizes, the standard is compound-wound. The **series-wound** motor is used where its varying speed characteristics and high no-load speed is not objectionable. It develops a very high starting torque, but has a limited output at the higher speeds.

**UNIVERSAL MOTORS,** which operate from either single-phase A.C. or D.C. power, are all of the series-wound type; that is, they have a field winding on the stator which is connected in series with a commutated winding on the rotor. The **straight series-wound**

motor has a salient pole field winding on the stator, like the ordinary D.C. motor, while the **compensated-series-wound** motor has a distributed winding arranged in slots around the periphery of the stator. The latter has the advantages of better speed regulation and commutation, and higher starting torque, while the former is simpler in construction and more easily ventilated. In general, the straight series-wound motor is used in sizes up to approximately 1/3 Hp., while the compensated-series-wound motor is used in the larger sizes. It is not feasible to build universal motors for low speed operation, the minimum full load speed ranging from 1800 r.p.m. on the larger sizes to 5000 r.p.m. on the smaller sizes. Typical full-load speed ratings are 5000, 7500 and 10,000 r.p.m., with no-load speeds ranging from 12,000 to 18,000 r.p.m. Universal motors are used on a variety of applications including portable tools, office appliances, vacuum cleaners, motion picture projectors, kitchen appliances, sewing machines and portable equipment in general.

## Operating Characteristics

**TEMPERATURE RISE.**—The standard temperature rise of small power motors, when operating under normal conditions, is 40° C., measured by thermometer, for open and drip-proof motors, and 55° C. for totally-enclosed motors. The motors frequently are offered in two general types designated " For short annual service " and " For long annual service." Both types can be operated continuously, but the latter type is recommended for applications where the motor must operate more than 1000 hours annually. They are designed especially for good running characteristics, such as high efficiency and power-factor. In the short annual service type, these characteristics may be sacrificed to some extent to obtain higher torques and lower costs.

**EFFICIENCY.**—Minimum efficiencies, power-factors, and apparent efficiencies of single-phase, 60-cycle, induction motors as established by N.E.M.A. are given in Table 7. The standard motors offered by the manufacturers have characteristics equaling or exceeding these values. In general, the efficiencies of polyphase induction motors are somewhat higher than those of single-phase motors of corresponding ratings, while the efficiencies of direct-current motors are slightly lower than those of single-phase motors.

**STARTING CURRENT.**—The maximum locked-rotor starting currents of single-phase motors as established by N.E.M.A. are given in Table 8. The locked-rotor currents

**Table 7.—Minimum Efficiencies, Power-factors, and Apparent Efficiencies of 60-cycle, Single-phase, Small Power Motors**

Motors operated at rated voltage, frequency, and load

| Poles | 2 | 4 | 6 | 8 | 2 | 4 | 6 | 8 | 2 | 4 | 6 | 8 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| R.p.m. | 3600 | 1800 | 1200 | 900 | 3600 | 1800 | 1200 | 900 | 3600 | 1800 | 1200 | 900 |
| Hp. | Efficiency, % | | | | Power-factor | | | | Apparent Efficiency, % * | | | |
| | Long Hour Service | | | | | | | | | | | |
| 1/8 | 45 | 53 | 45 | 38 | 57 | 52 | 43 | 36 | 28 | 30 | 21 | 15 |
| 1/6 | 49 | 58 | 49 | 42 | 62 | 56 | 46 | 38 | 34 | 36 | 25 | 18 |
| 1/4 | 53 | 62 | 53 | 45 | 66 | 60 | 49 | 40 | 39 | 42 | 29 | 20 |
| 1/3 | 54 | 63 | 54 | 46 | 67 | 61 | 50 | 41 | 41 | 44 | 31 | 22 |
| 1/2 | 55 | 65 | 55 | 47 | 69 | 63 | 52 | 43 | 44 | 47 | 33 | 23 |
| 3/4 | 57 | 67 | 57 | 49 | 72 | 65 | 53 | 44 | 46 | 49 | 34 | 24 |
| | Short Hour Service | | | | | | | | | | | |
| 1/8 | 35 | 41 | 35 | 30 | 55 | 50 | 41 | 34 | 23 | 24 | 17 | 12 |
| 1/6 | 39 | 46 | 39 | 33 | 57 | 52 | 43 | 35 | 25 | 27 | 19 | 13 |
| 1/4 | 43 | 51 | 43 | 37 | 62 | 56 | 46 | 38 | 30 | 32 | 22 | 15 |
| 1/3 | 46 | 54 | 46 | 39 | 64 | 58 | 48 | 39 | 33 | 35 | 26 | 18 |
| 1/2 | 49 | 58 | 49 | 42 | 66 | 60 | 49 | 40 | 37 | 39 | 27 | 19 |
| 3/4 | 52 | 61 | 52 | 44 | 68 | 62 | 51 | 42 | 39 | 42 | 29 | 20 |

* Apparent efficiency = (efficiency × power-factor)/100.

**Table 8.—Maximum Locked-rotor Starting Currents of Single-phase Small Power Motors ***

60 cycles, 900 to 3600 r.p.m. inclusive

| Horsepower | 1/6 and smaller | 1/4 | 1/3 | 1/2 | 3/4 |
|---|---|---|---|---|---|
| | Current in amperes | | | | |
| 110 volts | 20 | 23 | 31 | 45 | 61 |
| 220 volts | 10 | 11 1/2 | 15 1/2 | 22 1/2 | 30 1/2 |

* Excepting the split-phase motors used on washing and ironing machines.

of polyphase squirrel-cage induction motors vary between 400 and 800% of full-load current.

**TORQUE.**—Minimum torque values for 60-cycle induction motors as established by N.E.M.A. are given in Table 9. Typical speed-torque curves are shown in Figs. 1 and 2. D.C. motors, shunt- or compound-wound, develop 500% or more starting torque on full voltage.

**VOLTAGE VARIATION.**—Small power motors should operate successfully at rated load (and rated frequency for A.C. motors) at any voltage not more than 10% above or below normal, but not necessarily in accordance with the standards of performance established for operation at normal rating. An exception to this rule applies to the series-type universal motors for which the maximum allowable voltage variation is 6%. The starting currents of all types are proportional to the applied voltage and the starting,

**Table 9.—Minimum Torque Values for 60-cycle Small Power Motors**

In percent of full-load torque

| Hp. | Rating, 60 Cycles, 1725 r.p.m. | | | Rating, 60 Cycles, 1140 r.p.m. | | |
|---|---|---|---|---|---|---|
| | Locked-rotor | Pull-up | Breakdown | Locked-rotor | Pull-up | Breakdown |
| | | | **Split-phase and Capacitor-start Motors** | | | |
| 1/8 | 150 | 150 | 200 | 125 | 125 | 175 |
| 1/6 | 150 | 150 | 200 | 125 | 125 | 175 |
| 1/4 | 90 | 90 | 185 | 75 | 75 | 175 |
| | | | **Repulsion-start Induction Motors** | | | |
| 1/8 | 350 | 200 | 200 | 300 | 150 | 185 |
| 1/6 | 350 | 200 | 200 | 300 | 150 | 185 |
| 1/4 | 350 | 200 | 200 | 300 | 150 | 185 |
| 1/3 | 350 | 200 | 200 | 300 | 150 | 185 |
| 1/2 | 350 | 200 | 200 | 300 | 150 | 185 |
| 3/4 | 350 | 200 | 200 | | | |
| | | | **High-torque Capacitor Motors** | | | |
| 1/8 | 350 | 200 | 200 | 300 | 185 | 185 |
| 1/6 | 350 | 200 | 200 | 300 | 185 | 185 |
| 1/4 | 350 | 200 | 200 | 300 | 185 | 185 |
| 1/3 | 325 | 200 | 200 | 300 | 185 | 185 |
| 1/2 | 300 | 200 | 200 | 300 | 185 | 185 |
| 3/4 | 275 | 200 | 200 | | | |
| | | | **Polyphase Induction Motors** | | | |
| All | 150 | . . . | 200 | 135 | . . . | 200 |

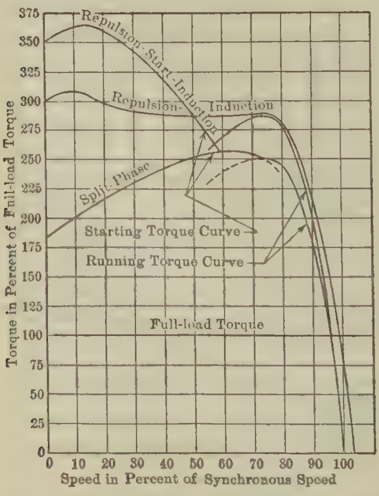


FIG. 1. Typical Speed-torque Curves of Small Power Split-phase, Repulsion Induction, and Repulsion-start Induction Motors

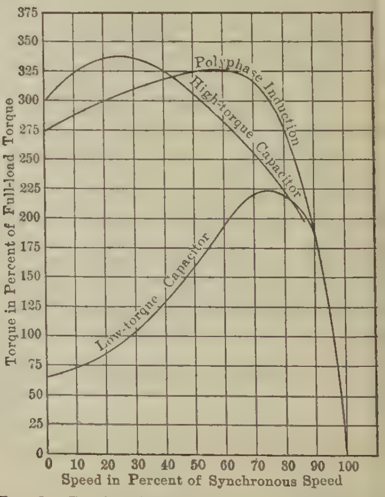


FIG. 2. Typical Speed-torque Curves of Small Power Polyphase Induction, and Capacitor Motors

pull-up, and breakdown torques are approximately porportional to the square of the voltage.

### Mechanical Features

Small power motors may be obtained as bare rotors and stators (with or without feet) for mounting directly on the driven machine. However, the usual construction consists of a stator frame provided with mounting feet and end-shield type bearings. Motors with the drive end-shield machined for bolting to the driven machine are available. Vertical motors, as well as the standard horizontal motors, are available in all types.

Wool-yarn-packed sleeve bearings are standard, but ball bearings can be obtained at extra cost. Vertical motors usually are provided with ball bearings, as are horizontal motors which are required to take considerable axial thrust.

Single-phase induction motors inherently produce a pulsating torque, resulting in vibration of the motor frame which, if transmitted to the motor support, may result in objectionable noise. Hence, such motors frequently are furnished with a resilient rather than a rigid connection between the motor frame and the motor base. This resilient connection may consist of rubber cushions, or of metal springs of sufficient flexibility to prevent or greatly reduce the transmission of vibration to the supporting structure. Some manufacturers offer motors with a resilient mounting combined with an automatic belt tightener base, which maintains a belt tension just sufficient to prevent slippage at all loads throughout the operating range.

Gear-motors of the various electrical types are available in ratings from 1/8 to 3/4 Hp. with output shaft speeds of 11 to 575 r.p.m. Both concentric and right-angle shaft types are available. The latter employ worm-and-wheel reduction units and have lower costs, but considerably lower efficiency. The horsepower output rating of a gear-motor may be considerably less than the rating of the motor itself, the difference depending on the type of gear and the speed ratio.

Standard motors are of the open or drip-proof construction, but practically all types can be obtained in the totally-enclosed construction. Most types also are available in the explosion-proof construction for Class I, Group *D* locations. See p. 25–17.

**WEIGHTS AND DIMENSIONS.**—Approximate net weights and overall dimensions of open, horizontal motors for long annual service are given in Fig. 4 and Table 10. The dimensions of totally-enclosed, and enclosed motors, excepting the explosion-proof type, are essentially the same as those of open motors. Explosion-proof motors usually have slightly greater overall dimensions. Motors for short annual service are, in general, slightly smaller than the motors for long annual service. The overall dimensions of vertical motors do not differ greatly from those of horizontal machines when shifted 90 deg.

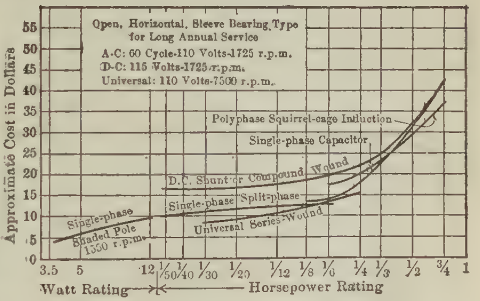


Fig. 3. Approximate 1936 Cost of Small Power Motors

**COSTS.**—Approximate costs (1936) of open, horizontal, sleeve bearing motors are given in Fig. 3. The figures shown are for 110-volt A.C., and 115-volt D.C. motors having a full-load speed of 1725 r.p.m. (excepting the universal motors). Shaded-pole, split-phase, and universal motors for 220 volts are slightly higher in cost, as are 230 volt D.C. motors. Polyphase motors are offered for 440 and 550 volts, as well as 110 and 220 volts, but the former are slightly higher in cost.

In general, 3450 r.p.m. motors are more costly than 1725 r.p.m. machines, although in the larger sizes the reverse is true. All 1140 r.p.m. machines are more costly than those rated 1725 r.p.m. Universal motors rated 5000 r.p.m. are more costly, and those rated 10,000 r.p.m. are less costly, than 7500 r.p.m. machines. Ball bearings, enclosed construction, vertical mounting, and special mechanical features result in increased cost.

## 3. SINGLE-PHASE LARGE POWER MOTORS

Standard horsepower and speed ratings of large power, single-phase motors are given in Table 11. The following types are available: *a*. Repulsion-start induction (and repulsion-induction); *b*. Capacitor; *c*. Brush-shifting repulsion; *d*. Polyphase induction with phase-shifting starter.

For details of construction and principle of operation of types *a* and *b*, see p. 25–19.

**Repulsion-start Induction and Repulsion-induction Motors** are available in sizes up to 10 Hp. Their high starting torque, low starting current, and low cost make them the most popular of all types. They are used on practically all applications requiring single-phase motors.

**Table 10.—Approximate Weights and Dimensions of Small Power Motors**

Open horizontal type, for long annual service; 60 cycles, 110 and 220 volts A.C.; 115 and 230 volts D.C.

| Hp. Rating | Full-load r.p.m. | Approx. Weight, lb. | Approx. Dimensions, in. A | C | O | P |
|---|---|---|---|---|---|---|
| **Single-phase:** Shaded Pole | | | | | | |
| 1/200 | 1550 | 2 | 3 | 4 3/4 | 3 | 3 |
| 1/150 | 1550 | 3 1/2 | 3 1/2 | 5 1/4 | 3 1/2 | 3 1/2 |
| 1/60 | 1550 | 5 | 4 | 5 1/2 | 4 | 3 1/2 |
| **Single-phase:** Split-phase; Capacitor Start *<br>**Polyphase:** Squirrel-cage Induction<br>**D.C.:** Compound-wound | | | | | | |
| 1/8 | 1725 | 23 | 6 1/2 | 10 | 6 3/4 | 6 1/2 |
| 1/8 | 1140 | 27 | 6 1/2 | 10 1/4 | 6 3/4 | 6 1/2 |
| 1/6 | 3450 | 25 | 5 3/4 | 10 1/4 | 5 3/4 | 5 1/2 |
| 1/6 | 1725 | 27 | 6 1/2 | 10 1/4 | 6 3/4 | 6 1/2 |
| 1/6 | 1140 | 30 | 6 1/2 | 10 3/4 | 6 3/4 | 6 1/2 |
| 1/4 | 3450 | 25 | 5 3/4 | 10 1/4 | 5 3/4 | 5 1/2 |
| 1/4 | 1725 | 30 | 6 1/2 | 10 3/4 | 6 3/4 | 6 1/2 |
| Universal Series-wound | | | | | | |
| 1/50 | 5,000 | 2 | 3 | 5 1/4 | 3 | 3 |
| 1/30 | 7,500 | 2 | 3 | 5 1/4 | 3 | 3 |
| 1/30 | 5,000 | 2 1/2 | 3 | 5 1/2 | 3 | 3 |
| 1/20 | 7,500 | 2 1/2 | 3 | 5 1/2 | 3 | 3 |
| 1/12 | 10,000 | 3 1/2 | 3 1/2 | 5 3/4 | 3 1/2 | 3 1/2 |
| 1/12 | 7,500 | 4 1/2 | 3 1/2 | 6 | 3 1/2 | 3 1/2 |
| 1/8 | 5,000 | 6 1/2 | 4 | 7 3/4 | 4 | 4 |
| 1/4 | 10,000 | 6 1/2 | 4 | 7 3/4 | 4 | 4 |
| 1/4 | 7,500 | 8 | 4 | 7 3/4 | 4 | 4 |
| 1/3 | 10,000 | 8 | 4 | 8 | 4 | 4 |

| Hp. Rating | Full-load r.p.m. | Approx. Weight, lb. | Approx. Dimensions, in. A | C | O | P |
|---|---|---|---|---|---|---|
| **Single-phase:** Split-phase<br>**D.C.:** Shunt- or Compound-wound | | | | | | |
| 1/60 | 1725 | 7 | 4 | 6 1/2 | 4 | 4 |
| 1/40 | 1140 | 12 | 4 3/4 | 7 3/4 | 5 | 4 3/4 |
| 1/30 | 1725 | 10 | 4 3/4 | 7 | 5 | 4 3/4 |
| 1/30 | 1140 | 17 | 5 3/4 | 9 | 5 3/4 | 5 1/2 |
| 1/20 | 3450 | 12 | 4 3/4 | 7 3/4 | 5 | 4 3/4 |
| 1/20 | 1725 | 10 | 4 3/4 | 7 | 5 | 4 3/4 |
| 1/20 | 1140 | 17 | 5 3/4 | 9 | 5 3/4 | 5 1/2 |
| 1/15 | 3450 | 12 | 4 3/4 | 7 3/4 | 5 | 4 3/4 |
| 1/15 | 1725 | 17 | 5 3/4 | 9 | 5 3/4 | 5 1/2 |
| 1/15 | 1140 | 24 | 6 1/2 | 10 1/4 | 6 3/4 | 6 1/2 |
| 1/10 | 3450 | 16 | 5 3/4 | 8 3/4 | 5 3/4 | 5 1/2 |
| 1/10 | 1725 | 17 | 5 3/4 | 9 | 5 3/4 | 5 1/2 |
| 1/10 | 1140 | 24 | 6 1/2 | 10 1/4 | 6 3/4 | 6 1/2 |
| 1/8 | 3450 | 16 | 5 3/4 | 8 3/4 | 5 3/4 | 5 1/2 |
| **Single-phase:** Capacitor *<br>**Polyphase:** Squirrel-cage Induction<br>**D.C.:** Compound-wound | | | | | | |
| 1/4 | 1140 | 35 | 6 1/2 | 12 3/4 | 6 3/4 | 6 1/2 |
| 1/3 | 3450 | 40 | 6 1/2 | 11 1/2 | 6 3/4 | 6 1/2 |
| 1/3 | 1725 | 30 | 6 1/2 | 11 1/4 | 6 3/4 | 6 1/2 |
| 1/3 | 1140 | 55 | 8 3/4 | 13 3/4 | 8 | 7 1/2 |
| 1/2 | 3450 | 40 | 6 1/2 | 11 1/2 | 6 3/4 | 6 1/2 |
| 1/2 | 1725 | 50 | 8 3/4 | 13 3/4 | 8 | 7 1/2 |
| 1/2 | 1140 | 70 | 9 1/2 | 14 1/2 | 9 | 8 1/2 |
| 3/4 | 3450 | 60 | 8 3/4 | 14 1/4 | 8 | 7 1/2 |
| 3/4 | 1725 | 60 | 9 1/2 | 14 | 9 | 8 1/2 |
| 1 | 3450 | 60 | 8 3/4 | 14 1/4 | 8 | 7 1/2 |

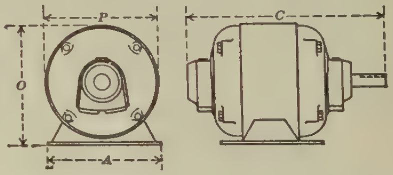


Fig. 4.

* Where capacitor is mounted on top of motor the overall height O is increased by 2 1/2 to 4 in.

**Table 11.—Standard Horsepower and Speed Ratings of Single-phase Large Power Motors**

| Hp. Rating | 60 Cycles | | | | 50 Cycles | | | | 25 Cycles | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Synchronous r.p.m. | | | | Synchronous r.p.m. | | | | Synchronous r.p.m. | |
| 1/2 | .... | .... | .... | 900 | .... | .... | .... | 750 | .... | ... |
| 3/4 | .... | .... | 1200 | 900 | .... | .... | 1000 | 750 | 1500 | ... |
| 1 | .... | 1800 | 1200 | 900 | .... | 1500 | 1000 | 750 | 1500 | ... |
| 1 1/2 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | ... |
| 2 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 3 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 5 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 7 1/2 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 10 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 15 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 20 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |
| 25 | 3600 | 1800 | 1200 | 900 | 3000 | 1500 | 1000 | 750 | 1500 | 750 |

**Capacitor Motors** are more costly than the repulsion-start induction or repulsion-induction type, but their simple construction, high efficiency and power-factor, and inherently quiet operation are features which frequently justify their use. In addition to their general-purpose capacitor motors, which are of the high torque type and are used on such applications as pumps, compressors, conveyors and machine tools, manufacturers offer special low-torque and high-torque capacitor motors for fan drives. The former are for use on direct-connected fans only, while the latter are suitable for belt-driven fans. Both types are specially designed for quiet operation, and with a somewhat lower full-load speed (higher slip) than the general purpose motors. The latter feature allows adjustable speed operation down to approximately 50% speed, by means of suitable control which varies the voltage impressed on the motor. Multi-speed capacitor motors also can be obtained.

**The Brush-shifting Repulsion Motor** is similar in construction to the repulsion-induction motor except that no squirrel-cage winding is provided on the rotor. Instead, a compensating transformer, connected in series with the stator winding, supplies power to brushes located at 90 deg. from the main brushes. By varying the position of the main brushes, the speed of the motor can be varied through a range of 2.5 to 1 on a constant torque load. The motor has varying speed characteristics, the no-load speed being approximately 60% above synchronous speed. Fig. 5 shows typical speed-torque curves for various brush settings. Ratings available range from $^1/_4$ to 3 Hp. at 1800/720 r.p.m. Typical applications include job printing presses, automobile generator test benches, balancing, polishing, and winding machines.

Single-phase motors are seldom used in sizes above 10 Hp. Hence, none of the types described above is available in the higher ratings. Where a larger single-phase motor is required, it is usual to use a polyphase induction motor with a split-phase starter of the resistance-reactance type. The horsepower output of a polyphase induction motor operating single-phase is approximately $^1/_2$ of its normal rating.

## Operating Characteristics

**TEMPERATURE RISE.**—The standard temperature rise of large power, single-phase motors when operating under normal rated conditions is 40° C., measured by thermometer, for the general-purpose open type; for splash-proof motors, it is 50° C.; for totally-enclosed motors, it is 55° C. Standard open motors having a temperature rise of 40° C. usually are given a *service factor* of 1.15; that is, the manufacturers guarantee successful operation at 1.15 times rated load without injurious heating, although the efficiency and power-factor may be slightly different from the rated load values.

**EFFICIENCY.**—Typical full-load efficiencies and power-factors of repulsion-induction type motors are given in Fig. 6. Capacitor motors will, in general, have slightly higher

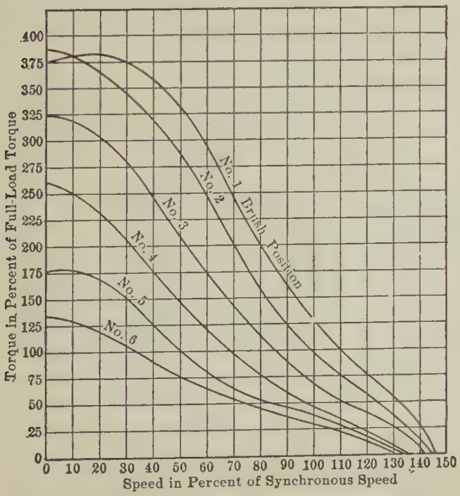


FIG. 5. Speed-torque Curves of Typical Brush Shifting Repulsion Motors

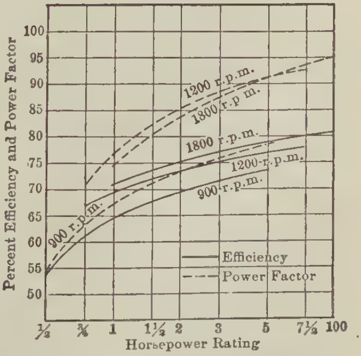


FIG. 6. Approximate Full-load Efficiency and Power Factor of 110- and 220-Volt, 60 Cycle, Repulsion Induction Motors

efficiencies and about the same power-factors, while repulsion-start induction motors have the same efficiencies but lower power-factors.

**TORQUE.**—Typical starting and breakdown torque values for constant-speed single-phase motors are given in Table 12. **Starting currents** range between 300 and 450% of full-load current. Brush-shifting repulsion motors have starting torques of 250 to 300% with the brushes in the high-speed position, reducing to 125% as the brushes are shifted to the low speed position. Their breakdown torques are approximately 300% of full-load torque.

**VOLTAGE VARIATION.**—Large power, single-phase motors should operate successfully at rated load and frequency at any voltage not more than 10% above or below normal but not necessarily in accordance with the standards of performance established for operation at normal rating. The starting currents are proportional to the applied voltage, and the starting, pull-up, and breakdown torques are proportional to the square of the voltage.

## Mechanical Features

Large power, single-phase motors usually are furnished with feet on the stator frame, and with end-shield type bearings. They are also available with the drive end shield machined for bolting to the driven machine. Vertical motors, as well as standard horizontal motors, can be obtained. The former may be obtained with ring, four-leg, or tripod base.

Sleeve bearings, wool-yarn-packed or oil-ring-lubricated, are standard for most horizontal machines. Ball bearings are standard on vertical and totally-enclosed motors, and can be obtained on other machines at extra cost. All motors can be obtained with pulleys and sliding bases for two-bearing belt drive. Gear motors are available in ratings from 1 to 5 Hp. with output shaft speeds of 13 to 600 r.p.m.

Standard motors are of the open type, but splash-proof, totally-enclosed, fan-cooled, and explosion-proof (Class I, Group *D* conditions) motors also are available.

**WEIGHTS AND DIMENSIONS.**—The N.E.M.A. has standardized mounting dimensions for various types of motors, indicating these dimensions by a series of frame numbers. These are used by all manufacturers for motors which have the standardized dimensions. The frame numbers and weights assigned to the various ratings of open and totally-enclosed, fan-cooled, 60- and 50-cycle, 110- and 220-volt single-phase motors are given in Table 13. The mounting dimensions corresponding to these frame numbers are given in Table 14 and Fig. 7 for horizontal machines, and Table 15 and Fig. 8 for vertical machines.

**Table 12.—Typical Torque Values for Large Power, 60-cycle, Single-phase Motors**

| Synchronous r.p.m. | General-purpose Motors * | | Capacitor Fan Motors | | | |
|---|---|---|---|---|---|---|
| | | | High Torque | | Low Torque | |
| | Starting Torque | Breakdown Torque | Starting Torque | Breakdown Torque | Starting Torque | Breakdown Torque |
| | Percent of Full-load Torque | | | | | |
| 3600 | 300 | 200 | 220 | 125–150 | 40 | 125–150 |
| 1800 | 300 | 200 | 220 | 125–150 | 40 | 125–150 |
| 1200 | 275 | 190 | 185 | 125–150 | 35 | 125–150 |
| 900 | 225 | 190 | 150 | 125–150 | 30 | 125–150 |

* Repulsion-start-induction, repulsion-induction, or capacitor.

**Table 13.—N.E.M.A. Standardized Frame Sizes of Open and Totally-enclosed Fan-cooled, Single-phase Motors**

110 and 220 volts, 50 and 60 cycles

| Poles | Rev. per Min. | | Horsepower | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 50 Cycles | 60 Cycles | 1/2 | 3/4 | 1 | 1 1/2 | 2 | 3 | 5 | 7 1/2 |
| | | | Frame Number * | | | | | | | |
| 2 | 3000 | 3600 | ... | ... | ... | 204 | 224 | 224 | 225 | 254 |
| 4 | 1500 | 1800 | ... | ... | 204 | 224 | 225 | 225 | 254 | ... |
| 6 | 1000 | 1200 | ... | 204 | 224 | 225 | 254 | ... | ... | ... |
| 8 | 750 | 900 | 224 | 225 | 254 | 254 | ... | ... | ... | ... |

Approximate Weights in Pounds of Open, Horizontal, Single-phase Motors

| Frame No. | Motor Only | Base | Pulley | Frame No. | Motor Only | Base | Pulley |
|---|---|---|---|---|---|---|---|
| 204 | 80 | 20 | 1 | 225 | 135 | 25 | 3 |
| 224 | 110 | 25 | 3 | 254 | 205 | 30 | 4 |

* See Tables 14 and 15 for standard mounting dimensions.

**Table 14.—Dimensions and Tolerances of N.E.M.A. Standardized Frames for Horizontal A.C. and D.C. Motors**

See Fig. 7. All dimensions in inches

| Frame No. | Pulley | | | Key | | *AA* Min. | *A* Max. | *B* Max. | *D* Note 1 | *E* | *F* | *BA* | *H* | *N-W* | *U* | *V* Min. | *AL* | *AM* | *AO* | *AR* | *AU* | *AX* | *AY* Max. | *XA* | *XB* † |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Diam. | Width Over-all | Length of Hub | Width | Thickness | | | | | | | | | | | | | | | | | | | | |
| 204 | 3 | 3 | 2 | 3/16 | 3/16 | 3/4 | 10 | 8 1/2 | 5 | 4 | 3 1/4 | 3 1/8 | 13/32 | 2 1/4 | 3/4 | 2 | 14 | 12 | 5 | 5 1/4 | 1/2 | 1 3/4 | 1/2 | 1/4 | 3 |
| 224 | 4 | 3 1/2 | 2 3/4 | 1/4 | 1/4 | 3/4 | 11 | 8 3/4 | 5 1/2 | 4 1/2 | 3 3/8 | 3 1/2 | 13/32 | 3 | 1 | 2 3/4 | 15 1/2 | 12 1/4 | 5 1/2 | 5 3/8 | 1/2 | 1 3/4 | 1/2 | 1/4 | 3 1/2 |
| 225 | 4 | 3 1/2 | 2 3/4 | 1/4 | 1/4 | 3/4 | 11 | 9 1/2 | 5 1/2 | 4 1/2 | 3 3/4 | 3 1/2 | 13/32 | 3 | 1 | 2 3/4 | 15 1/2 | 13 | 5 1/2 | 5 3/4 | 1/2 | 1 3/4 | 1/2 | 1/4 | 3 1/2 |
| 254 | 4 1/2 | 4 1/2 | 3 1/8 | 1/4 | 1/4 | 1 | 12 1/2 | 10 3/4 | 6 1/4 | 5 | 4 1/8 | 4 1/4 | 17/32 | 3 3/8 | 1 1/8 | 3 1/8 | 17 3/4 | 15 1/8 | 6 1/4 | 6 5/8 | 5/8 | 2 | 5/8 | 1/4 | 4 |
| 284 | 5 | 4 1/2 | 3 1/2 | 1/4 | 1/4 | 1 | 14 | 12 1/2 | 7 | 5 1/2 | 4 3/4 | 4 3/4 | 17/32 | 3 3/4 | 1 1/4 | 3 1/2 | 19 3/4 | 16 7/8 | 7 | 7 1/2 | 5/8 | 2 | 5/8 | 1/4 | 4 1/2 |
| 324 | *6 | *5 1/2 | 4 1/2 | 3/8 | 3/8 | 1 1/4 | 16 | 14 | 8 | 6 1/4 | 5 1/4 | 5 1/4 | 21/32 | 4 7/8 | 1 5/8 | 4 5/8 | 22 3/4 | 19 1/4 | 8 | 8 1/2 | 3/4 | 2 1/2 | 3/4 | *3/8 | 5 1/4 |
| 326 | *8 | 6 3/4 | 4 1/2 | 3/8 | 3/8 | 1 1/4 | 16 | 15 1/2 | 8 | 6 1/4 | 6 | 5 1/4 | 21/32 | 4 7/8 | 1 5/8 | 4 5/8 | 22 3/4 | 20 3/4 | 8 | 9 1/4 | 3/4 | 2 1/2 | 3/4 | *3/8 | 5 1/4 |
| 364 | 9 | 7 3/4 | 5 1/4 | 1/2 | 1/2 | 1 1/2 | 18 | 15 1/4 | 9 | 7 | 5 5/8 | 5 7/8 | 21/32 | 5 5/8 | 1 7/8 | 5 3/8 | 25 1/2 | 20 1/2 | 9 | 9 1/8 | 3/4 | 2 1/2 | 3/4 | 3/8 | 6 |
| 365 | 9 | 7 3/4 | 5 1/4 | 1/2 | 1/2 | 1 1/2 | 18 | 16 1/4 | 9 | 7 | 6 1/8 | 5 7/8 | 21/32 | 5 5/8 | 1 7/8 | 5 3/8 | 25 1/2 | 21 1/2 | 9 | 9 5/8 | 3/4 | 2 1/2 | 3/4 | 3/8 | 6 |
| 404 | 10 | 7 3/4 | 6 | 1/2 | 1/2 | 2 | 20 | 16 1/4 | 10 | 8 | 6 1/8 | 6 5/8 | 13/16 | 6 3/8 | 2 1/8 | 6 1/8 | 28 3/4 | 22 3/8 | 10 | 9 7/8 | 7/8 | 3 | 7/8 | 3/8 | 7 |
| 405 | 10 | 7 3/4 | 6 | 1/2 | 1/2 | 2 | 20 | 17 3/4 | 10 | 8 | 6 7/8 | 6 5/8 | 13/16 | 6 3/8 | 2 1/8 | 6 1/8 | 28 3/4 | 23 7/8 | 10 | 10 5/8 | 7/8 | 3 | 7/8 | 3/8 | 7 |
| 444 | 11 | 9 3/4 | 6 1/2 | 5/8 | 5/8 | 2 | 22 | 18 1/2 | 11 | 9 | 7 1/4 | 7 1/2 | 13/16 | 7 1/8 | 2 3/8 | 6 7/8 | 31 1/4 | 24 5/8 | 11 | 11 | 7/8 | 3 | 7/8 | 5/8 | 7 1/2 |
| 445 | 11 | 9 3/4 | 6 1/2 | 5/8 | 5/8 | 2 | 22 | 20 1/2 | 11 | 9 | 8 1/4 | 7 1/2 | 13/16 | 7 1/8 | 2 3/8 | 6 7/8 | 31 1/4 | 26 5/8 | 11 | 12 | 7/8 | 3 | 7/8 | 5/8 | 7 1/2 |
| 504 | 12 | 11 | 7 1/4 | 5/8 | 5/8 | 2 | 25 | 21 | 12 1/2 | 10 | 8 | 8 1/2 | 15/16 | 7 7/8 | 2 5/8 | 7 5/8 | 35 | 28 | 12 1/2 | 12 1/2 | 1 | 3 1/2 | 1 | 5/8 | 8 |
| 505 | 14 | 13 | 8 | 3/4 | 3/4 | 2 1/2 | 25 | 23 | 12 1/2 | 10 | 9 | 8 1/2 | 15/16 | 8 5/8 | 2 7/8 | 8 3/8 | 35 | 30 | 12 1/2 | 13 1/2 | 1 | 3 1/2 | 1 | 5/8 | 8 |
| 364S | Short Shaft for Direct Drive | | | 3/8 | 3/8 | 1 1/2 | 18 | 15 1/4 | 9 | 7 | 5 5/8 | 5 7/8 | 21/32 | 3 1/4 | 1 5/8 | 3 | | | | | | | | | |
| 365S | | | | 3/8 | 3/8 | 1 1/2 | 18 | 16 1/4 | 9 | 7 | 6 1/8 | 5 7/8 | 21/32 | 3 1/4 | 1 5/8 | 3 | | | | | | | | | |
| 404S | | | | 1/2 | 1/2 | 2 | 20 | 16 1/4 | 10 | 8 | 6 1/8 | 6 5/8 | 13/16 | 3 3/4 | 1 7/8 | 3 1/2 | | | | | | | | | |
| 405S | | | | 1/2 | 1/2 | 2 | 20 | 17 3/4 | 10 | 8 | 6 7/8 | 6 5/8 | 13/16 | 3 3/4 | 1 7/8 | 3 1/2 | | | | | | | | | |
| 444S | | | | 1/2 | 1/2 | 2 | 22 | 18 1/2 | 11 | 9 | 7 1/4 | 7 1/2 | 13/16 | 4 1/4 | 2 1/8 | 4 | | | | | | | | | |
| 445S | | | | 1/2 | 1/2 | 2 | 22 | 20 1/2 | 11 | 9 | 8 1/4 | 7 1/2 | 13/16 | 4 1/4 | 2 1/8 | 4 | | | | | | | | | |
| 504S | | | | 1/2 | 1/2 | 2 1/2 | 25 | 21 | 12 1/2 | 10 | 8 | 8 1/2 | 15/16 | 4 1/4 | 2 1/8 | 4 | | | | | | | | | |
| 505S | | | | 1/2 | 1/2 | 2 1/2 | 25 | 23 | 12 1/2 | 10 | 9 | 8 1/2 | 15/16 | 4 1/4 | 2 1/8 | 4 | | | | | | | | | |

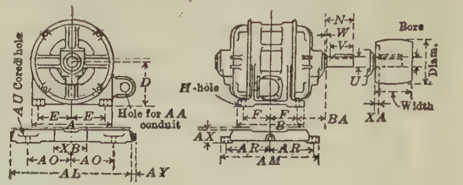


Fig. 7.

* The standard pulley for 3000 and 3600 r.p.m. motors in frames 324 and 326 is 5 in. diameter, 6 3/4 in. width overall, and the *XA* dimension is 1/4 in.

Note 1.—Dimension *D* will never be exceeded. When exact dimension is required, shims up to 1/32 in. may be necessary on frame sizes whose dimension *D* is 8 in. and less, and on larger frames, shims up to 1/16 in. may be necessary.

Note 2.—Tolerances on shaft extension diameters are: Shaft diam., 3/4 in. to 1 1/2 in. inclusive, + 0.0000 in., − 0.0005 in.; shaft diameter 1 5/8 in. to 3 3/8 in. inclusive, + 0.000 in., − 0.001 in.

† Dimension *XB* represents total movement of motor on base for tightening belt.

Note 3.—Tolerances on keyways: Width 3/16 in. to 3/4 in. inclusive, + 0.000 in., − 0.002 in.

Note 4.—Tolerances in making keyways parallel to center line of shaft: Shaft diam. 3/4 in. to 1 1/2 in. inclusive, 0.002 in.; shaft diameters 1 5/8 in. to 3 3/8 in. inclusive, 0.003 in. Limited to 12 in. in length.

**Table 15.—Dimensions and Tolerances of N.E.M.A. Standardized Frames for Vertical A.C. and D.C. Motors**

Solid Shaft.* Ring Construction. All dimensions in inches. See Fig. 8.

| Frame No. | *AK* † | *AJ* | *A* Max. | *AH* | *H* | *U* | Key | |
|---|---|---|---|---|---|---|---|---|
| | | | | | | | Width | Thickness |
| 204V | 11 | 12 1/2 | 14 | 2 3/4 | 17/32 | 3/4 | 3/16 | 3/16 |
| 224V | 11 | 12 1/2 | 14 | 2 3/4 | 17/32 | 1 | 1/4 | 1/4 |
| 225V | 11 | 12 1/2 | 14 | 2 3/4 | 17/32 | 1 | 1/4 | 1/4 |
| 254V | 11 | 12 1/2 | 14 | 2 3/4 | 17/32 | 1 1/8 | 1/4 | 1/4 |
| 284V | 14 | 16 | 18 | 4 1/2 | 13/16 | 1 1/4 | 1/4 | 1/4 |
| 324V | 14 | 16 | 18 | 4 1/2 | 13/16 | 1 5/8 | 3/8 | 3/8 |
| 326V | 14 | 16 | 18 | 4 1/2 | 13/16 | 1 5/8 | 3/8 | 3/8 |
| 364V | 14 | 16 | 18 | 4 1/2 | 13/16 | 1 7/8 | 1/2 | 1/2 |
| 365V | 14 | 16 | 18 | 4 1/2 | 13/16 | 1 7/8 | 1/2 | 1/2 |
| 404V | 14 | 16 | 18 | 4 1/2 | 13/16 | 2 1/8 | 1/2 | 1/2 |
| 405V | 14 | 16 | 18 | 4 1/2 | 13/16 | 2 1/8 | 1/2 | 1/2 |
| 444V | 14 | 16 | 18 | 4 1/2 | 13/16 | 2 1/8 | 1/2 | 1/2 |
| 445V | 14 | 16 | 18 | 4 1/2 | 13/16 | 2 1/8 | 1/2 | 1/2 |
| 504V | 18 | 22 | 25 | 5 | 15/16 | 2 1/8 | 1/2 | 1/2 |
| 505V | 18 | 22 | 25 | 5 | 15/16 | 2 1/8 | 1/2 | 1/2 |

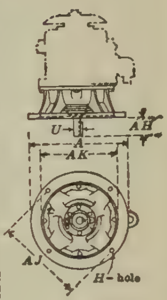


Fig. 8

Note 1.—Available length of shaft extension for coupling hub must be at least two times the diameter.

* Dimensions given also apply to hollow shaft motors, with the exception of *AH*, *U*, key dimensions, and Note 1.

† The tolerance for the *AK* dimension shall be: +0.005 in., −0.000 in.

**COSTS.**—Approximate costs of open, horizontal, 60- and 50-cycle, 110- and 220-volt, sleeve-bearing motors of the repulsion-induction or repulsion-start induction type are given in Fig. 9. Capacitor motors have a somewhat higher cost (increase ranges from 5 to 75%). Brush shifting repulsion motors (1800/720 r.p.m.) have a cost approximately twice that of corresponding ratings of 1800 r.p.m. repulsion-induction motors. Special mechanical features increase the cost.

## 4. POLYPHASE INDUCTION LARGE POWER MOTORS

Standard horsepower and speed ratings of large power, polyphase induction motors are given in Table 16. Motors rated 200 Hp. or less, having a continuous time rating, and for use without restriction to a particular application are designated general-purpose motors.

**TYPES** of polyphase induction motors in general use are as follows:

*a.* Squirrel-cage motors: 1. Class A, normal torque, normal starting current; 2. Class B, normal torque, low starting current; 3. Class C, high torque, low starting current; 4. Class D, high slip. *b.* Wound-rotor motors. *c.* Multi-speed motors. *d.* Quiet operating motors. *e.* Crane and hoist motors. *f.* Elevator motors. *g.* Adjustable-speed brush-shifting motors. *h.* Adjustable varying-speed brush-shifting motors.

**Squirrel-cage Motors.**—The various types of squirrel-cage motors differ principally in their starting characteristics. Class A motors develop normal starting torque (Table 17) and draw a starting current of 6 or 7 times full-load current. They have the highest efficiencies and power-factors, and the lowest cost of all types, and meet the requirements of the majority of applications. Although these motors are suitable for starting on full voltage, reduced voltage starting sometimes is required for motors rated 7 1/2 Hp. and larger to meet local starting current regulations.

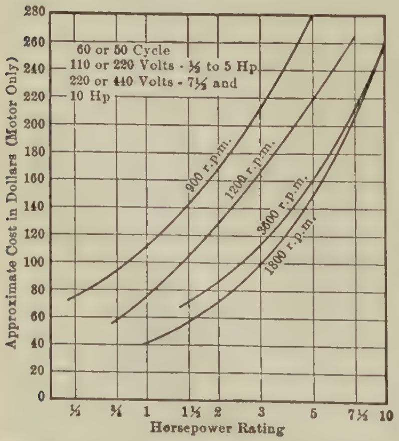


Fig. 9. Approximate 1936 Cost of Open Sleeve-bearing Repulsion Induction or Repulsion-start Induction Motors

**Table 16.—Standard Horsepower and Speed Ratings of Large Power, Polyphase Induction Motors**

| Hp. | Synchronous, Rev. per Min. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 60 Cycles | | | | | | | | 25 Cycles | | |
| 1/4 | .... | .... | .... | ... | ... | 600 | ... | ... | .... | ... | ... |
| 1/3 | .... | .... | .... | ... | 720 | 600 | ... | ... | .... | ... | ... |
| 1/2 | .... | .... | .... | 900 | 720 | 600 | ... | ... | .... | 750 | 500 |
| 3/4 | .... | .... | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 1 | .... | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 1 1/2 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 2 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 3 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 5 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 7 1/2 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 10 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 15 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 20 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 25 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 30 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 40 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 50 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 60 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 75 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 100 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 125 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 150 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 200 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 250 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 300 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 350 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 400 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 500 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 600 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | 1500 | 750 | 500 |
| 700 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | .... | ... | ... |
| 800 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | .... | ... | ... |
| 900 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | .... | ... | ... |
| 1000 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | .... | ... | ... |
| 1250 | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | ... | .... | ... | ... |
| 1500 | 3600 | 1800 | 1200 | 900 | 720 | 600 | ... | ... | .... | ... | ... |
| 1750 | 3600 | 1800 | 1200 | 900 | 720 | ... | ... | ... | .... | ... | ... |
| 2000 | 3600 | 1800 | 1200 | 900 | ... | ... | ... | ... | .... | ... | ... |

NOTE 1.—The 1500 r.p.m., 25-cycle ratings apply to squirrel-cage motors only.

NOTE 2.—The ratings above the heavy lines are suitable for belt drive, while those below are for direct connection only.

NOTE 3.—Standard 50-cycle horsepower ratings are the same as for 60 cycles, but the speeds are 5/6 those of the 60-cycle speeds.

**Table 17.—Torques and Starting Currents of 60-cycle, Class A, Squirrel-cage Induction Motors**

| Rev. per min. | 3600 | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 |
|---|---|---|---|---|---|---|---|---|
| | Percent of Full-load Values | | | | | | | |
| Min. starting torque | 150 | 150 | 135 | 125 | 120 | 115 | 110 | 105 |
| Min. breakdown torque | 200 | 200 | 200 | 200 | 200 | 200 | 200 | 200 |
| Typical starting current | 780 | 710 | 660 | 570 | 520 | 500 | 500 | 500 |

Class B motors are built with a high-reactance rotor winding, which limits the starting current to about 5 times full-load current. This allows the use of full voltage starting in some cases where reduced voltage starting would be necessary if a Class A motor were used. The starting torques are the same as for the Class A motors, but the power-factors and breakdown torques are slightly lower.

Class C motors, usually built with a "double squirrel-cage" rotor winding, combine high starting torque (200% or more) with the low starting current of Class B motors. Their power-factor, efficiency, and breakdown torques are all somewhat lower than for

Class A motors. They are used on such applications, as reciprocating compressors which start under load, where high starting torque and good running characteristics are required.

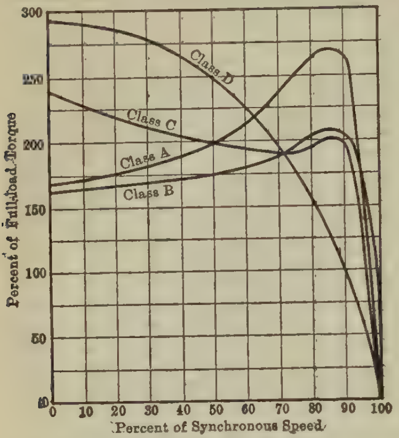


Fig. 10. Typical Speed-torque Curves of Squirrel-cage Induction Motors

Class D motors, built with a high-resistance rotor winding, develop a high starting torque with low starting current, but have high slip at running speed and low efficiency. They are used principally for drives imposing rapidly recurring peak loads, and for drives involving a heavy starting duty, but light or intermittent running duty. In the former case, a fly-wheel frequently is provided to equalize the load peaks. This type of motor should not be used to drive a continuous steady load, as its efficiency is quite low.

Fig. 10 shows typical speed-torque curves, and Fig. 11 shows typical speed-current curves of each of the four types of squirrel-cage induction motors.

**The Wound-rotor Induction Motor** has an insulated 3-phase winding on the rotor, whose terminals are connected to three slip rings. Stationary brushes bear on the latter, through which the rotor circuit is completed external to the motor. By inserting various amounts of resistance in series with the rotor winding the torque and current characteristics of the motor can be varied as shown in Fig. 12. High starting torque thus can be obtained with very low starting current. With all resistance cut out, the characteristics of this motor are similar to those of the Class A squirrel-cage motors, but with somewhat lower efficiency. With suitable control, wound-rotor motors can operate continuously at reduced speeds down to 50% speed, but the efficiency is reduced in proportion to the speed. At reduced speeds the motor has varying speed characteristics, *i.e.*, a change in load results in considerable change in speed. Wound-rotor motors are used occasionally where the application requires a high ratio of starting torque to starting current (high torque efficiency), or where severe accelerating duty is involved. However, these motors are used principally on applications requiring an adjustable speed drive. They can be used economically for driving variable torque loads (centrifugal pumps, fans, etc.) at reduced speeds, but are not economical for continuous reduced speed operation of constant torque loads.

**Multi-speed Squirrel-cage Motors** may be provided with: 1. A single stator winding which is connected to give either one of two speeds; 2. Two or more stator windings giving a corresponding number of speeds; 3. A combination of [1] and [2]. The single-winding motor is lowest in cost, and has the best characteristics, but it gives only two speeds having a ratio of 2 : 1. Two or more windings must be employed where more than two speeds, or other speed ratios, are required.

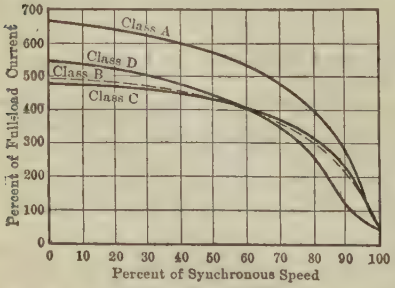


Fig. 11. Typical Speed-current Curves of Squirrel-cage Induction Motors

Multi-speed motors may be classified into three general types: *a. Variable torque motors* have horsepower ratings at each speed proportional to the square of the speed (*viz.*, 20/5 Hp., 1800/900 r.p.m.), and are used for driving centrifugal pumps, fans, and agitators; *b. Constant torque motors* have horsepower ratings at each speed directly proportional to the speed (*viz.*, 30/20/15/10 Hp., 1800/1200/900/600 r.p.m.), and are used for such applications as conveyors, stokers, reciprocating compressors, printing presses, and dough mixers; *c. Constant horsepower motors* have the same horsepower rating at all speeds (*viz.*, 5 Hp., 1800/600 r.p.m.). Typical applications are lathes, boring mills, planers, radial drills, grinders, milling machines, winches, and drag line scrapers. Multi-speed squirrel-cage motors may have electrical characteristics corresponding to Classes A, B, C, or D, but in general the same characteristics are obtained at all speeds.

**Multi-speed Wound-rotor Motors** also are available. Where speed adjustment (by varying the secondary resistance) is required on all speeds, only two speeds can be obtained and these must have a ratio of 2 : 1. In many applications, speed adjustment is required on only one speed, in which case two-, three-, or four-speed motors, having any synchronous speeds required, can be obtained. Except on the speed which is to be adjusted, the rotor winding operates short-circuited as a squirrel-cage motor. Typical applications of multi-speed wound-rotor motors include hoists for dragging and lifting service, conveyors, elevators, oil well pumps, and rotary car dumpers.

**Quiet-operating Motors** may be of the squirrel-cage (Class A, B, C, or D) or the wound-rotor type. There is no apparent difference in the construction of these motors and the standard motors, but the details of design, such as slot combination, uniformity of air gap, and magnetic flux density are given special consideration. Though more costly than standard motors, quiet-operating motors often are used in such locations as office buildings, schools, and hospitals, where noise must be kept to a minimum. The most common application of the quiet-operating motor is the driving of ventilating fans.

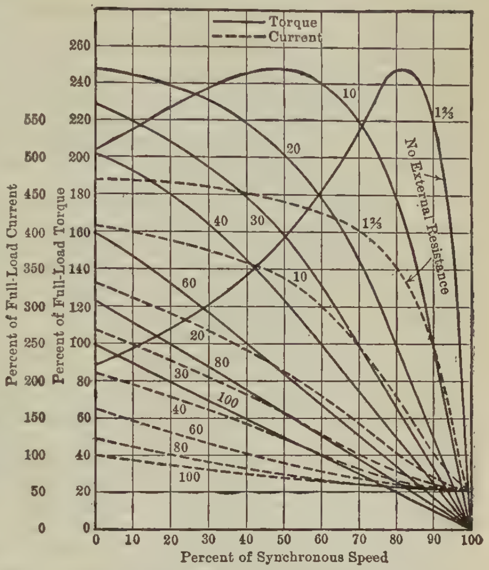


FIG. 12. Typical Speed-torque and Speed-current Curves of Wound-rotor Induction Motors

**Table 18.—N.E.M.A. Standardized Frame Sizes for Two- and Three-phase, General-purpose, Squirrel-cage Induction Motors**

Open type,* 60 cycles

| Hp. | Speed, rev. per min. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 3600 | | 1800 | 1200 | 900 | 720 | 600 | 514 | 450 | |
| | Frame Number † | | | | | | | | | |
| 1/2 | 110, 220, 440, and 550 volts | ... | ... | ... | 204 | 224 | 225 | ... | ... | 110, 220, 440, and 550 volts |
| 3/4 | | ... | ... | 204 | 224 | 225 | 254 | 284 | 284 | |
| 1 | | ... | 204 | 204 | 225 | 254 | 254 | 284 | 324 | |
| 1 1/2 | | 204 | 224 | 224 | 254 | 254 | 284 | 324 | 326 | |
| 2 | | 224 | 225 | 225 | 254 | 284 | 324 | 326 | 365 | 220, 440, and 550 volts |
| 3 | | 224 | 225 | 254 | 284 | 324 | 326 | 365 | 404 | |
| 5 | | 225 | 254 | 284 | 324 | 326 | 365 | 404 | 405 | |
| 7 1/2 | | 254 | 284 | 324 | 326 | 365 | 404 | 405 | 444 | |
| 10 | | 284 | 324 | 326 | 365 | 404 | 405 | 444 | 445 | |
| 15 | | 324 | 326 | 365 | 404 | 405 | 444 | 445 | 504 | |
| 20 | | 326 | 364 | 404 | 405 | 444 | 445 | 504 | 505 | |
| 25 | 220, 440, and 550 volts | 364S | 365 | 405 | 444‡ | 445 | 504 | 505 | ... | |
| 30 | | 365S | 405 | 444‡ | 445‡ | 504 | 505 | ... | ... | |
| 40 | | 404S | 444‡ | 445‡ | 504‡ | 505 | ... | ... | ... | |
| 50 | | 405S | 445S‡ | 504‡ | 505‡ | ... | ... | ... | ... | |
| 60 | | 444S | 504S‡ | 505‡ | ... | ... | ... | ... | ... | |
| 75 | | 445S | 505S‡ | ... | ... | ... | ... | ... | ... | |
| 100 | | 504S | ... | ... | ... | ... | ... | ... | ... | |
| 125 | | 505S | ... | ... | ... | ... | ... | ... | ... | |

* The frame numbers given for 3600 r.p.m. motor of 2 to 7 1/2 Hp. inclusive, and lower speed motors in frames 365 and smaller, also apply to totally-enclosed, fan-cooled motors.

† Standard mounting dimensions for these frame numbers are given in Figs. 7 and 8, and Tables 14 and 15.

‡ These frame numbers will apply also for 2200-volt motors.

**Crane and Hoist Motors** are available in both the squirrel-cage and wound-rotor types. The squirrel-cage motors are similar to Class D motors, having a high starting torque, low starting current and high full-load slip. The open motors are rated 30 min., 50° C. temperature rise, and the totally-enclosed motors are rated 15 min., 55° C. temperature rise. They are used mainly on the smaller hoists. The open type wound-rotor motors are rated 30 min., 50° C. temperature rise in sizes up to 40 Hp., and are rated 1 hour, 50° C. temperature rise in the larger sizes. The totally-enclosed, wound-rotor motors are rated 30 min., 55° C. temperature rise in all sizes. All crane and hoist motors can be obtained with a shaft extension on both ends, one for the driving pinion or coupling, the other for a brake. In addition to crane and hoist duty, these motors often are used for such applications as bascule lift and swing bridges, rotary car dumpers, larry cars, coke pushers, and hydraulic gates.

**Single-speed Elevator Motors** may be of the squirrel-cage or wound-rotor type. The former are similar to the Class D motors, having high starting torque and low starting current. They are available in ratings from 1 to 60 Hp. at speeds of 720 to 1800 r.p.m., and are intermittently rated, 30 min., 50° C. temperature rise. They are used on low-speed freight and passenger elevators, operating at a maximum car speed of about 150 ft. per minute. The squirrel-cage motor is most popular, and is used also on other applications, such as skip hoists. The wound-rotor type is used occasionally, since it is suitable for a greater number of starts and stops per hour.

**Two-speed Elevator Motors** also are available. They are either squirrel-cage or combination squirrel-cage and wound-rotor machines. The former may employ a single winding, giving a 2 : 1 speed ratio, or separate windings in the same slots giving a speed ratio up to 6 : 1. A high-resistance rotor winding, giving Class D characteristics, is employed. These motors are available in ratings from 10 to 60 Hp., with a top speed of 1200 or 900 r.p.m., and are rated 1 hour, 50° C. temperature rise on the high speed. The combination squirrel-cage and wound rotor motor really consists of two motors, a high-speed (900 r.p.m.) wound-rotor motor and a low-speed (225 or 150 r.p.m.) squirrel-cage motor, built as a single unit. These motors are available in ratings of 20 to 75 Hp., and are rated 2 hours, 50° C. temperature rise on the high speed. Two-speed motors are used on elevators having car speeds up to 550 ft. per minute. Higher speed elevators usually require D.C. driving motors. The two-speed squirrel-cage motor is most popular, but the combination motor is suitable for more severe service (greater number of starts and stops per hour).

**Adjustable-speed Brush-shifting Motors** are similar in construction to wound-rotor motors, except that the primary winding is on the rotor, the secondary winding is on the stator, and an additional adjusting winding, connected to a commutator, is provided on the rotor. Power is supplied to the primary winding through collector rings, and the secondary (stator) winding is connected to brushes which bear on the commutator. A means is provided for shifting the brushes around the commutator whereby the speed

**Table 19.—Frame Sizes for Two- and Three-phase, General-purpose Wound-rotor Induction Motors**

Open type, 60 cycles

| Hp. | Speed, rev. per min. | | | | | | |
|---|---|---|---|---|---|---|---|
| | 1800 | | 1200 | 900 | 720 | 600 | |
| | Frame Number * | | | | | | |
| 1 | 110, 220, 440, and 550 volts | . . . | . . . | 225 | . . . | . . . | 110, 220, 440, and 550 volts |
| 1 1/2 | | . . . | . . . | 254 | . . . | . . . | |
| 2 | | 225 | 225 | 254 | 324 | . . . | |
| 3 | | 225 | 254 | 284 | 326 | 365 | 220, 440, and 550 volts |
| 5 | | 254 | 284 | 324 | 365 | 404 | |
| 7 1/2 | | 284 | 324 | 326 | 404 | 405 | |
| 10 | | 324 | 326 | 365 | 405 | 444 | |
| 15 | | 326 | 365 | 404 | 444 | 445 | |
| 20 | 220, 440, and 550 volts | 364 | 404 | 405 | 445 | 504 | |
| 25 | | 365 | 405 | 444† | 504 | 505 | |
| 30 | | 405 | 444† | 445† | 505 | . . . | |
| 40 | | 444† | 445† | 504† | . . . | . . . | |
| 50 | | 445S† | 504† | 505† | . . . | . . . | |
| 60 | | 504S† | 505† | . . . | . . . | . . . | |
| 75 | | 505S† | . . . | . . . | . . . | . . . | |

* See Figs. 7 and 8 and Tables 14 and 15 for standard mounting dimensions for these frame numbers.

† These frame numbers will also apply for 2200-volt motors.

adjustment is obtained. These motors can be obtained with a 3 : 1 or 4 : 1 speed range, the top speed being 135 to 150% of synchronous speed. The motors are limited in rating to about 8 Hp. per pole (60-cycle motors) at the top speed for 3 : 1 speed range, and can be obtained only for low voltage (550 volts and below) power supply. They are constant torque machines, *i.e.*, the horsepower output varies directly with the speed. For any given brush setting, the speed is essentially independent of the load. Though considerably higher in cost than wound-rotor or multi-speed squirrel-cage motors, they have the advantages of a wide speed range with an infinite number of speed points, good efficiency at all speeds, and good speed regulation. Typical applications are textile machinery (cloth calenders and finishing machines, sanforizing machines and cloth-printing machines), stokers, printing presses, small paper machines, and conveyors.

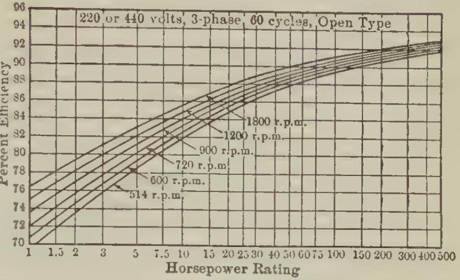


FIG. 13. Approximate Full-load Efficiencies of Class A Squirrel-cage Induction Motors

**Adjustable Varying-speed Brush-shifting Motors** are similar to the adjustable speed machines, except that no adjusting winding is provided; instead, the single rotor winding is connected to a commutator. Brushes bearing on the commutator are supplied with power from the secondary of a transformer whose primary is connected in series with the stator winding. Speed adjustment is obtained by varying the position of the brushes. The motor has varying speed characteristics, the no-load speed being approximately 50% above the full load speed. Capacities up to 15 Hp. per pole at the top speed can be obtained, with a speed range up to 3 : 1. Top speed is about 10% above synchronous speed. Either low- or high-voltage motors can be obtained. In some cases the stator winding is arranged for re-connection from delta to star, when going from the upper part of the speed range to the lower part. Higher efficiency and power-factor thereby are obtained in the lower speed range. These motors have been used largely for driving large fans operating over a wide speed range.

## Operating Characteristics

**TEMPERATURE RISE.**—The temperature rise of standard squirrel-cage (Class A, B, C, or D), wound-rotor, multi-speed, and quiet-operating motors, when operating under rated conditions, is 40° C. rise, measured by thermometer for open motors; 50° C. for splash-proof motors; 55° C. for totally-enclosed motors. Adjustable-speed brush-shifting motors usually are rated 40° C., when operating from maximum full-load speed down to 50% speed. They are rated 50° C. when operating at lower speeds. Open motors rated 40° C. usually are given a *service factor* of 1.15; *i.e.*, manufacturers guarantee successful operation at 1.15 times rated load without injurious heating, although the efficiency and power-factor may be slightly different from the rated load values.

**EFFICIENCY.**—Typical full-load efficiencies and power-factors of standard Class A squirrel-cage induction motors are given in Figs. 13 and 14 respectively. The efficiencies of Class B and Class C squirrel-cage motors and wound-rotor motors are somewhat lower, and those of Class D squirrel-cage motors considerably lower, than those of Class A squirrel-cage motors. The power-factors of Class B and C squirrel-cage motors also are somewhat lower than those of Class A motors. Adjustable-speed brush-shifting motors have efficiencies at the maximum speed slightly lower than

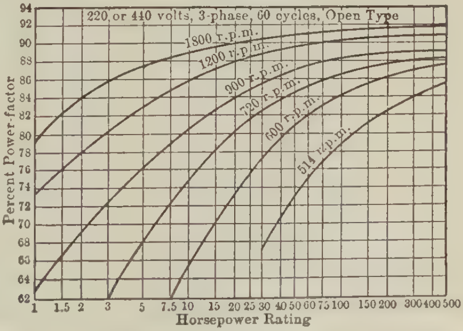


FIG. 14. Approximate Full-load Power Factors of Class A Squirrel-cage Induction Motors

those of wound-rotor motors, but at reduced speeds their efficiencies are considerably higher than those of wound-rotor motors operating at reduced speeds.

**TORQUE AND STARTING CURRENT.**—Minimum starting and breakdown torques and typical starting current of Class A, squirrel-cage motors are given in Table 17. Relative values for the other classes of squirrel-cage motors are indicated by the curves of Figs. 10 and 11.

As shown in Fig. 12, the starting torque and starting current of wound-rotor motors vary over a wide range, depending on the amount of external resistance in the rotor circuit. It is usual to start these motors with sufficient external resistance to give 100% torque and current at standstill, the resistance being short-circuited in steps as the motor comes up to speed. The minimum breakdown torque for wound-rotor motors is 200% of full-load torque.

Adjustable-speed brush-shifting motors usually are started with the brushes in the low-speed position. The starting torque then is 140 to 200% of full-load torque, and the starting current is 125 to 175% of top speed full-load current. Their breakdown torques vary between 140 and 200%, with the brushes in the low-speed position, and are not less than 250% with the brushes in the high-speed position.

**VOLTAGE VARIATION.**—All polyphase induction motors should operate successfully at rated load and frequency at any voltage not more than 10% above or below normal, but not necessarily in accordance with the standards of performance established for operation at normal rating. The starting currents are approximately proportional to the applied voltage, and the starting and breakdown torques are approximately proportional to the square of the voltage.

**Table 20.—Frame Sizes for Two- and Three-phase, Squirrel-cage Induction Motors**

Totally-enclosed, 60 cycles, 550 volts or less

| Horsepower.... | 1/2 | 3/4 | 1 | 1 1/2 | 2 | 3 | 5 | 7 1/2 | 10 | 15 |
|---|---|---|---|---|---|---|---|---|---|---|
| | Frame Number * | | | | | | | | | |
| 1800 r.p.m..... | ... | ... | 204 | 224 | 254 | 284 | 326 | 365 | 405 | 504 |
| 1200 r.p.m..... | ... | ... | 204 | 225 | 254 | 324 | 365 | 404 | 444 | 504 |
| 900 r.p.m..... | 204 | 224 | 225 | 254 | 284 | 324 | 365 | 404 | 444 | 504 |

* See Figs. 7 and 8 and Tables 14 and 15 for standard mounting dimensions.

**Table 21.—Frame Sizes for Three-phase, Two-speed, Single-winding, Squirrel-cage Induction Motors**

Open type, 60 cycles, 550 volts or less

| Variable Torque Motors | | | | | | Constant Torque Motors | | | | | | Constant Hp. Motors | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Hp. at | | Rev. per min. | | | | Hp. at | | Rev. per min. | | | | Hp. | Rev. per min. | | | |
| High Speed | Low Speed | 3600 and 1800 | 1800 and 900 | 1200 and 600 | 900 and 450 | High Speed | Low Speed | 3600 and 1800 | 1800 and 900 | 1200 and 600 | 900 and 450 | | 3600 and 1800 | 1800 and 900 | 1200 and 600 | 900 and 450 |
| Frame Numbers. (See Tables 14 and 15 for mounting dimensions.) | | | | | | | | | | | | | | | | |
| ...... | ...... | ..... | ..... | ..... | ..... | 1/6 | 1/12 | ..... | ..... | ..... | 204 | 1/6 | ..... | ..... | 204 | 225 |
| ...... | ...... | ..... | ..... | ..... | ..... | 1/4 | 1/8 | ..... | ..... | ..... | 224 | 1/4 | ..... | ..... | 224 | 225 |
| ...... | ...... | ..... | ..... | ..... | ..... | 1/3 | 1/6 | ..... | ..... | 204 | 224 | 1/3 | ..... | ..... | 224 | 254 |
| 1/2 | 0.12 | ..... | ..... | ..... | 204 | 1/2 | 1/4 | ..... | ..... | 224 | 225 | 1/2 | ..... | 204 | 225 | 254 |
| 3/4 | .19 | ..... | ..... | 204 | 224 | 3/4 | 3/8 | ..... | ..... | 224 | 254 | 3/4 | ..... | 224 | 254 | 284 |
| 1 | .25 | 204 | 204 | 204 | 225 | 1 | 1/2 | 204 | 204 | 225 | 254 | 1 | 204 | 225 | 254 | 324 |
| 1 1/2 | .37 | 204 | 224 | 224 | 254 | 1 1/2 | 3/4 | 204 | 224 | 254 | 284 | 1 1/2 | 224 | 254 | 284 | 326 |
| 2 | .50 | 224 | 225 | 225 | 254 | 2 | 1 | 224 | 225 | 254 | 324 | 2 | 225 | 254 | 324 | 365 |
| 3 | .75 | 224 | 225 | 254 | 284 | 3 | 1 1/2 | 224 | 254 | 284 | 326 | 3 | 225 | 284 | 326 | 404 |
| 5 | 1.2 | 225 | 254 | 284 | 324 | 5 | 2 1/2 | 225 | 284 | 326 | 365 | 5 | 254 | 324 | 365 | 405 |
| 7 1/2 | 1.9 | 254 | 284 | 324 | 326 | 7 1/2 | 3 3/4 | 254 | 284 | 364 | 404 | 7 1/2 | 284 | 326 | 404 | 444 |
| 10 | 2.5 | 284 | 324 | 326 | 365 | 10 | 5 | 284 | 324 | 365 | 405 | 10 | 324 | 365 | 405 | 445 |
| 15 | 3.7 | 324 | 326 | 365 | 404 | 15 | 7 1/2 | 324 | 364 | 404 | 444 | 15 | 326 | 404 | 444 | 504 |
| 20 | 5.0 | 326 | 364 | 404 | 405 | 20 | 10 | 326 | 365 | 405 | 445 | 20 | 364 | 405 | 445 | 505 |
| 25 | 6.2 | 364 | 365 | 405 | 444 | 25 | 12 1/2 | 364 | 404 | 444 | 504 | 25 | 365 | 444 | 504 | ..... |
| 30 | 7.5 | 365 | 405 | 444 | 445 | 30 | 15 | 365 | 405 | 445 | 504 | 30 | 405 | 445 | 505 | ..... |
| 40 | 10 | 404 | 444 | 445 | 504 | 40 | 20 | 404 | 444 | 504 | 505 | 40 | 444 | 504 | ..... | ..... |
| 50 | 12 | 405 | 445 | 504 | 505 | 50 | 25 | 405 | 445 | 505 | ..... | 50 | 445 | 505 | ..... | ..... |
| 60 | 15 | 444 | 504 | 505 | ..... | 60 | 30 | 444 | 504 | ..... | ..... | 60 | 504 | ..... | ..... | ..... |
| 75 | 19 | 445 | 505 | ..... | ..... | 75 | 37 | 445 | 505 | ..... | ..... | 75 | 505 | ..... | ..... | ..... |
| 100 | 25 | 504 | ..... | ..... | ..... | 100 | 50 | 504 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |
| 125 | 31 | 505 | ..... | ..... | ..... | 125 | 62 | 505 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... |

**Table 22.—Frame Sizes for Variable Torque Two- and Three-phase, Two-speed, Two-winding, Squirrel-cage Induction Motors**

Open type, 60 cycles, 550 volts or less

| 1800–1200 r.p.m. | | 1800–900 r.p.m. | | 1800–600 r.p.m. | | 1200–900 r.p.m. | |
|---|---|---|---|---|---|---|---|
| Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* |
| .......... | ... | .......... | ... | .......... | ... | 1/2– 0.3 | 204 |
| 3/4– 0.33 | 204 | 3/4– 0.19 | 204 | 3/4–0.08 | 204 | 3/4– 0.4 | 224 |
| 1 – 0.5 | 204 | 1 – 0.25 | 204 | 1 –0.11 | 204 | 1 – 0.6 | 225 |
| 1 1/2– 0.7 | 224 | 1 1/2– 0.37 | 224 | 1 1/2–0.17 | 224 | 1 1/2– 0.8 | 254 |
| 2 – 0.9 | 225 | 2 – 0.5 | 225 | 2 –0.22 | 225 | 2 – 1.1 | 254 |
| 3 – 1.3 | 254 | 3 – 0.75 | 254 | 3 –0.33 | 254 | 3 – 1.7 | 284 |
| 5 – 2.2 | 284 | 5 – 1.25 | 284 | 5 –0.56 | 284 | 5 – 2.7 | 324 |
| 7 1/2– 3.3 | 324 | 7 1/2– 1.9 | 324 | 7 1/2–0.83 | 324 | 7 1/2– 4.2 | 326 |
| 10 – 4.4 | 326 | 10 – 2.5 | 326 | 10 –1.1 | 326 | 10 – 5.6 | 365 |
| 15 – 6.7 | 364 | 15 – 3.7 | 364 | 15 –1.7 | 364 | 15 – 8.4 | 404 |
| 20 – 8.9 | 365 | 20 – 5 | 365 | 20 –2.2 | 365 | 20 –11.3 | 405 |
| 25 –11 | 405 | 25 – 6.2 | 405 | 25 –2.8 | 405 | 25 –14 | 444 |
| 30 –13 | 444 | 30 – 7.5 | 444 | 30 –3.3 | 444 | 30 –17 | 445 |
| 40 –18 | 445 | 40 –10 | 445 | 40 –4.4 | 445 | 40 –22 | 504 |
| 50 –22 | 504 | 50 –13 | 504 | 50 –5.6 | 504 | 50 –28 | 505 |
| 60 –27 | 505 | 60 –15 | 505 | 60 –6.7 | 505 | .......... | ... |

| 1200–720 r.p.m. | | 1200–600 r.p.m. | | 1200–400 r.p.m. | | 900–720 r.p.m. | |
|---|---|---|---|---|---|---|---|
| Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* |
| .......... | ... | .......... | ... | .......... | ... | 1/3– 0.22 | 224 |
| 1/2– 0.2 | 204 | 1/2– 0.12 | 204 | .......... | ... | 1/2– 0.32 | 224 |
| 3/4– 0.3 | 224 | 3/4– 0.19 | 224 | .......... | ... | 3/4– 0.48 | 225 |
| 1 – 0.4 | 225 | 1 – 0.25 | 225 | .......... | ... | 1 – 0.64 | 254 |
| 1 1/2– 0.5 | 254 | 1 1/2– 0.37 | 254 | 1 1/2–0.17 | 254 | 1 1/2– 0.96 | 254 |
| 2 – 0.7 | 254 | 2 – 0.5 | 254 | 2 –0.22 | 254 | 2 – 1.3 | 284 |
| 3 – 1.1 | 284 | 3 – 0.75 | 284 | 3 –0.33 | 284 | 3 – 1.9 | 324 |
| 5 – 1.8 | 324 | 5 – 1.2 | 324 | 5 –0.55 | 324 | 5 – 3.2 | 326 |
| 7 1/2– 2.7 | 326 | 7 1/2– 1.9 | 326 | 7 1/2–0.83 | 326 | 7 1/2– 4.8 | 365 |
| 10 – 3.6 | 365 | 10 – 2.5 | 365 | 10 –1.1 | 365 | 10 – 6.4 | 404 |
| 15 – 5.4 | 404 | 15 – 3.7 | 404 | 15 –1.7 | 404 | 15 – 9.6 | 405 |
| 20 – 7.2 | 405 | 20 – 5 | 405 | 20 –2.2 | 405 | 20 –13 | 444 |
| 25 – 9 | 444 | 25 – 6.2 | 444 | 25 –2.8 | 444 | 25 –16 | 445 |
| 30 –11 | 445 | 30 – 7.5 | 445 | 30 –3.3 | 445 | 30 –19 | 504 |
| 40 –15 | 504 | 40 –10 | 504 | 40 –4.4 | 504 | 40 –26 | 505 |
| 50 –18 | 505 | 50 –12 | 505 | 50 –5.6 | 505 | .......... | ... |

| 900–600 r.p.m. | | 900–450 r.p.m. | | 720–600 r.p.m. | | 720–400 r.p.m. | |
|---|---|---|---|---|---|---|---|
| Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* |
| .......... | ... | .......... | ... | 1/4– 0.17 | 224 | .......... | ... |
| 1/3– 0.15 | 224 | 1/3– 0.08 | 224 | 1/3– 0.23 | 224 | .......... | ... |
| 1/2– 0.22 | 224 | 1/2– 0.12 | 224 | 1/2– 0.35 | 225 | .......... | ... |
| 3/4– 0.33 | 225 | 3/4– 0.19 | 225 | 3/4– 0.52 | 254 | 3/4–0.23 | 254 |
| 1 – 0.5 | 254 | 1 – 0.25 | 254 | 1 – 0.7 | 254 | 1 –0.31 | 254 |
| 1 1/2– 0.7 | 254 | 1 1/2– 0.37 | 254 | 1 1/2– 1 | 284 | 1 1/2–0.46 | 284 |
| 2 – 0.9 | 284 | 2 – 0.5 | 284 | 2 – 1.4 | 324 | 2 –0.62 | 324 |
| 3 – 1.3 | 324 | 3 – 0.75 | 324 | 3 – 2.1 | 326 | 3 –0.93 | 326 |
| 5 – 2.2 | 326 | 5 – 1.2 | 326 | 5 – 3.5 | 365 | 5 –1.5 | 365 |
| 7 1/2– 3.3 | 365 | 7 1/2– 1.9 | 365 | 7 1/2– 5.2 | 404 | 7 1/2–2.3 | 404 |
| 10 – 4.4 | 404 | 10 – 2.5 | 404 | 10 – 7 | 405 | 10 –3.1 | 405 |
| 15 – 6.7 | 405 | 15 – 3.7 | 405 | 15 –10 | 444 | 15 –4.6 | 444 |
| 20 – 8.9 | 444 | 20 – 5 | 444 | 20 –14 | 445 | 20 –6.2 | 445 |
| 25 –11 | 445 | 25 – 6.2 | 445 | 25 –17 | 504 | 25 –7.7 | 504 |
| 30 –13 | 504 | 30 – 7.5 | 504 | 30 –21 | 505 | 30 –9.2 | 505 |
| 40 –18 | 505 | 40 –10 | 505 | .......... | ... | .......... | ... |

* See Tables 14 and 15 for standard mounting dimensions.

## Mechanical Features

Large power polyphase induction motors usually are furnished with feet on the stator frame, and end-shield type bearings, although pedestal type bearings, with or without base, may be used in sizes above 1 Hp. per r.p.m. The smaller motors also can be obtained with a machined face on the drive end end-shield, or provided with a flange for bolting to the driven machine. Vertical motors, as well as the standard horizontal motors, can be obtained. The former usually have a ring base, although a four-leg or tripod base can be obtained in the small sizes.

Sleeve bearings, oil-ring-lubricated, are standard for most horizontal machines. Ball bearings are standard on totally-enclosed, and splash-proof motors, and on all but the larger vertical motors. They can be obtained at increased cost on all motors. Ball

(*Continued on page 25–38*).

**Table 23.—Frame Sizes of Constant Torque and Constant Horsepower, Two- and Three-phase, Two-speed, Two-winding Squirrel-cage Induction Motors**

Open Type, 60 cycles, 550 volts or less

Constant Torque Motors

| 1800–1200 r.p.m. | | 1800–900 r.p.m. | | 1200–900 r.p.m. | | 1200–720 r.p.m. | |
|---|---|---|---|---|---|---|---|
| Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* |
| .......... | ... | .......... | ... | 1/3– 1/4 | 204 | 1/3– 1/5 | 224 |
| .......... | ... | .......... | ... | 1/2– 3/8 | 224 | 1/2– 3/10 | 224 |
| 3/4– 1/2 | 204 | 3/4– 3/8 | 204 | 3/4– 9/16 | 224 | 3/4– 1/2 | 225 |
| 1 – 2/3 | 224 | 1 – 1/2 | 224 | 1 – 3/4 | 225 | 1 – 6/10 | 254 |
| 1 1/2– 1 | 224 | 1 1/2– 3/4 | 224 | 1 1/2– 1 1/8 | 254 | 1 1/2– 9/10 | 254 |
| 2 – 1 1/3 | 225 | 2 – 1 | 225 | 2 – 1 1/2 | 254 | 2 – 1 1/5 | 284 |
| 3 – 2 | 254 | 3 – 1 1/2 | 254 | 3 – 2 1/4 | 284 | 3 – 1 4/5 | 324 |
| 5 – 3 1/3 | 284 | 5 – 2 1/2 | 284 | 5 – 3 3/4 | 324 | 5 – 3 | 326 |
| 7 1/2– 5 | 324 | 7 1/2– 3 3/4 | 324 | 7 1/2– 5 2/3 | 326 | 7 1/2– 4 1/2 | 365 |
| 10 – 6 2/3 | 326 | 10 – 5 | 326 | 10 – 7 1/2 | 365 | 10 – 6 | 404 |
| 15 –10 | 365 | 15 – 7 1/2 | 365 | 15 –11 | 404 | 15 – 9 | 405 |
| 20 –13 | 404 | 20 –10 | 404 | 20 –15 | 405 | 20 –12 | 444 |
| 25 –17 | 405 | 25 –12 | 405 | 25 –19 | 444 | 25 –15 | 444 |
| 30 –20 | 444 | 30 –15 | 444 | 30 –22 | 445 | 30 –18 | 445 |
| 40 –27 | 445 | 40 –20 | 445 | 40 –30 | 504 | 40 –24 | 504 |
| 50 –33 | 504 | 50 –25 | 504 | 50 –37 | 505 | 50 –30 | 505 |
| 60 –40 | 505 | 60 –30 | 505 | .......... | ... | .......... | ... |

Constant Torque Motors

| 1200–600 r.p.m. | | 1200–400 r.p.m. | | 900–600 r.p.m. | | 900–450 r.p.m. | |
|---|---|---|---|---|---|---|---|
| Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* |
| .......... | ... | .......... | ... | 1/6– 1/9 | 204 | 1/6– 1/12 | 224 |
| .......... | ... | .......... | ... | 1/4– 1/6 | 224 | 1/4– 1/8 | 225 |
| 1/3– 1/6 | 224 | .......... | ... | 1/3– 2/9 | 224 | 1/3– 1/6 | 225 |
| 1/2– 1/4 | 224 | .......... | ... | 1/2– 1/3 | 225 | 1/2– 1/4 | 254 |
| 3/4– 3/8 | 225 | .......... | ... | 3/4– 1/2 | 254 | 3/4– 3/8 | 284 |
| 1 – 1/2 | 254 | 1 – 1/3 | 254 | 1 – 2/3 | 284 | 1 – 1/2 | 284 |
| 1 1/2– 3/4 | 254 | 1 1/2– 1/2 | 284 | 1 1/2– 1 | 284 | 1 1/2– 3/4 | 324 |
| 2 – 1 | 284 | 2 – 2/3 | 324 | 2 – 1 1/3 | 324 | 2 – 1 | 326 |
| 3 – 1 1/2 | 324 | 3 – 1 | 326 | 3 – 2 | 326 | 3 – 1 1/2 | 365 |
| 5 – 2 1/2 | 326 | 5 – 1 2/3 | 365 | 5 – 3 1/3 | 365 | 5 – 2 1/2 | 404 |
| 7 1/2– 3 3/4 | 365 | 7 1/2– 2 1/2 | 404 | 7 1/2– 5 | 404 | 7 1/2– 3 3/4 | 405 |
| 10 – 5 | 404 | 10 – 3 1/3 | 405 | 10 – 6 2/3 | 405 | 10 – 5 | 444 |
| 15 – 7 1/2 | 405 | 15 – 5 | 444 | 15 –10 | 444 | 15 – 7 1/2 | 445 |
| 20 –10 | 444 | 20 – 6 2/3 | 445 | 20 –13 | 445 | 20 –10 | 504 |
| 25 –12 | 444 | 25 – 8 1/3 | 504 | 25 –17 | 504 | 25 –12 | 505 |
| 30 –15 | 445 | 30 –10 | 504 | 30 –20 | 505 | .......... | ... |
| 40 –20 | 504 | 40 –13 | 505 | .......... | ... | .......... | ... |
| 50 –25 | 505 | .......... | ... | .......... | ... | .......... | ... |

Constant Horsepower Motors

| Hp. | 1800–1200 r.p.m. | 1800–900, 1200–900 r.p.m. | 1800–600, 1200–600, 900–600 r.p.m. | 1800–450, 900–450, 600–450 r.p.m. |
|---|---|---|---|---|
| | Frame Number * | | | |
| 1/6 | ... | ... | 224 | 224 |
| 1/4 | ... | ... | 224 | 225 |
| 1/3 | ... | 204 | 225 | 254 |
| 1/2 | 204 | 224 | 254 | 284 |
| 3/4 | 224 | 225 | 284 | 324 |
| 1 | 225 | 254 | 284 | 326 |
| 1 1/2 | 254 | 254 | 324 | 365 |
| 2 | 254 | 284 | 326 | 404 |
| 3 | 284 | 324 | 365 | 405 |
| 5 | 324 | 326 | 404 | 444 |
| 7 1/2 | 326 | 365 | 405 | 445 |
| 10 | 365 | 404 | 444 | 504 |
| 15 | 404 | 405 | 445 | 505 |
| 20 | 405 | 444 | 504 | ... |
| 25 | 444 | 445 | 505 | ... |
| 30 | 445 | 504 | ... | ... |
| 40 | 504 | 505 | ... | ... |
| 50 | 505 | ... | ... | ... |

* See Tables 14 and 15 for standard mounting dimensions.

## Table 24.—Frame Sizes for Three-phase, Four-speed, Two-winding, Squirrel-cage Induction Motors

Open type, 60 cycles, 550 volts or less

| Variable Torque Motors | | | | Constant Torque Motors | | | | Constant Hp. Motors | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1800-1200-900-600 r.p.m. | | 1200-900-600-450 r.p.m. | | 1800-1200-900-600 r.p.m. | | 1200-900-600-450 r.p.m. | | 1800-1200-900-600 r.p.m. | | 1200-900-600-450 r.p.m. | |
| Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* | Hp. | Frame No.* |
| ........ | ... | ........ | ... | ........ | ... | ........ | ... | 1/4 | 225 | 1/4 | 254 |
| ........ | ... | ........ | ... | ........ | ... | ........ | ... | 1/3 | 254 | 1/3 | 254 |
| ........ | ... | ........ | ... | ........ | ... | ........ | ... | 1/2 | 254 | 1/2 | 284 |
| 3/4- 0.33- 0.19-0.08 | 224 | 3/4- 0.42- 0.19-0.11 | 224 | 3/4- 1/2- 3/8- 1/4 | 225 | 3/4- 1/2- 3/8- 1/4 | 254 | 3/4 | 284 | 3/4 | 324 |
| 1 - 0.44- 0.25-0.11 | 224 | 1 - 0.56- 0.25-0.14 | 225 | 1 - 2/3- 1/2- 1/3 | 254 | 1 - 3/4- 1/2- 3/8 | 254 | 1 | 284 | 1 | 326 |
| 1 1/2- 0.67- 0.37-0.17 | 225 | 1 1/2- 0.84- 0.37-0.21 | 254 | 1 1/2- 1 - 3/4- 1/2 | 254 | 1 1/2- 1 - 3/4- 1/2 | 284 | 1 1/2 | 324 | 1 1/2 | 365 |
| 2 - 0.89- 0.5 -0.22 | 254 | 2 - 1.1 - 0.50-0.27 | 254 | 2 - 1 1/3- 1 - 2/3 | 284 | 2 - 1 1/2- 1 - 3/4 | 324 | 2 | 326 | 2 | 404 |
| 3 - 1.3 - 0.75-0.33 | 254 | 3 - 1.7 - 0.75-0.42 | 284 | 3 - 2 - 1 1/2- 1 | 284 | 3 - 2 1/4- 1 1/2- 1 | 326 | 3 | 365 | 3 | 405 |
| 5 - 2.2 - 1.2 -0.55 | 284 | 5 - 2.8 - 1.2 -0.70 | 324 | 5 - 3 1/3- 2 1/2- 1 2/3 | 324 | 5 - 3 3/4- 2 1/2- 1 3/4 | 365 | 5 | 404 | 5 | 444 |
| 7 1/2- 3.3 - 1.9 -0.83 | 324 | 7 1/2- 4.2 - 1.9 -1.1 | 326 | 7 1/2- 5 - 3 3/4- 2 1/2 | 326 | 7 1/2- 5 1/2- 3 3/4- 2 3/4 | 404 | 7 1/2 | 405 | 7 1/2 | 445 |
| 10 - 4.4 - 2.5 -1.1 | 326 | 10 - 5.6 - 2.5 -1.4 | 365 | 10 - 6 2/3- 5 - 3 1/3 | 365 | 10 - 7 1/2- 5 - 3 | 405 | 10 | 444 | 10 | 504 |
| 15 - 6.7 - 3.7 -1.7 | 365 | 15 - 8.4 - 3.7 -2.1 | 404 | 15 -10 - 7 1/2- 5 | 404 | 15 -11 - 7 1/2- 5 1/2 | 444 | 15 | 445 | 15 | 505 |
| 20 - 8.9 - 5 -2.2 | 404 | 20 -11 - 5 -2.7 | 405 | 20 -13 -10 - 6 2/3 | 405 | 20 -15 -10 - 7 1/2 | 445 | 20 | 504 | .... | ... |
| 25 -11 - 6.2 -2.7 | 405 | 25 -14 - 6 -3.5 | 444 | 25 -17 -12 - 8 1/3 | 444 | 25 -19 -12 - 9 1/2 | 504 | 25 | 504 | .... | ... |
| 30 -13 - 7.6 -3.2 | 444 | 30 -17 - 7.5 -4.2 | 445 | 30 -20 -15 -10 | 445 | 30 -22 -15 -11 | 505 | ..... | ... | .... | ... |
| 40 -18 -10 -4.5 | 445 | 40 -22 -10 -5.5 | 504 | 40 -27 -20 -13 | 504 | ........ | ... | ..... | ... | .... | ... |
| 50 -22 -12 -5.5 | 504 | 50 -28 -12 -7 | 505 | 50 -33 -25 -17 | 505 | ........ | ... | ..... | ... | .... | ... |
| 60 -27 -15 -6.7 | 505 | ........ | ... | ........ | ... | ........ | ... | ..... | ... | .... | ... |

* See Tables 14 and 15 for standard mounting dimensions.

## Table 25.—Approximate Weights and Dimensions of General-Purpose Squirrel-cage, Induction Motors

Open type, 60 cycles. Dimensions in inches. See Fig. 15.

| Hp. | 1800 r.p.m. | | | | 1200 r.p.m. | | | | 900 r.p.m. | | | | 720 r.p.m. | | | | 600 r.p.m. | | | | 514 r.p.m. | | | | 450 r.p.m. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Wt.,lb. | *A* | *C* | *O* | Wt.,lb. | *A* | *C* | *O* | Wt.,lb. | *A* | *C* | *O* | Wt.,lb. | *A* | *C* | *O* | Wt.,lb. | *A* | *C* | *O* | Wt.,lb. | *A* | *C* | *O* | Wt.,lb. | *A* | *C* | *O* |
| 25 | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | 1550 | 28 | 46 | 29 |
| 30 | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | 1550 | 28 | 46 | 29 | 1800 | 28 | 50 | 29 |
| 40 | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | 1800 | 28 | 50 | 29 | 1800 | 28 | 50 | 29 | 2000 | 31 | 48 | 32 |
| 50 | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | 1800 | 28 | 50 | 29 | 2000 | 31 | 48 | 32 | 2000 | 31 | 48 | 32 | 2400 | 31 | 54 | 32 |
| 60 | .... | .. | .. | .. | .... | .. | .. | .. | 1800 | 28 | 50 | 29 | 2000 | 31 | 48 | 32 | 2400 | 31 | 54 | 32 | 2400 | 31 | 54 | 32 | 3000 | 31 | 57 | 32 |
| 75 | .... | .. | .. | .. | 1550 | 28 | 46 | 29 | 2000 | 31 | 48 | 32 | 2400 | 31 | 54 | 32 | 3000 | 31 | 57 | 32 | 3000 | 31 | 57 | 32 | 4000 | 39 | 57 | 40 |
| 100 | 1800 | 28 | 44 | 30 | 1800 | 28 | 50 | 29 | 2400 | 31 | 54 | 32 | 3000 | 31 | 57 | 32 | 4000 | 39 | 57 | 40 | 4000 | 39 | 57 | 40 | 4500 | 39 | 62 | 40 |
| 125 | 2000 | 28 | 47 | 30 | 2600 | 31 | 49 | 33 | 3000 | 31 | 57 | 32 | 4000 | 39 | 57 | 40 | 4000 | 39 | 57 | 40 | 4500 | 39 | 62 | 40 | 5000 | 39 | 64 | 40 |
| 150 | 2100 | 28 | 47 | 30 | 2600 | 31 | 49 | 33 | 3000 | 31 | 57 | 32 | 4000 | 39 | 57 | 40 | 4500 | 39 | 62 | 40 | 5000 | 39 | 64 | 40 | 5500 | 39 | 68 | 40 |
| 200 | 2600 | 31 | 49 | 33 | 3100 | 31 | 52 | 33 | 4000 | 39 | 57 | 40 | 4500 | 39 | 62 | 40 | 5000 | 39 | 64 | 40 | 5500 | 39 | 68 | 40 | 6500 | 50 | 67 | 50 |
| 250 | 3100 | 31 | 52 | 33 | 3400 | 31 | 55 | 33 | 4500 | 39 | 62 | 40 | 5000 | 39 | 64 | 40 | 5500 | 39 | 68 | 40 | 6500 | 50 | 67 | 50 | 7000 | 50 | 67 | 50 |
| 300 | 3400 | 31 | 55 | 33 | 3800 | 31 | 58 | 33 | 5000 | 39 | 64 | 40 | 5500 | 39 | 68 | 40 | 6500 | 50 | 67 | 50 | 7000 | 50 | 67 | 50 | 8000 | 50 | 70 | 50 |
| 350 | 3400 | 31 | 55 | 33 | 4500 | 43 | 53 | 43 | 5000 | 39 | 64 | 40 | 7000 | 50 | 67 | 50 | 7000 | 50 | 67 | 50 | 8000 | 50 | 70 | 50 | .... | .. | .. | .. |
| 400 | 3800 | 31 | 58 | 33 | 5200 | 43 | 55 | 43 | 6000 | 43 | 65 | 43 | 7000 | 50 | 67 | 50 | 7000 | 50 | 67 | 50 | 8000 | 50 | 70 | 50 | .... | .. | .. | .. |
| 500 | 5200 | 43 | 55 | 43 | 6000 | 43 | 65 | 43 | 6500 | 43 | 65 | 43 | 8000 | 50 | 70 | 50 | 8000 | 50 | 70 | 50 | 9000 | 50 | 74 | 50 | .... | .. | .. | .. |

bearings usually are grease lubricated, although oil lubrication may be used on large high-speed motors and for the thrust bearing of large vertical machines.

Motors which may be belted (flat belt) with two bearings are indicated in Table 16. Sliding bases and pulleys are available. Larger ratings require an extended base and shaft with an outboard bearing. Slide rails usually are supplied for belt adjustment. With multiple V-belt drive, two-bearing construction is suitable for ratings about 25 to 50% larger than with flat belt drive. One manufacturer offers a line of "sound isolating" bases for use with quiet-operating motors arranged for V-belt drive. These prevent or greatly reduce the transmission of motor vibration through the base to the building structure, where it might otherwise be amplified and become exceedingly objectionable. Their use is justified in locations where noise must be kept to a minimum.

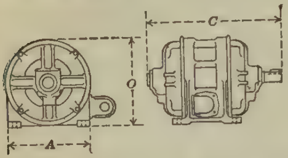


Fig. 15. Approximate Dimensions and Weights of General Purpose Squirrel-cage Induction Motors, Open Type, 60 Cycle

Gear-motors are available in ratings up to about 75 Hp., with output shaft speeds of 13 to 600 r.p.m. High-speed (1800 r.p.m. at 60 cycles) motors of either the squirrel-cage or wound-rotor type are employed.

In addition to the standard open type motors, drip-proof, splash-proof, totally-enclosed, totally-enclosed fan-cooled, and explosion-proof (Class I, Group D or Class II, Group G conditions) motors are available. Totally-enclosed motors are available in sizes up to 5 Hp., and totally-enclosed fan-cooled in sizes up to 400 Hp. (at 1800 r.p.m.). In the larger sizes, enclosed, self-ventilated, or enclosed, separately-ventilated motors must be used where enclosed motors are necessary.

**WEIGHTS AND DIMENSIONS.**—Frame numbers assigned to the various types of continuously rated squirrel-cage and wound-rotor motors are given in Tables 18 to 24 inclusive. Standardized mounting dimensions corresponding to the various frame numbers are given in Tables 14 and 15 and Figs. 7 and 8. Approximate net weights are given in Table 26.

Approximate weights and overall dimensions of squirrel-cage motors built in larger frames than those having standardized dimensions are given in Table 25 and Fig. 15.

**COSTS.**—Approximate 1936 costs of open, horizontal, sleeve-bearing, 60-cycle, squirrel-cage induction motors (Class A) are given in Fig. 16. Other types are more costly, the approximate percentage increase over the cost of Class A motors being as follows: Squirrel-cage, Class B, 0 to 10%; squirrel-cage, Class C, 5 to 15%; squirrel-cage, Class D, 30 to 50%; wound-rotor, 30 to 150%; quiet-operating squirrel-cage, 15%; quiet-operating wound-rotor, 25%.

Special mechanical features increase the cost approximately as follows: Ball bearings, 5%; sliding base, 5 to 15%; pulley, 2 to 5%; splash-proof motors (ball bearing), 7 1/2 to 15% over ball-bearing motor; totally-enclosed fan-cooled (ball bearing), 40 to 80% over ball-bearing motor; explosion proof (Class I, Group D conditions), 50 to 100% over

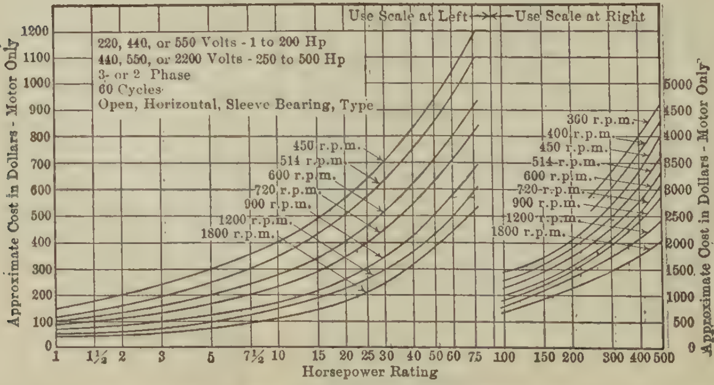


Fig. 16. Approximate 1936 Costs of Open Horizontal Sleeve-bearing 60-cycle Squirrel-cage Induction Motors, Class A

ball-bearing motor; enclosed, separately-ventilated, 20%; enclosed, self-ventilated, 40%; vertical, solid shaft, no thrust, 20 to 40%.

## 5. POLYPHASE SYNCHRONOUS LARGE POWER MOTORS

Standard horsepower and speed ratings of large power, polyphase, synchronous motors are given in Table 27.

**TYPES** in general use are as follows: *a*. High-speed motors; 1. General purpose; 2. Large high-speed. *b*. Low-speed motors; *c*. Special high-torque motors.

**High Speed Motors** are those operating at speeds of 500 r.p.m. or higher. The *general purpose* group includes those rated 200 Hp. or less with speed ratings of 500 to 1800 r.p.m. inclusive. The *large high-speed* group includes motors rated 250 Hp. or more, and all 2 pole ratings. The former have somewhat higher torques (see Table 28), but otherwise both groups have similar characteristics. High-speed synchronous motors have somewhat the same field of application as general purpose squirrel-cage induction motors, but their use is confined largely to ratings of 50 Hp. or above. Except in the large sizes, the synchronous motors are more costly than the induction motors, but their high efficiency and unity or leading power-factor frequently justify their use. Typical applications include: Pumps; motor-generator sets; fans; blowers; reciprocating and centrifugal compressors; rubber mills; line shafts.

**Low-speed Motors** are those operating at speeds of 450 r.p.m. or below. In general, these have lower costs than corresponding ratings of induction motors, as well as higher efficiencies and unity, or leading, power-factor. Typical applications include: Recipro-

**Table 26.—Approximate Weights of Polyphase Induction Motors Built in N.E.M.A. Standardized Frames**

| Frame No. | Approximate Weight, lb. | | | | | Frame No. | Approximate Weight, lb. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Squirrel-Cage | | Wound-rotor, Open | Base | Pulley | | Squirrel-Cage | | Wound-rotor, Open | Base | Pulley |
| | Open | Totally Enclosed, Fan-cooled | | | | | Open | Totally Enclosed, Fan-cooled | | | |
| 204 | 70 | ... | ... | 20 | 1 1/4 | 365 | 450 | 675 | 525 | 90 | 21 |
| 224 | 100 | 140 | 120 | 25 | 2 3/4 | 404 | 550 | 800 | 700 | 100 | 28 |
| 225 | 110 | 150 | 135 | 25 | 2 3/4 | 405 | 625 | 875 | 750 | 105 | 28 |
| 254 | 150 | 200 | 170 | 30 | 3 1/2 | 444 | 800 | 1100 | 900 | 125 | 40 |
| 284 | 200 | 275 | 235 | 35 | 5 | 445 | 875 | 1200 | 1000 | 130 | 40 |
| 324 | 300 | 350 | 310 | 45 | 8 1/2 | 504 | 1050 | 1400 | 1225 | 160 | 47 |
| 326 | 350 | 425 | 370 | 50 | 15 | 505 | 1225 | 1600 | 1400 | 170 | 84 |
| 364 | 425 | 600 | 500 | 75 | 21 | ...... | ...... | ...... | ...... | ...... | ...... |

**Table 27.—Standard Horsepower and Speed Ratings of Large Power Polyphase Synchronous Motors**

| Horsepower Ratings, High- and Low-speed Motors | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 20 | 40 | 75 | 150 | 225* | 350 | 500 | 800 | 1250 | 2000 | 3000 | 4500 |
| 25 | 50 | 100 | 175* | 250 | 400 | 600 | 900 | 1500 | 2250 | 3500 | 5000 |
| 30 | 60 | 125 | 200 | 300 | 450 | 700 | 1000 | 1750 | 2500 | 4000 | ....... |

Speed Ratings

| High-speed Motors | | | | Low-speed Motors | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| No. of Poles | Cycles | | | No. of Poles | Cycles | | | No. of Poles | Cycles | | | No. of Poles | Cycles | | |
| | 60 | 50 | 25 | | 60 | 50 | 25 | | 60 | 50 | 25 | | 60 | 50 | 25 |
| | Rev. per Min. | | | | Rev. per Min. | | | | Rev. per Min. | | | | Rev. per Min. | | |
| 2 | 3600 | 3000 | 1500 | 8 | ..... | ..... | 375 | 26 | 277 | 231 | 115 | 56 | 128 | 107 | ..... |
| 4 | 1800 | 1500 | 750 | 10 | ..... | ..... | 300 | 28 | 257 | 214 | 107 | 60 | 120 | 100 | ..... |
| 6 | 1200 | 1000 | 500 | 12 | ..... | ..... | 250 | 30 | 240 | 200 | 100 | 66 | 109 | 91 | ..... |
| 8 | 900 | 750 | ..... | 14 | ..... | 428 | 214 | 32 | 225 | 188 | 94 | 72 | 100 | 83 | ..... |
| 10 | 720 | 600 | ..... | 16 | 450 | 375 | 188 | 36 | 200 | 167 | 83 | 76 | 95 | 79 | ..... |
| 12 | 600 | 500 | ..... | 18 | 400 | 333 | 166 | 40 | 180 | 150 | 75 | 80 | 90 | ..... | ..... |
| 14 | 514 | ..... | ..... | 20 | 360 | 300 | 150 | 44 | 164 | 136 | ..... | 84 | 86 | ..... | ..... |
| ...... | ..... | ..... | ..... | 22 | 327 | 273 | 136 | 48 | 150 | 125 | ..... | 90 | 80 | ..... | ..... |
| ...... | ..... | ..... | ..... | 24 | 300 | 250 | 125 | 52 | 138 | 115 | ..... | ..... | ..... | ..... | ..... |

* Low speed motor only.

cating compressors; reciprocating, centrifugal and screw type pumps; Jordan engines; ball mills; metal rolling mills; electroplating generators; line shafts.

**High-torque Motors.**—Synchronous motors can be obtained with sufficient starting torque for any application and with moderate starting current. To meet the requirements of high starting torque with very low starting current several special types of synchronous motors are available.

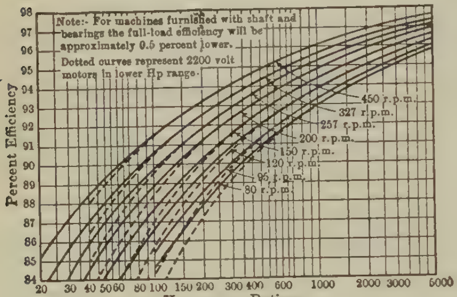


Fig. 17. Approximate Full-load Efficiency of Low-speed Synchronous Motors

The *super-synchronous* motor offered by one manufacturer is of conventional construction except that the stator as well as the rotor is mounted on bearings and a brake is provided which can be applied to prevent the stator from rotating. In starting the motor the brake is released, and the stator comes up to speed in reverse rotation while the rotor and connected load remain stationary. After the motor has been synchronized the brake is applied, gradually bringing the stator to rest, the rotor simultaneously accelerating the load up to speed. Thus the full synchronous pull-out torque of the motor is available for starting the load, and the current drawn from the line, in percent of full-load current, is essentially no greater than the percent starting torque required. Accurate control of the rate of acceleration of the load also is obtained.

The *synchronous induction* motor is essentially a wound-rotor induction motor which starts in the normal manner, but after it is up to speed direct-current excitation is applied to the rotor winding. This causes the motor to pull into step and operate as a synchronous motor. Thus the high starting torque with low starting current of the wound-rotor induction motor is obtained. Disadvantages of this type of motor are that its efficiency is somewhat lower than for other types of synchronous motors, and a high-current, low-voltage exciter is required for excitation.

Synchronous motors with *phase-wound* amortisseur windings are also available. These have the usual salient-pole field, but the amortisseur or starting winding, instead of being of the squirrel-cage type, is insulated, phase-wound and brought out to collector rings through which the circuit may be completed external to the motor. In some cases these motors are built with a revolving armature and a stationary field, eliminating the necessity for collector rings for the field or amortisseur winding, but requiring collector rings for the A.C. armature winding. In either case the motor is started with a high resistance in the amortisseur winding circuit, which is short circuited in steps as the motor comes up to speed. Thus high starting torque with relatively low starting current is obtained.

All of these special types of synchronous motors are considerably more costly than the conventional type, and are justified only in particular applications involving severe starting duty and low starting current.

## Operating Characteristics

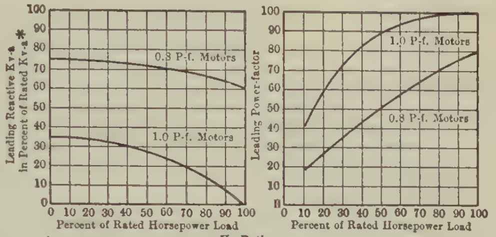
**POWER-FACTOR.**—Standard power-factor ratings of synchronous motors are unity and 0.8 leading, but motors rated at leading power-factors down to 0.2 are available. The unity-power-factor machines are least costly and have the highest efficiency, but the leading power-factor machines often are justified where additional leading kva. is required.

**TEMPERATURE RISE.**—The temperature rise of standard open-type, unity power-factor, high-speed synchronous motors is 40° C., measured by thermometer, for the armature winding, and 50° C., measured by resistance, for the field winding; for leading power-factor, high-speed motors it is 40° C., by thermometer, for the armature winding, and 60° C. by resistance, for the field winding. Splash-proof motors are rated 50° C. rise by thermometer, for the armature winding, and 60° C. rise, by resistance, for the field winding. The temperature rise of standard open-type unity- and 0.8-power-factor, low-speed synchronous motors is 50° C., by thermometer, for the armature winding, and 60° C., by resistance, for the field winding.

Like general purpose induction motors, high-speed synchronous motors rated 40° C.

usually are given a service factor of 15% *i. e.*, they may be operated at 1.15 times rated load without injurious heating.

**EFFICIENCY.**—Typical full-load efficiencies of unity-power-factor high-speed motors are given in Table 28 and of low-speed synchronous motors in Fig. 17. Efficiencies for machines supplied without bearings do not include windage or friction losses. Motors rated 0.8 power-factor have efficiencies ranging from 0.5% to 4% lower than for unity power-factor machines.



*$\text{Rated Kv-a} = 0.746 \times \frac{\text{Hp. Rating}}{\text{Full-load Eff.} \times \text{P-f. Rating}}$

FIG. 18. Power Factor Relations of Synchronous Motors

A synchronous motor operating at full-load with rated excitation delivers *leading reactive kva.* to the power system. The value is determined by

$$\text{kva.} = \frac{0.746 \times \text{Hp. rating}}{\text{Efficiency} \times \text{Power-factor}} \sqrt{1 - (\text{Power-factor})^2}$$

When operating at partial loads, with rated excitation maintained, the power-factor becomes more leading, and more leading kva. is supplied to the system. See Fig. 18.

**TORQUE.**—Minimum values of starting, pull-in, and pull-out torques for general-purpose, large high-speed, and low-speed synchronous motors are given in Table 29. The pull-in torque of a synchronous motor varies with the fly-wheel effect ($WR^2$) of its connected load; the higher the $WR^2$, the lower the load which the motor is capable of pulling into synchronism. The values of pull-in torque given in Table 29 are based on loads having a normal $WR^2$. However, motors with considerably higher starting, pull-in, and pull-out torques can be obtained, though usually at increased cost.

**Table 28.—Approximate Full-load Efficiencies of General Purpose and Large High-speed Synchronous Motors**

Unity Power-factor, 3-Phase, 60 Cycles

| Hp. | Voltage | Revolutions per Minute | | | | | |
|---|---|---|---|---|---|---|---|
| | | 514 | 600 | 720 | 900 | 1200 | 1800 |
| | | Approximate Full-load Efficiency | | | | | |
| 20 | 220, 440, 550 | 85.6 | 86.5 | 87.5 | 88.3 | 89.0 | 89.4 |
| | 2200 | 84 | 85 | 86 | 86.5 | 87.5 | 87.8 |
| 25 | 220, 440, 550 | 86.5 | 87.3 | 88.2 | 88.9 | 89.6 | 89.8 |
| | 2200 | 85.4 | 86.1 | 87 | 87.7 | 88.4 | 88.8 |
| 30 | 220, 440, 550 | 87.2 | 87.9 | 88.7 | 89.4 | 90 | 90.3 |
| | 2200 | 86.5 | 87 | 87.8 | 88.5 | 89.1 | 89.4 |
| 40 | 220, 440, 550 | 88.2 | 88.8 | 89.6 | 90.1 | 90.6 | 90.9 |
| | 2200 | 87.8 | 88.2 | 89 | 89.6 | 90.1 | 90.4 |
| 50 | 220, 440, 550 | 88.9 | 89.6 | 90.3 | 90.6 | 91.2 | 91.4 |
| | 2200 | 88.8 | 89.2 | 89.8 | 90.3 | 90.8 | 91 |
| 60 | 220, 440, 550 | 89.5 | 90.1 | 90.6 | 91.1 | 91.5 | 91.7 |
| | 2200 | 89.4 | 89.8 | 90.5 | 90.8 | 91.3 | 91.4 |
| 75 | 220, 440, 550 | 90.2 | 90.8 | 91.1 | 91.5 | 92 | 92.1 |
| | 2200 | 90.2 | 90.8 | 91.1 | 91.4 | 91.8 | 91.9 |
| 100 | 220, 440, 550 | 91 | 91.4 | 91.8 | 92.1 | 92.5 | 92.6 |
| | 2200 | 91 | 91.4 | 91.8 | 92.1 | 92.5 | 92.5 |
| 125 | 220<br>440<br>550<br>2200 | 91.6 | 92 | 92.2 | 92.5 | 92.9 | 92.9 |
| 150 | | 92 | 92.3 | 92.6 | 92.9 | 93.1 | 93.2 |
| 200 | | 92.7 | 93 | 93.1 | 93.4 | 93.6 | 93.6 |
| 250 | | 93.2 | 93.4 | 93.6 | 93.7 | 93.9 | 93.9 |
| 300 | | 93.5 | 93.7 | 93.8 | 94 | 94.1 | 94.2 |
| 350 | | 93.8 | 94 | 94.1 | 94.2 | 94.3 | 94.3 |
| 400 | | 94.1 | 94.2 | 94.3 | 94.4 | 94.5 | 94.5 |
| 450 | | 94.3 | 94.4 | 94.5 | 94.6 | 94.6 | 94.6 |
| 500 | | 94.5 | 94.6 | 94.6 | 94.7 | 94.8 | 94.8 |
| 600 | | 94.8 | 94.8 | 94.9 | 95 | 95 | 95 |
| 700 | | 95 | 95 | 95.1 | 95.2 | 95.1 | 95.1 |
| 800 | | 95.2 | 95.2 | 95.2 | 95.3 | 95.3 | 95.3 |
| 900 | | 95.4 | 95.4 | 95.4 | 95.4 | 95.4 | 95.4 |
| 1000 | | 95.5 | 95.5 | 95.5 | 95.6 | 95.5 | 95.5 |

**Table 29.—Minimum Starting, Pull-in, and Pull-out Torques of Standard Synchronous Motors**

| Speed Range, R.p.m. | Hp. Range | Unity Power-factor | | | 0.8 Power-factor | | |
|---|---|---|---|---|---|---|---|
| | | Torque, % of Full-load Torque | | | Torque, % of Full-load Torque | | |
| | | Starting | Pull-in * | Pull-out | Starting | Pull-in * | Pull-out |
| General Purpose Motors | | | | | | | |
| 514–1200 | ........... | 110 | 110 | 175 | 125 | 125 | 250 |
| 1800 | ........... | 110 | 110 | 150 | 125 | 125 | 200 |
| Large High-speed Motors | | | | | | | |
| 514–1800 | 200–500 | ... | ... | ... | 125 | 125 | 200 |
| 514–1800 | 250–500 | 110 | 110 | 150 | ... | ... | ... |
| 514–1800 | 600 and over | 85 | 85 | 150 | 100 | 100 | 200 |
| Low-speed Motors | | | | | | | |
| ........... | ........... | 40 | 30 | 140 | 40 | 30 | 200 |

* Based on normal load $WR^2$.

**STARTING CURRENT.**—Full-voltage starting currents range from 500 to 800% for high-speed motors, and from 250 to 500% for low-speed motors.

**VOLTAGE VARIATION.**—All synchronous motors should operate successfully at rated load and frequency at any voltage not more than 10% above or below normal, but not necessarily in accordance with the standards of performance established for operation

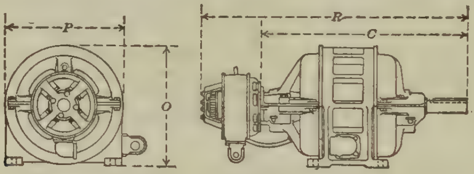


Fig. 19. Dimensions of High-speed Synchronous Motors. See Table 30

**Table 30.—Approximate Weights and Dimensions of High-speed Synchronous Motors**

Unity Power-factor, Open Type, 60 Cycles. Dimensions in inches. See Fig. 19

| Hp. | Approx. Wt., lb. | C | R | O | P | Approx. Wt., lb. | C | R | O | P | Approx. Wt., lb. | C | R | O | P |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1800 r.p.m. | | | | | 1200 r.p.m. | | | | | 900 r.p.m. | | | | |
| 50 | ..... | .. | .. | .. | .. | 1600 | 46 | 60 | 25 | 25 | 1800 | 48 | 63 | 29 | 29 |
| 60 | ..... | .. | .. | .. | .. | 1600 | 46 | 60 | 25 | 25 | 2100 | 50 | 65 | 29 | 29 |
| 75 | 1750 | 48 | 60 | 29 | 29 | 1700 | 48 | 61 | 29 | 29 | 2100 | 50 | 65 | 29 | 29 |
| 100 | 1750 | 48 | 60 | 29 | 29 | 2000 | 50 | 65 | 29 | 29 | 2400 | 52 | 67 | 32 | 31 |
| 125 | 2000 | 52 | 63 | 29 | 29 | 2400 | 52 | 67 | 32 | 31 | 2900 | 57 | 72 | 32 | 31 |
| 150 | 2200 | 52 | 63 | 29 | 29 | 2000 | 50 | 65 | 29 | 29 | 2900 | 57 | 72 | 32 | 31 |
| 200 | 2000 | 50 | 65 | 29 | 29 | 3200 | 61 | 77 | 32 | 31 | 3200 | 61 | 77 | 32 | 31 |
| 250 | 3600 | 60 | 71 | 33 | 31 | 3000 | 55 | 70 | 32 | 31 | 5800 | 70 | 84 | 45 | 44 |
| 300 | 4500 | 60 | 75 | 40 | 39 | 4500 | 60 | 75 | 40 | 39 | 5800 | 70 | 84 | 45 | 44 |
| 350 | 5000 | 62 | 77 | 40 | 39 | 4500 | 60 | 75 | 40 | 39 | 6700 | 71 | 87 | 45 | 44 |
| 400 | 5000 | 62 | 77 | 40 | 39 | 5000 | 62 | 77 | 40 | 39 | 6700 | 71 | 87 | 45 | 44 |
| 450 | 5500 | 80 | 65 | 40 | 39 | 5500 | 80 | 65 | 40 | 39 | 7300 | 73 | 89 | 45 | 44 |
| 500 | 5500 | 80 | 65 | 40 | 39 | 5500 | 80 | 65 | 40 | 39 | 7300 | 73 | 89 | 45 | 44 |
| | 720 r.p.m. | | | | | 600 r.p.m. | | | | | 514 r.p.m. | | | | |
| 50 | 2100 | 80 | 65 | 29 | 29 | 2,400 | 52 | 67 | 32 | 31 | .... | .. | .. | .. | .. |
| 60 | 2400 | 52 | 67 | 32 | 31 | 2,400 | 52 | 67 | 32 | 31 | .... | .. | .. | .. | .. |
| 75 | 2400 | 52 | 67 | 32 | 31 | 2,900 | 57 | 72 | 32 | 31 | .... | .. | .. | .. | .. |
| 100 | 2900 | 57 | 72 | 32 | 31 | 3,200 | 61 | 77 | 32 | 31 | 3,200 | 61 | 77 | 32 | 31 |
| 125 | 2900 | 57 | 72 | 32 | 31 | 3,200 | 61 | 77 | 32 | 31 | 4,800 | 62 | 78 | 51 | 50 |
| 150 | 4500 | 62 | 78 | 46 | 45 | 4,800 | 62 | 78 | 51 | 50 | 5,800 | 57 | 75 | 56 | 54 |
| 200 | 4800 | 62 | 78 | 51 | 50 | 6,300 | 67 | 83 | 51 | 50 | 5,800 | 57 | 75 | 56 | 54 |
| 250 | 6300 | 67 | 83 | 51 | 50 | 6,300 | 67 | 83 | 51 | 50 | 7,500 | 62 | 82 | 56 | 54 |
| 300 | 6300 | 67 | 83 | 51 | 50 | 7,500 | 62 | 82 | 56 | 54 | 9,000 | 68 | 88 | 56 | 54 |
| 350 | 7000 | 68 | 82 | 51 | 50 | 7,500 | 62 | 82 | 56 | 54 | 9,000 | 68 | 88 | 56 | 54 |
| 400 | 7000 | 68 | 82 | 51 | 50 | 9,000 | 68 | 88 | 56 | 54 | 9,000 | 68 | 88 | 56 | 54 |
| 450 | 8300 | 67 | 86 | 51 | 50 | 9,000 | 68 | 88 | 56 | 54 | 10,200 | 70 | 90 | 56 | 54 |
| 500 | 8300 | 67 | 86 | 51 | 50 | 10,200 | 70 | 90 | 56 | 54 | 10,200 | 70 | 90 | 56 | 54 |

at normal rating. Their starting currents and pull-out torques are approximately proportional to the applied voltage, while their starting and pull-in torques are approximately proportional to the square of the voltage. Their pull-out torques also are approximately proportional to the excitation current.

## Mechanical Features

High-speed synchronous motors usually are furnished with feet on the stator frame and end-shield-type bearings. Pedestal-type bearings, with or without base, may be used in sizes above 1 Hp. per r.p.m. A direct-connected exciter often is supplied. This is overhung on an extension of the motor shaft on the collector ring end (opposite the drive end). The exciter frame is bolted to the corresponding end-shield, or is supported from the base of pedestal bearing machines.

Low-speed synchronous motors commonly are supplied without base, shaft or bearings, the rotor being mounted directly on the shaft of the driven machine. In this case, a completely split rotor spider, or split hub spider, may be supplied to facilitate assembly at the installation. Foundation caps usually are provided for support of the stator frame. A shaft, pedestal-type bearings, and base or sole plates are provided where a self-contained unit is required.

Direct-connected exciters seldom are used with low-speed synchronous motors, since higher speed belt- or motor-driven exciters are more economical.

Sleeve bearings, oil-ring-lubricated are standard for most horizontal synchronous motors, although grease-lubricated ball bearings can be obtained at increased cost. Vertical motors usually are furnished with ball bearings in the smaller sizes, and oil-lubricated plate-type bearings in the larger sizes.

The horsepower limits for two-bearing belt drive are essentially the same as for induction motors (Table 16). Pulleys and sliding bases are available. Vertical synchronous motors, as well as the standard horizontal machines, can be obtained.

In addition to the open motors, drip-proof, splash-proof, totally-enclosed fan-cooled, enclosed self-ventilated, and enclosed separately-ventilated motors are available.

**WEIGHTS AND DIMENSIONS.**—Approximate overall dimensions and net weights of high-speed synchronous motors are given in Table 30 and Fig. 19. Low-speed synchronous motors generally are specially designed for the particular application, with resultant wide variation in weights and dimensions for each rating.

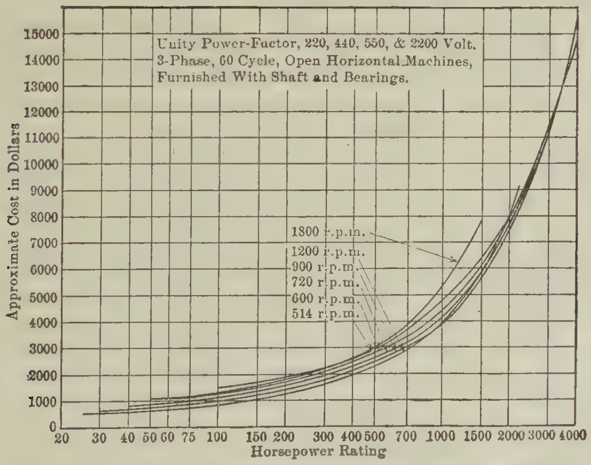


Fig. 20. Approximate 1936 Costs of High-speed Synchronous Motors

**COSTS.**—Approximate costs of open, horizontal, unity-power-factor high-speed synchronous motors, furnished with shaft and two bearings are given in Fig. 20. Special features increase the cost approximately as follows: 0.8 power-factor, 17%; 4000-volt (75 Hp. and larger), 7 1/2 %; 6600-volt (400 Hp. and larger), 17 to 30%; ball bearings, 10%; drip-proof covers, 5%; enclosed self-ventilated construction, 40%; enclosed

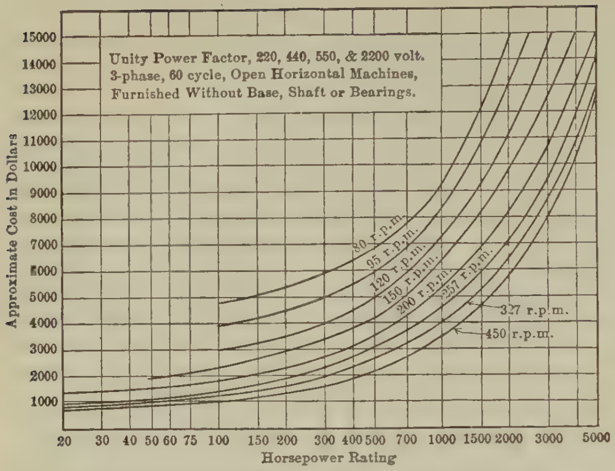


FIG. 21. Approximate Costs of Open, Horizontal, Unity-power-factor, Low-speed Synchronous Motors

separately ventilated construction, 10%; splash proof construction, 15%; totally-enclosed fan-cooled construction, 125 to 150%; vertical construction, 40 to 65%.

Approximate costs of open, horizontal, unity-power-factor low-speed synchronous motors furnished without base, shaft, or bearings, but with foundation caps are given in Fig. 21. Special features increase the cost as follows: 0.8 power-factor, 20%; 4000-volt (75 Hp. and larger), 5 to 10%; 6600-volt (400 Hp. and larger), 17 to 30%; split rotor, 2 to 5%; split stator, 7 1/2 to 15%; shaft and two bearings, 30%; base, shaft, and two bearings, 35%; ball bearings, 10%; splash-proof construction, 15%; vertical construction, 70%.

## 6. DIRECT-CURRENT LARGE POWER MOTORS

The following types of large power, direct-current motors are in general use:

*a.* General purpose constant speed motors; *b.* Adjustable speed motor; *c.* Large direct-current motors; *d.* Fan motors; *e.* Crane and hoist motors; *f.* Mill-type motors; *g.* Metal rolling mill motors; *h.* Elevator motors; *i.* Low-speed, engine-type motors.

**General Purpose Motors.**—Standard horsepower and speed ratings for general purpose constant speed motors are given in Table 31. These motors are available with shunt, series, or compound-wound field windings. The shunt-wound motors have the lowest speed regulation (change in speed with change in load), are lowest in cost and have the

**Table 31.—Standard Horsepower and Speed Ratings for General Purpose Constant Speed D.C. Motors**

Ratings above the horizontal lines are suitable for two-bearing belt drive

| Hp. | Full-load Speed in R.p.m. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | .... | .... | .... | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 3/4 | .... | .... | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 1 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 1 1/2 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 2 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 3 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 5 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 7 1/2 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 10 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 15 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 20 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 25 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 30 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 40 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 50 | 3500 | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 60 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 75 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 100 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 125 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 150 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |
| 200 | .... | 1750 | 1150 | 850 | 690 | 575 | 500 | 450 | 400 | 350 | 300 | 250 | 200 | 150 | 100 |

widest application. The compound-wound motors have somewhat higher speed regulation, and are used on loads requiring high starting torque, and on those subject to torque pulsation. Series-wound motors have very high speed regulation, and should not be used with belt drive or on any load in which the torque might drop below 15% of full-load torque, since a dangerously high speed would be developed. Series motors are used on loads requiring a very high starting torque or severe accelerating duty, and where high speed regulation is advantageous or of no consequence.

Shunt-wound constant speed motors, having a full load speed of 1750 r.p.m. or below, usually can be operated at some increase in speed, without increase in horsepower, by inserting a resistance in series with the field winding. The maximum increase in speed permissible varies from about 10% for the higher speed ratings to 100% for the lower speed ratings. Speed variation by shunt field control generally is not advisable on compound-wound motors, since, with a weakened field, the motor characteristics approach those of a series motor and high speed regulation results.

**Adjustable Speed Motors** are shunt-wound machines, designed particularly for adjustable speed operation by varying the strength of the shunt field. Standard horsepower and basic (minimum) speed ratings are the same as for the constant speed motors (Table 31). Speed ranges (ratio of maximum speed to basic speed) of 3 : 1 and 4 : 1 are standard, although motors having a speed range of 6 : 1 can be obtained at increased cost. In any case, the maximum speed usually is limited to the values shown in Table 32. Three different horsepower ratings may be given to adjustable speed motors: 1. A standard continuous horsepower rating over entire speed range, with a temperature rise of 50° C. from basic speed to 150% of basic speed, and a temperature rise of 40° C. from 150% of basic speed to the maximum speed. 2. A standard intermittent horsepower rating over the entire speed range, with a temperature rise of 50° C. in 1 hour. 3. A *tapered rating* consisting of a certain standard horsepower from basic speed to 150% of basic speed, the next larger standard horsepower at 3 times basic speed and higher, and a horsepower varying along a straight line from the lower value to the upper value as the speed varies from 150% of basic speed to 3 times basic speed. The temperature rise is 50° C. continuous from basic speed to 150% of basic speed, and 40° C. continuous from 150% of basic speed to maximum speed.

In general, a motor rated in accordance with [1] will have the same cost as a motor of the next larger horsepower rated in accordance with [2], and the same cost as a tapered rating motor whose lower horsepower is the same. Adjustable speed D.C. motors are used most extensively on machine tools.

**Large D.C. Motors.**—Standard horsepower rating of these motors are as follows: 250, 300, 350, 400, 500, 600, 700, 800, 900.

Standard speed ratings, in r.p.m., are: 1150, 850, 690, 575, 500, 450, 400, 350, 300, 250, 225, 200, 175, 150, 125, 110, 100. These speed ratings apply to all standard sizes of large D.C. motors, excepting that 500 Hp. is the largest size to which the 1150 r.p.m. rating applies. These motors may be constant speed or adjustable speed (constant horsepower). The maximum speed range varies from 5 : 1 for the smaller and lower speed ratings, to about 1 1/2 : 1 for the larger and higher speed ratings. Typical applications include pumps, fans, rubber mills and paper mills.

**Crane and Hoist Motors** normally are series-wound, totally-enclosed machines, and are rated 15 minutes, 55° C. temperature rise, or 30 minutes, 55° C. rise. Standard ratings range from 1 to 50 Hp. at speeds of 500 to 1500 r.p.m. Shaft extensions at both ends, for brake and drive, respectively, and pinion, gear and reduced speed axle mounted integrally with the motor are features available. Although higher in cost than standard motors, their compactness, rugged mechanical construction, and low-inertia armature justifies their use on cranes, hoists, and similar loads involving intermittent service and severe reversing duty.

**Table 32.—Maximum Speed by Field Control of Shunt-wound Adjustable Speed D.C. Motors**

| Continuous Hp. Rating | * Maximum Speed, r.p.m. | Continuous Hp. Rating | * Maximum Speed, r.p.m. | Continuous Hp. Rating | *Maximum Speed, r.p.m. | Continuous Hp. Rating | * Maximum Speed, r.p.m. |
|---|---|---|---|---|---|---|---|
| 1/2 | 3450 | 3 | 2300–2500 | 20 | 1500–1800 | 60 | 1200–1500 |
| 3/4 | 3450 | 5 | 2000–2300 | 25 | 1500–1800 | 75 | 1200 |
| 1 | 2500–2875 | 7 1/2 | 2000–2300 | 30 | 1200–1800 | 100 | 1200 |
| 1 1/2 | 2500–2875 | 10 | 1800–2300 | 40 | 1200–1600 | 125 | 1200 |
| 2 | 2500–2875 | 15 | 1500–1800 | 50 | 1200–1500 | 150 | 1200 |

* Maximum speed also is limited to six times the basic (minimum) speed rating.

**Fan Motors** are shunt-wound machines suitable for direct connection to centrifugal fans. The standard horsepower ratings are 1/2, 3/4, 1, 1 1/2, 2, 3, 5, 7 1/2, 10, 15, 20, 25, 30, 40 and 50. Standard speed ratings for both the constant speed and adjustable speed motors are given in Table 33. In the case of the adjustable speed machines, the normal horsepower rating corresponds to that obtained at maximum speed (weakened field). At the minimum speed (full field), it is assumed that the horsepower will be reduced in proportion to the cube of the reduction in speed.

### Table 33.—Standard Speed Ratings for D.C. Fan Motors

| Adjustable Speed, minimum r.p.m. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 94 | 112 | 133 | 160 | 187 | 227 | 280 | 320 | 380 | 455 | 575 |
| Adjustable Speed, maximum r.p.m. | | | | | | | | | | |
| *140 | *170 | *200 | *240 | *280 | *340 | *420 | *480 | †580 | *680 | 840 |
| 150 | 180 | 210 | 250 | 290 | 360 | 440 | 500 | 600 | 700 | 870 |
| 160 | 190 | 220 | 260 | 300 | 380 | 460 | 520 | 620 | 720 | 900 |
| 170 | 200 | 230 | 270 | 320 | 400 | 480 | 540 | 640 | 750 | 950 |
| .... | .... | 240 | 280 | 340 | 420 | .... | 560 | 660 | 780 | ... |
| .... | .... | .... | .... | .... | .... | .... | 580 | 680 | 810 | ... |
| .... | .... | .... | .... | .... | .... | .... | .... | .... | 840 | ... |

* Standard speed ratings of constant speed fan motors.
† Corresponding speed rating of constant speed motor is 575 r.p.m.

**Mill-type Motors** are heavy duty machines, designed particularly for such service as steel mill auxiliaries, electric power shovels, and cranes. The basic motor is series-wound, totally-enclosed, intermittently rated 1 hour, 75° C. temperature rise with horsepower and speed ratings as shown in Table 34. The ratings from 7 1/2 to 75 Hp. inclusive have been standardized by the A.I.S.E.E. with standardized shaft, mounting dimensions, and other features. In addition to the basic motor, shunt- and compound-wound, open, semi-enclosed, and enclosed forced ventilated, 1/2 hour and continuous rated motors are available. In general the horsepower and speed ratings for such motors differ from those of the basic motors. All motors have Class B insulation, roller or double-row ball bearings, and shaft extensions on both ends. A pinion, gear and reduced speed axle with sleeve or roller bearings also can be obtained. Rugged mechanical construction, low-inertia armatures, compactness, and accessibility of all parts are features of these motors.

### Table 34.—Standard Horsepower and Speed Ratings of Mill-type D.C. Motors

Series wound, totally enclosed, 1 hour 75° C. temperature rise

| Hp............. | 5 | 7 1/2 | 10 | 15 | 25 | 35 | 50 | 75 | 100 | 150 | 200 | 250 | 350 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Full-load, r.p.m... | 900 | 800 | 725 | 650 | 575 | 525 | 500 | 475 | 460 | 450 | 410 | 375 | 375 |

**Metal Rolling-mill Motors** are large, heavy-duty machines, designed for metal rolling mill drive. Standard horsepower and speed ratings are given in Table 35. The motors are arranged for adjustable speed operation by shunt field control. The standard speed range varies from about 1.5 : 1 to 2 : 1, depending on the horsepower and basic speed rating. However, speed ranges up to 5 : 1 can be obtained in some cases, though at higher cost. The motors normally are furnished with base and pedestal type bearings.

### Table 35.—Standard Horsepower and Speed Ratings of Adjustable Speed Metal Rolling Mill D.C. Motors

| Standard Basic-Speed Ratings, r.p.m. |
|---|
| 1150, 850, 690, 575, 500, 450, 400, 350, 300, 250, 225, 200, 175, 150, 125, 110, 100, 90, 80, 70, 65, 60, 55, 50 |

Standard Horsepower Ratings

| Hp. | Basic Speeds, r.p.m. | Hp. | Basic Speeds, r.p.m. | Hp. | Basic Speeds, r.p.m. |
|---|---|---|---|---|---|
| 250 | 100 to 1150 | 800 | 60 to 850 | 2250 | 50 to 350 |
| 300 | 100 to 1150 | 900 | 55 to 850 | 2500 | 50 to 350 |
| 350 | 100 to 1150 | 1000 | 50 to 850 | 3000 | 50 to 300 |
| 400 | 100 to 1150 | 1250 | 50 to 575 | 3500 | 50 to 250 |
| 500 | 90 to 1150 | 1500 | 50 to 500 | 4000 | 50 to 225 |
| 600 | 80 to 850 | 1750 | 50 to 500 | 4500 | 50 to 200 |
| 700 | 70 to 850 | 2000 | 50 to 400 | 5000 | 50 to 175 |

**Geared Elevator Hoisting Machines** employ high-speed motors similar to the general purpose motors. They usually are adjustable speed machines, with shunt field control or variable armature voltage control. Gearless elevators require a special low-speed driving motor. The hoisting drum is mounted on the motor shaft between the two bearings. A brake wheel for a solenoid brake usually is cast integrally with the drum. Heavy shafts and large bearings are provided. The motors are of the adjustable speed type with armature voltage control.

**Low-speed Engine-type D.C. Motors** are available in ratings from 100 to 250 Hp., at speeds of 150 to 327 r.p.m. The standard speeds are the same as for low-speed, 60-cycle, synchronous motors (Table 27). The motors are compound-wound, and are furnished without base, shaft, or bearings. They are used to drive low-speed machines, such as reciprocating compressors, the rotor being mounted on the shaft of the driven machine.

## Operating Characteristics

**TEMPERATURE RISE** of standard general purpose motors is 40° C., measured by thermometer for open motors, 50° C. for semi-enclosed, drip-proof, and splash-proof machines, and 55° C. for totally-enclosed and totally-enclosed fan-cooled machines. Open motors rated 40° C. usually are given a service factor of 1.15; that is, the manufacturers guarantee successful operation at 1.15 times rated load without injurious heating. D.C. motors generally are capable of delivering 50% overload in torque momentarily, without temperature guarantee.

**EFFICIENCY.**—Typical approximate full-load efficiencies of general purpose, constant-speed shunt-wound open-type motors are given in Fig. 22.

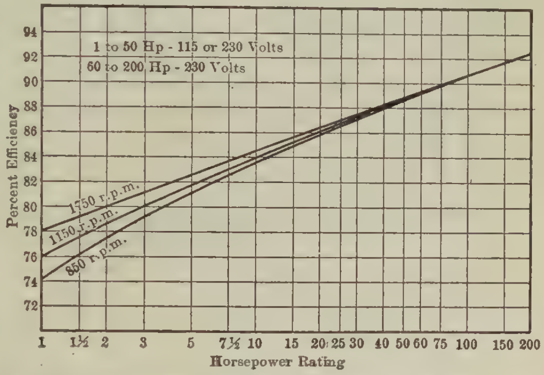


Fig. 22. Approximate Full-load Efficiency of General Purpose Constant-speed Shunt-wound Direct-current Motors, Open Type

**TORQUE.**—With full voltage on the shunt field (if provided), and with the armature current limited to full-load current the starting torque of any type of D.C. motor is 100% of full-load torque. For shunt-wound motors the starting torque will vary directly with the armature current; for series-wound motors, it varies approximately as the square of the armature current; for compound-wound motors the variation lies between these two limits, depending on the ratio of shunt and series windings. The armature current, in turn, is determined by the amount of resistance inserted in series with this winding at start. Standard starters usually limit the current to from 125 to 200% of full load current.

**SPEED REGULATION** from full load to no load, hot, based on full load speed should not exceed the following:

Constant-speed motors, shunt-wound: 5 Hp. and smaller, 12%; 7 1/2 to 200 Hp. inclusive, 10%; 250 Hp. and larger, 6%; Compound-wound, constant-speed motors, 25%.

Adjustable speed motors, shunt-wound: 2 to 5 Hp. inclusive, 22%; 7 1/2 to 200 Hp. inclusive, 15%; 250 Hp. and larger, at maximum speed, 4%; at basic speed, 6 to 15%.

At normal operating temperature, rated load and voltage, the variation above or below rated speed should not exceed the following:

Constant-speed motors, shunt- or compound-wound, 7 1/2 Hp. at 1150 r.p.m. or smaller, 7 1/2 %; larger than 7 1/2 Hp. at 1150 r.p.m., 5%.

Adjustable-speed motors, shunt-wound, 5 Hp., 650 r.p.m. (minimum speed) or smaller, 7 1/2 %; larger than 5 Hp., 650 r.p.m., 5%.

The variation in speed from full load, cold, to full load, hot, should not exceed 10%, based on full-load speed.

**VOLTAGE VARIATION.**—All D.C. motors should operate successfully at normal rated load at any voltage not more than 10% above or below normal, but not necessarily in accordance with the standards of performance established for operation at normal rated voltage. In general, with a given load, an increase in voltage results in an increase in speed and *vice versa.*

Any shunt-wound motor may be operated as an adjustable speed machine by maintaining constant voltage on the shunt field while varying the voltage on the armature.

### Table 36.—Frame Sizes for General Purpose Constant Speed D.C. Motors

Open Type

| Speed, r.p.m. | Horsepower 1/2 | 3/4 | 1 | 1 1/2 | 2 | 3 | 5 | 7 1/2 |
|---|---|---|---|---|---|---|---|---|
| | Frame Number * | | | | | | | |
| 1750 | ........ | .......... | 204 | 224 | 225 | 225 | 254 | 284 |
| 1150 | ........ | 204 | 204 | 224 | 225 | 254 | 284 | ....... |
| 850 | 204 | 224 | 225 | 254 | 254 | 284 | .......... | ....... |

* See Tables 14 and 15, and Figs. 7 and 8 for standard mounting dimensions.

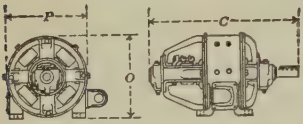


FIG. 23. Dimensions General Purpose Constant-speed Direct-current Motors. See Table 37

The speed will be approximately proportional to the armature voltage. The basic speed of such a machine is the maximum speed (full voltage on the armature). At reduced speeds, the horsepower is reduced approximately in proportion to the speed. Thus *variable voltage control* gives constant torque characteristics, as contrasted with speed variation by field control which gives essentially constant horsepower characteristics. Where the source of variable voltage direct current is such that the voltage provided is essentially independent of the current drawn, the motor is truly an adjustable speed machine, *i.e.*, its speed when once adjusted to given value remains essentially constant,

### Table 37.—Approximate Weights and Dimensions of Open-type General Purpose Constant Speed D.C. Motors

| Hp. | Approx. Wt., lb. | $C$ | $O$ | $P$ | Approx. Wt., lb. | $C$ | $O$ | $P$ | Approx. Wt., lb. | $C$ | $O$ | $P$ | Approx. Wt., lb. | $C$ | $O$ | $P$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3450 r.p.m. | | | | 1750 r.p.m. | | | | 1150 r.p.m. | | | | 850 r.p.m. | | | |
| 5 | .... | .. | .. | .. | .... | .. | .. | .. | .... | .. | .. | .. | 450 | 32 | 19 | 19 |
| 7 1/2 | .... | .. | .. | .. | .... | .. | .. | .. | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 |
| 10 | .... | .. | .. | .. | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 | 650 | 38 | 21 | 21 |
| 15 | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 | 650 | 38 | 21 | 21 | 725 | 40 | 21 | 21 |
| 20 | 500 | 34 | 19 | 19 | 650 | 38 | 21 | 21 | 725 | 40 | 21 | 21 | 1000 | 43 | 24 | 24 |
| 25 | 500 | 34 | 19 | 19 | 725 | 40 | 21 | 21 | 950 | 43 | 24 | 24 | 1000 | 43 | 24 | 24 |
| 30 | 650 | 38 | 21 | 21 | 725 | 40 | 21 | 21 | 1000 | 43 | 24 | 24 | 1325 | 49 | 28 | 27 |
| 40 | 725 | 40 | 21 | 21 | 950 | 43 | 24 | 24 | 1000 | 43 | 24 | 24 | 1750 | 54 | 31 | 30 |
| 50 | 950 | 43 | 24 | 24 | 1000 | 43 | 24 | 24 | 1325 | 49 | 28 | 27 | 1950 | 55 | 31 | 30 |
| 60 | .... | .. | .. | .. | 1000 | 43 | 24 | 24 | 1750 | 54 | 31 | 30 | 2600 | 63 | 35 | 34 |
| 75 | .... | .. | .. | .. | 1325 | 49 | 28 | 27 | 1950 | 55 | 31 | 30 | 2800 | 64 | 35 | 34 |
| 100 | .... | .. | .. | .. | 1950 | 55 | 31 | 30 | 2600 | 63 | 35 | 34 | 2800 | 64 | 35 | 34 |
| 125 | .... | .. | .. | .. | 2600 | 63 | 35 | 34 | 2800 | 64 | 35 | 34 | 3450 | 66 | 36 | 34 |
| 150 | .... | .. | .. | .. | 2800 | 64 | 35 | 34 | 3500 | 68 | 36 | 34 | 4600 | 78 | 41 | 39 |
| 200 | .... | .. | .. | .. | 3500 | 68 | 36 | 34 | 4600 | 78 | 41 | 39 | 5300 | 80 | 41 | 39 |
| | 690 r.p.m. | | | | 575 r.p.m. | | | | 500 r.p.m. | | | | 450 r.p.m. | | | |
| 2 | .... | .. | .. | .. | .... | .. | .. | .. | 450 | 32 | 19 | 19 | 450 | 32 | 19 | 19 |
| 3 | 450 | 32 | 19 | 19 | 450 | 32 | 19 | 19 | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 |
| 5 | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 | 650 | 38 | 21 | 21 | 650 | 38 | 21 | 21 |
| 7 1/2 | 500 | 34 | 19 | 19 | 650 | 38 | 21 | 21 | 725 | 40 | 21 | 21 | 725 | 40 | 21 | 21 |
| 10 | 650 | 38 | 21 | 21 | 725 | 40 | 21 | 21 | 950 | 43 | 24 | 24 | 1000 | 43 | 24 | 24 |
| 15 | 950 | 43 | 24 | 24 | 1000 | 43 | 24 | 24 | 1200 | 48 | 28 | 27 | .... | .. | .. | .. |
| 20 | 1000 | 43 | 24 | 24 | 1325 | 49 | 28 | 27 | 1750 | 54 | 31 | 30 | .... | .. | .. | .. |
| 25 | 1325 | 49 | 28 | 27 | 1750 | 54 | 31 | 30 | 1950 | 55 | 31 | 30 | .... | .. | .. | .. |
| 30 | 1750 | 54 | 31 | 30 | 1950 | 55 | 31 | 30 | 2600 | 63 | 35 | 34 | .... | .. | .. | .. |
| 40 | 1950 | 55 | 31 | 30 | 2600 | 63 | 35 | 34 | 2600 | 63 | 35 | 34 | .... | .. | .. | .. |
| 50 | 2600 | 63 | 35 | 34 | 2800 | 64 | 35 | 34 | 3100 | 64 | 36 | 34 | .... | .. | .. | .. |
| 60 | 2800 | 64 | 35 | 34 | 3100 | 64 | 36 | 34 | 3450 | 66 | 36 | 34 | .... | .. | .. | .. |
| 75 | 3100 | 64 | 36 | 34 | 3450 | 66 | 36 | 34 | 3450 | 66 | 36 | 34 | .... | .. | .. | .. |
| 100 | 3450 | 66 | 36 | 34 | 4500 | 73 | 41 | 39 | 5100 | 75 | 41 | 39 | .... | .. | .. | .. |
| 125 | 4500 | 73 | 41 | 39 | 5100 | 75 | 41 | 39 | 6000 | 76 | 43 | 42 | .... | .. | .. | .. |
| 150 | 5300 | 80 | 41 | 39 | 6000 | 76 | 43 | 42 | 6700 | 80 | 43 | 42 | .... | .. | .. | .. |
| 200 | 6500 | 78 | 43 | 42 | 6900 | 86 | 43 | 42 | .... | .. | .. | .. | .... | .. | .. | .. |
| | 400 r.p.m. | | | | 350 r.p.m. | | | | 300 r.p.m. | | | | | | | |
| 1 1/2 | 450 | 32 | 19 | 19 | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 | .... | .. | .. | .. |
| 2 | 450 | 32 | 19 | 19 | 500 | 34 | 19 | 19 | 500 | 34 | 19 | 19 | .... | .. | .. | .. |
| 3 | 500 | 34 | 19 | 19 | 650 | 38 | 21 | 21 | 650 | 38 | 21 | 21 | .... | .. | .. | .. |
| 5 | 725 | 40 | 21 | 21 | 725 | 40 | 21 | 21 | 950 | 43 | 24 | 24 | .... | .. | .. | .. |
| 7 1/2 | 950 | 43 | 24 | 24 | 1000 | 43 | 24 | 24 | .... | .. | .. | .. | .... | .. | .. | .. |

**Table 38.—Approximate Weights of D.C. Motors Built in N.E.M.A. Standardized Frames**

| Frame No. | Open Motors | Totally-enclosed, Fan-cooled Motors | Base | Pulley |
|---|---|---|---|---|
| | Approximate Weight, lb. | | | |
| 204 | 105 | ... | 20 | 1 |
| 224 | 150 | 160 | 25 | 3 |
| 225 | 160 | 170 | 25 | 3 |
| 254 | 210 | 225 | 30 | 4 |
| 284 | 280 | 300 | 35 | 5 |

regardless of the load. A D.C. generator, driven at constant speed with voltage variation by field control, meets this requirement and commonly is used. On the other hand, if the applied armature voltage varies appreciably with the current drawn, as would be the case if a variable resistance were used to reduce the voltage, the drive would be of the adjustable varying speed type, *i.e.*, the speed varies appreciably with the load.

## Mechanical Features

Large power D.C. motors usually have feet on the stator frame and end-shield-type bearings, in sizes up to about 200 Hp. In the larger sizes, a base with pedestal-type bearings usually is supplied. Vertical motors, as well as the standard horizontal motors, can be obtained.

Sleeve bearings, oil-ring-lubricated, are standard for most horizontal machines. Ball bearings are standard on totally-enclosed, splash-proof, and the smaller vertical motors. They can be obtained at increased cost on all motors. Ball bearings usually are grease lubricated.

Motors which may be belted (flat belt) with two bearings are indicated in Table 31. Sliding bases and pulleys are available. Larger ratings require an extended base and shaft with outboard bearing. Slide rails usually are supplied for belt adjustment. With multiple V-belt drive, 2-bearing construction is suitable for ratings from 25 to 50% larger than with flat belt drive.

Gear-motors are generally available in sizes up to 7 1/2 Hp., with output shaft speeds of 13 to 600 r.p.m. High-speed (1750 r.p.m.) motors are employed.

In addition to the standard open type motors, drip-proof, splash-proof, totally-enclosed, totally-enclosed fan-cooled, and explosion-proof (Class I, Group D or Class II, Group G conditions) motors are available.

**WEIGHTS AND DIMENSIONS.**—Frame numbers assigned to general-purpose constant-speed D.C. motors are given in Table 36. Approximate weights and overall dimensions of general-purpose constant-speed motors built in larger frames than those having standardized dimensions are given in Table 37 and Fig. 23. Approximate net weights are given in Table 38.

**COSTS.**—Approximate costs of open, horizontal, sleeve-bearing, shunt-wound general-purpose, constant speed D.C. motors are given in Fig. 24. Adjustable speed motors have approximately the same cost as constant-speed motors, based on the continuous constant horsepower and basic speed ratings of the adjustable-speed machine. Approximate costs of other types can be determined from the following percentage increases in cost over standard constant-speed motors: Series- or compound-wound, 4%; quiet operating, 25%.

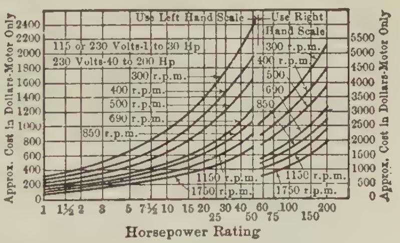


Fig. 24. Approximate 1936 Costs of General Purpose, Constant-speed, Shunt-wound, Direct-current Motors

Special mechanical features increase the cost approximately as follows: Ball bearings, 5%; sliding base, 2–5%; pulley, 1–3%; drip-proof construction, 4%; splash-proof construction, 12%; totally-enclosed fan-cooled construction, 50–100%; vertical construction, 20%.

## 7. MOTOR APPLICATIONS

The types of motors commonly used on various applications are given in Table 39. The items listed in the table are those of general application, for which conditions generally are well standardized. It is not possible to list the diverse application where motors operate under unusual conditions. Since many factors affecting the choice of motor vary with each individual installation, the recommendations given should be considered typical and not necessarily applying in every case. Manufacturers of motors should be consulted with reference to the selection of the proper motor for unusual or non-standard conditions.

**Table 39.—Types of Motors Commonly Used for Various Applications**

| Application | Power Supply | | |
|---|---|---|---|
| | Single Phase A.C. | Polyphase A.C. | Direct Current |
| | Type of Motor Commonly Used* | | |
| Air compressors, reciprocating | | | |
| 3/4 Hp. and below | Cp. | Sq.-In., Class A | Gp., Cs., Cw. |
| 1 to 10 Hp. | Rp.-In. or Cp. | " " | " " " |
| 15 to 75 Hp. | ........ | " " | " " " |
| 100 Hp. and larger | ........ | Syn. L. S. | Et., L. S. |
| Air compressors, centrifugal | ........ | Sq.-In., Class A | Gp., Cs., Cw. |
| Blowers, positive displacement, rotary | | | |
| 1 to 10 Hp. | Rp.-In. or Cp. | Sq.-In., Class A. | Gp., Cs., Sh. |
| 15 to 100 Hp. | ........ | " " | " " " |
| 125 Hp. and larger | ........ | Syn. L. S. | Et., L. S. |
| Conveyors | Rp.-In. or Cp. | Sq.-In., Class C. | Gp., Cs., Cw. |
| Cranes | ........ | Ch. Wr. | Ch. Sr. |
| Fans | | | |
| 1/4 Hp. and less | Sp. | Sq.-In., Class A | Gp., Cs., Sh. |
| 1/3 to 10 Hp. | Cp. | " " | " " " |
| 15 to 100 Hp. | ........ | " " | " " " |
| Over 100 Hp. | ........ | Syn. H. S. | " " " |
| Generators, D.C. | | | |
| 3/4 to 3 Kw. | Rp.-In. or Cp. | Sq.-In., Class A | Gp., Cs., Sh. |
| 5 to 20 Kw. | ........ | " " B | " " " |
| 25 to 40 Kw. | ........ | " " A | " " " |
| 50 Kw. and larger | ........ | Syn. H. S. | " " " |
| Hoists | Rp.-In., V | Sq.- or Wr.-In., Ch. | Sr. Ch. |
| Machine Tools | | | |
| Bolt cutters | ........ | Sq.-In., Class A, Us. or Ms. | Sh., Cs., or As. |
| Bolt heading, upsetting and forging machines | ........ | Sq.-In., Class D, or Wr. | Cw., Cs. |
| Bolt pointers | ........ | " " A | Sh., Cs. |
| Boring or turning mills | ........ | " " A, Us. or Ms. | Sh., Cs. or As. |
| Buffing machines | ........ | " " A. | Sh., Cs. |
| Bulldozers | ........ | " " B or Wr. | Cw., Cs. |
| Drilling machines | ........ | " " A, Us. or Ms. | Sh., Cs. or As. |
| Gear cutters | ........ | " " A, " " | " " " |
| Grinding wheels | ........ | " " A | " " " |
| Hammers | ........ | " " D, or Wr. | Cw., Cs. |
| Lathes | ........ | " " A, Us. or Ms. | Sh., Cs. or As. |
| Milling machines | ........ | " " A, " " | " " " |
| Nut facers | ........ | " " A | Sh., Cs. |
| Nut tappers | ........ | " " A, Us. or Ms. | Sh., Cs. or As. |
| Pipe threading or cutting-off machine | ........ | " " A, " " | " " " |
| Planers | ........ | " " A, " " | " " " |
| Polishing machines | ........ | " " A | Sh., Cs. |
| Punch presses | ........ | " " D, or Wr. | Cw., Cs. |
| Rolls, bending and straightening | ........ | Wr.-In. | Cw., Cs. |
| Saws | ........ | Sq.-In., Class A, Us. or Ms. | Sh., Cs. or As. |
| Shears | ........ | " " D, or Wr. | Cw., Cs. |
| Slotters and key-seaters | ........ | " " A, Us. or Ms. | Sh., Cs. or As. |
| Pumps, centrifugal | | | |
| 1/4 Hp. and less | Sp. | Sq.-In., Class A | Gp., Cs., Sh. |
| 1/3 to 3/4 Hp. | Cp. | " " A | " " " |
| 1 to 10 Hp. | Rp.-In. or Cp. | " " A | " " " |
| 15 to 100 Hp. | ........ | " " A | " " " |
| 125 Hp. and over | ........ | Syn., H.S. or L. S. | " " " |
| Refrigerating compressors, reciprocating | | | |
| 3/4 Hp. and less | Cp. | Sq.-In., Class C | Gp., Cs., Cw. |
| 1 to 10 Hp. | Rp.-In. or Cp. | " " C | " " " |
| 15 to 75 Hp. | ........ | " " C | " " " |
| Over 75 Hp. | ........ | Syn. L. S. | Et., L. S. |
| Refrigerating compressors, centrifugal | ........ | Sq.-In., Class A, or Wr. | Sh., Cs. or As. |
| Vacuum pumps | | | |
| 1 to 10 Hp. | Rp.-In. or Cp. | Sq.-In., Class A | Gp., Cs., Cw. |
| 15 to 75 Hp. | ........ | " " A | " " " |
| 100 Hp. and over | ........ | Syn., L. S. | Et., L. S. |
| Valves | Rp.-In. or Cp. | Sq.-In., Class D | Sr. |

* As. = adjustable speed. Ch. = crane and hoist type. Cp. = capacitor. Cs. = constant speed. Cw. = compound wound. Et. = engine type. Gp. = general purpose. H.S. = high speed. In. = induction. L.S. = low speed. Ms. = multi-speed. Rp. = repulsion. Sh. = shunt wound. Sp. = split phase. Sq. = squirrel-cage. Sr. = series wound. Syn. = synchronous. Us. = single speed. V. = reversible type. Wr. = wound rotor.

# ELECTRIC MOTOR CONTROL

## 1. GENERAL

**FUNCTIONS.**—The basic function of all motor control is to provide a means for making and breaking the connections between the motor windings and the power lines, thereby starting and stopping the motor. Practically all controllers also provide overload protection for the motor. As will be described, many other functions are provided by certain types of control.

**CLASSIFICATION.** — Motor controllers may be classified as: *a.* Full-voltage controllers; *b.* Reduced-voltage controllers. The former connect the motor directly to the power lines, that is, provide full voltage at the motor terminals during the starting period, as well as for running. The latter connect the motor to a voltage-reducing device during the starting period, and, after the motor has come up to speed, transfer the connections to the power lines. Auto-transformers commonly are used to obtain the reduced voltage in A.C. motor starters, although resistors and reactors frequently are employed. Resistors always are used for this purpose in D.C. motor starters. Full-voltage controllers are the simplest and least expensive, but reduced-voltage controllers often are required to limit the current drawn from the power lines at start.

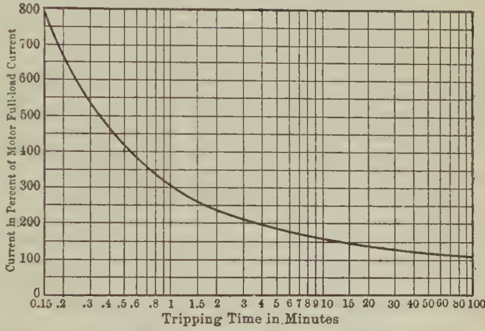


FIG. 1. Tripping Characteristics of Typical Temperature-Overload Relay

Motor controllers also may be classified according to the degree of automatic operation provided. On this basis, there are three standard types: *a.* Manual controllers; *b.* Magnetic controllers; *c.* Semi-magnetic controllers. In Type *a*, all connections to the motor are made by hand-operated switches. In Type *b* controllers, the switches are electrically operated. To start the motor it is only necessary to close a pilot circuit by means of a push button, thermostat, or similar device which may be located at some distance from the controller. Reduced-voltage magnetic controllers are provided with a relay which automatically effects the transfer from starting to running connections. In Type *c* controllers, part of the starting operation is performed by hand-operated switches and the remainder by electrically-operated devices.

**LINE SWITCHES** on low-voltage controllers (600 volts and below) usually are of the *air-break* type, *i.e.*, the circuit is broken in air. Oil-immersed switches usually are employed for the higher voltages. However, where excessive moisture or corrosive agents are present in the atmosphere, the switches, and in some cases the complete equipment, may be oil-immersed, even though controlling low voltage. Electromagnets, either solenoid or clapper type, usually are employed to close electrically-operated switches, although pilot motor-operating mechanisms sometimes are used on oil circuit breakers.

**OVERLOAD PROTECTION.—Thermal Type Overload Relays** usually are used. The most common form consists of a heater element which carries line current (or a current proportional to line current, obtained through a current transformer) mounted adjacent to a bimetallic strip. An overload on the motor causes an increase in line current, which increases the temperature of the heater element. The heat radiated to the bimetallic strip causes it to deflect and open a contact which disconnects the motor from the line. The time required for the relay to trip varies inversely with the magnitude of the overload, as does the length of time the motor can operate without injurious heating. With proper design, the relay will follow closely the heating characteristics of the motor. It thus adequately protects the motor and yet will not take it off the line unnecessarily (see Fig. 1). After the relay has tripped, it must be allowed to cool before it can be reset and the motor restarted. This also allows the motor to cool.

Another form of thermal overload relay, used on small motor starters, is the *solder-film* type. In this, the tripping mechanism normally is held to a stationary element by a thin film of solder. The heater which carries line current will, on an overload of sufficient duration, melt the solder film, thus releasing the tripping mechanism. After tripping, the solder cools and solidifies, allowing the tripping mechanism to be reset.

**Non-thermal Overload Relays** also are used. These incorporate a time delay feature, as an oil or air dashpot, to vary the tripping time with the magnitude of the overload. A disadvantage of this type is that when once tripped they can be reset immediately, and the motor started before it has had time to cool off.

**Fuses.**—Ordinary fuses do not, in general, give satisfactory overload protection. Their time lag is so short that they usually must be selected oversize to prevent tripping on the starting current of the motor, or on high overloads of short duration; as a result they do not give adequate protection against moderate overloads of long duration.

**Relay Resets.**—Most overload relays are of the "hand-reset" type; that is, after tripping they must be reset manually before the motor again can be started. In a few applications, particularly those in which the motor starts and stops automatically (for example, in response to a thermostat), it may be desirable to use an automatically reset overload relay which resets itself after a time delay of sufficient duration to permit the motor to cool. Relays of this type are available for small motors. They frequently are mounted on the motor itself and, in the case of fractional-horsepower, single-phase motors, may be designed to open directly the power circuit to the motor.

Two overload relay elements placed in different phases normally are used with 3-phase and 2-phase motors, while a single element is used with single-phase and D.C. motors.

**UNDER-VOLTAGE PROTECTION AND RELEASE.**—Most controllers are so arranged that on failure of line voltage the circuit to the motor is opened. Manual starters for small motors are an exception. If the line switch is held in the closed position by a magnet energized from the line, this feature is obtained automatically. Where the line switch is held closed by a mechanical latch, an *under-voltage device* must be provided. This is energized from the line, and, on being de-energized, operates to trip the latch. If the control is so arranged that on return of voltage the line switch recloses, *under-voltage release* is obtained. On the other hand if the line switch does not reclose on return of voltage, *under-voltage protection* is obtained. The latter is most common. The former should be used only where there is no hazard involved in the motor starting unexpectedly after a voltage failure.

In many cases, voltage failures are of very short duration, *viz.*, 1 $^1/_2$ seconds or less. Thus it is often desirable to provide control that will restart the motor if the voltage failure does not last over a certain predetermined time, but which will disconnect it permanently if the voltage failure is of longer duration. Control so arranged provides *time delay under-voltage protection*. If the line switch is held closed by a magnet energized from the line, a time delay push button station usually is used. This is so arranged that its contacts are held closed for a definite time (1 to 4 seconds) after a failure of voltage, but can be opened instantaneously by the *stop* button. When the line switch is held closed by a mechanical latch, an under-voltage tripping mechanism which incorporates a time delay in its action must be provided.

**SHORT-CIRCUIT PROTECTION.**—In general, motor controllers should not be expected to interrupt the heavy current which may flow in the event of a short circuit. Either fuses or a circuit breaker of suitable interrupting capacity should be provided for this purpose. These are usually placed in the motor branch circuit at the point where it is connected to the feeder circuit, thus serving also as a branch circuit protective device.

Controllers with high interrupting capacity contactors, or circuit breakers for line switches, are available for use where high interrupting capacity is required. An instantaneous over-current relay or trip coils should be provided, in addition to the overload relay, to insure immediate opening on short-circuit current.

**ENCLOSURES.**—Open type controllers (no enclosing case) are used where they are under competent supervision and installed where accessible only to authorized persons. In any case, all switches and devices whose voltage rating is in excess of 600 volts should be mounted on the back of the panel, or suitably enclosed to prevent accidental contact.

Controllers for small motors almost always are enclosed, and enclosures can be obtained for all sizes. A number of different types of enclosure are available. The standard *general-purpose enclosure* most commonly used is designed primarily to protect against accidental contact; it also protects against dust and light indirect splashing, but it is not dust-tight. It usually is of sheet metal construction. The *drip-tight enclosure* is similar to the general-purpose type with the addition of drip shields, or their equivalent. It is used for applications where condensation may be severe as in cooling rooms and laundries.

*Weather-resisting Enclosures* are of cast construction, or of sheet metal of suitable rigidity, with proper protection against weather hazards, as rain and sleet. They are suitable for outdoor applications as on docks, canal locks, and construction work, and also for applications in subways and tunnels. A *water-tight enclosure* is so designed that a stream of water from a hose (not less than 1 in. diameter) under a head of 35 ft. can be played on the case from a distance of 10 ft. for several minutes without leakage. It is used for outdoor applications and in dairies, breweries, and similar locations. *Submersible enclosures* are so designed that they may be submerged in water as in quarries, mines, and manholes. *Dust-tight enclosures* are provided with gaskets or their equivalent to exclude dust, and are used in such situations as often occur in steel and cement mills.

Controllers for Class I, Group D, hazardous locations (see p. 25–17) may be of the air-break or oil-immersed type. Air-break controllers depend on the strength of the enclosing case to resist the force of an internal explosion, and on long, close fits at all joints to cool the flame and prevent ignition of the surrounding atmosphere. In oil-immersed controllers, all contacts and connections are immersed in oil to prevent ignition of the surrounding atmosphere. This construction is preferable where the control is placed in a corrosive atmosphere. Controllers for Class II, Group G, conditions, and others designed to meet the requirements of the U. S. Bureau of Mines, also are available.

**Combination Switches** consist of a full-voltage magnetic starter combined with a manually-operated motor circuit switch (fused or unfused), or a circuit breaker, mounted in a single enclosing case. Thus, there are provided means for disconnecting the motor and starter, and (with fuses or circuit breaker) protection against short-circuits in the starter, motor, or wiring on the motor side of the switch. General-purpose, dust-tight and water-tight enclosing cases are available. Oil-immersed switches suitable for corrosive atmospheres, or for Class I, Group D, hazardous locations, also can be obtained. Frequently, a saving in space and in installation expense is obtained by the use of a combination magnetic switch, instead of the two separate devices. The former are available in ratings up to 25 Hp. at 220 volts, and 50 Hp. at 440 or 550 volts.

**LOCATION.**—In general, controllers should be placed within sight of the motor which they control. If this is not feasible, the controller or disconnecting device should be provided with means for locking it in the open position; or a manually operated switch, which will prevent restarting, should be placed within sight of the motor. This switch may be located in the pilot circuit of the controller.

## 2. CONTROL FOR SINGLE-PHASE MOTORS

**TYPES.**—The following types of control for single-phase motors are in common use: *a.* Manual full-voltage; *b.* Magnetic full-voltage; *c.* Manual reduced-voltage; *d.* Adjustable speed (for capacitor motors) *e.* Reversing.

**Manual Full-voltage Controllers** are available in sizes up to 1 $^1/_2$ Hp. at 110 volts, 3 Hp. at 220 volts, and 5 Hp. at 440 and 550 volts. They consist of a single- or double-pole line switch, and an overload relay, usually of the solder-film type, mounted on a compound base. They may be obtained open, or enclosed in a sheet metal case. Explosion-proof enclosures, with air-break switches, for Class I, Group D, conditions also are available. The smaller size controllers ($^3/_4$ Hp. and below) usually are arranged so they can be flush mounted, using a sectional switch box and tumbler switch flush plate. Small single-phase motors ($^3/_4$ Hp. and below) can be obtained with a tumbler switch mounted on the motor to serve as a starter. These do not provide overload protection, and should not be used where the switch may be called upon to interrupt the stalled current of the motor.

Manual, full-voltage controllers are the lowest in cost of all types and are used where full-voltage starting is acceptable, and remote control or automatic operation is not required. They do not provide under-voltage protection.

**Magnetic Full-voltage Controllers** are available in sizes up to 7 $^1/_2$ Hp. at 110 volts, 15 Hp. at 220 volts, and 25 Hp. at 440 and 550 volts. They consist of a magnetic contactor mounted on a compound base. A thermal type overload relay usually is provided in sizes above 1 Hp. at 110 volts and 1 $^1/_2$ Hp. at 220 volts. Various types of enclosures are available, including general purpose, dust-tight, water-tight, and explosion-proof for Class I, Group D conditions (both air break and oil-immersed). In the smaller sizes, enclosing cases with start-stop push buttons in the cover can be obtained. Otherwise, a separately-mounted push button station, either instantaneous or time-delay type, must be used. These are also available with general-purpose, dust-tight, water-tight, and explosion-proof enclosures. Frequently a thermostat, pressure switch or similar device is used for pilot circuit control, providing under-voltage release.

Since full-voltage starting of single-phase motors generally is acceptable, and remote or automatic control often is necessary, full-voltage magnetic starters have a wide field of application covering practically all types of drive.

**Manual Reduced-voltage Controllers** consist of a rheostat, which at start is inserted in series with the motor winding, thus limiting the current drawn from the line. As the motor comes up to speed, the operating handle of the rheostat is turned, which short-circuits the resistance step by step, until the motor is on full voltage. A magnet, whose coil is energized from the line voltage, holds the operating handle in the running position. If voltage fails, the handle is released and the motor is disconnected from the line. Overload protection ordinarily is not included. Manual reduced-voltage starters usually are provided with a general-purpose enclosing case. They are used only when the full-voltage starting current of the motor would cause objectionable voltage disturbance. Since the starting torque of the motor is reduced in proportion to the square of the current drawn, it is apparent that a reduction in the current drawn from the line will be obtained only if the load does not require the full-voltage torque of the motor to start.

**Adjustable Speed Controllers** for capacitor motors usually consist of an auto-transformer provided with reduced voltage taps, with a suitable switch for connecting the motor to any one of two or three taps, giving a corresponding number of motor speeds. A manual snap switch may be used in sizes up to 1 Hp. The magnetic type, employing contactors and a push-button station, is available in sizes up to 10 Hp.

**Reversing Controllers** usually are of the full-voltage type. Manually-operated drum or cam-switches are available for this service, in sizes up to 1 $^1/_2$ Hp. at 110 volts, 3 Hp. at 220 volts, and 5 Hp. at 440 or 550 volts. These controllers do not provide overload or undervoltage protection. Magnetic reversing switches are available in sizes up to 3 Hp. at 110 volts, 7 $^1/_2$ Hp. at 220 volts, and 10 Hp. at 440 and 550 volts. They consist of two magnetic contactors, mechanically interlocked to prevent closing of both at the same time. Both overload and under-voltage protection is provided. A 3-button push-button station is required. Enclosing cases of the general purpose type usually are provided, although water-tight and dust-tight enclosing cases also are available.

**COST.**—The approximate 1936 cost of manual, full-voltage controllers ranges from \$3 for the $^3/_4$-Hp. controller, to \$8 for the 1 $^1/_2$-Hp. controller, 110 volts, and from \$3 for the $^3/_4$-Hp. controller to \$8 for the 3-Hp. controller, 220 volts. For magnetic full-voltage type controllers, costs, which include a separately mounted start-stop push-button station, range from \$12 for the $^3/_4$-Hp. controller to \$52 for the 7 $^1/_2$-Hp. controller, 110 volts, and from \$12 for the $^3/_4$-Hp. controller to \$52 for the 15-Hp. controller, 220 volts.

## 3. CONTROL FOR POLYPHASE INDUCTION MOTORS

**TYPES.**—The following types of control for polyphase induction motors are in general use:

**Squirrel-cage Induction Motors.**—*a.* Manual full-voltage; *b.* Magnetic full-voltage; *c.* Manual reduced-voltage (auto-transformer type); *d.* Manual reduced-voltage (resistor type); *e.* Magnetic reduced-voltage (auto-transformer type); *f.* Magnetic reduced-voltage (resistor type); *g.* Multi-speed; *h.* Reversing.

**Wound-rotor Induction Motors.**—*a.* Manual primary and secondary; *b.* Magnetic primary, manual secondary; *c.* Magnetic primary and secondary.

### Squirrel-Cage Motors

**Manual Full-voltage Controllers** of the air-break type for squirrel-cage induction motors are available in sizes up to 5 Hp. at 220 volts, and 7 $^1/_2$ Hp. at 440 and 550 volts. They consist of a manually-operated switch and a 2-element, thermal type, overload relay, mounted in an enclosing case. Under-voltage protection is not provided.

Standard enclosing cases are of the general-purpose type, but explosion-proof cases for Class I, Group D conditions also are available.

Oil circuit breaker type manual full-voltage controllers are available in all sizes. These can be obtained with both overload and under-voltage protection. Except in the very large sizes, they are, in general, more costly than magnetic contactor type controllers. They are used mainly where high current-interrupting capacity is required.

**Magnetic Full-voltage Controllers** are available in all sizes. They are the most popular of all types. The common form consists of a magnetic contactor combined with a thermal type overload relay mounted in an enclosing case. (The air-break type is used for voltages of 600 and below, and the oil-immersed type for higher voltages.) A start-stop push-button station, or similar device, for closing and opening the pilot circuit also is required. Enclosing cases of the general-purpose, dust-tight, and water-tight type are available. Explosion-proof controllers for Class I, Group D, conditions are available in both the air-break and oil-immersed types. In sizes up to 5 Hp. at 220 volts, and 7 $^1/_2$ Hp. at 440 and 550 volts, enclosing cases of the general-purpose, dust-tight or water-tight, and explosion proof (air-break) type, with start-stop push buttons mounted in the cover, are available.

Some manufacturers offer oil-immersed, full-voltage magnetic controllers for high-voltage motors (2200 to 4600 volts) with an interrupting capacity of 50,000 kva. These usually are provided with instantaneous over-current trips, in addition to an overload relay, giving rapid opening on short circuit current. Oil circuit breaker controllers, with whatever interrupting capacity is required, are available in all sizes.

**Manual Reduced-voltage Auto-transformer Controllers** are available in sizes up to 150 Hp. at 220 volts, 300 Hp. at 440 and 550 volts, 800 Hp. at 2000 to 2500 volts, and 900 Hp. at 2501 to 6600 volts. They consist of an auto-transformer with two or three sets of low voltage taps and an oil-immersed double-throw manually-operated switch. In starting, the operating handle is first moved in one direction connecting the motor to a set of reduced-voltage taps of the auto-transformer and held in that position until the motor has come up to speed, after which the handle is moved quickly in the opposite direction to connect the motor directly to the line. The operating handle usually is interlocked mechanically, so that it must be moved to the start position before it can be moved to the run position. Overload and under-voltage (instantaneous or time-delay) protection are provided. General-purpose enclosing cases are standard, but dust-tight cases also are available.

The manual auto-transformer type starters are, in general, the least costly of the various reduced-voltage types. They are used whenever manual control is acceptable, and a reduction in the current drawn, or torque developed, by the motor on full-voltage is desirable.

The current drawn from the line is reduced in proportion to the square of the voltage tap used. For example, if the motor is connected to the 80% voltage tap, the current drawn from the line will be approximately 64% of the full-voltage starting current of the motor. Since the torque developed by an induction motor also varies as the square of the voltage impressed on its terminals, the ratio of starting torque to starting current (*torque efficiency*) remains essentially the same as with full-voltage starting.

**Manual Reduced-voltage Resistor Type Controllers** are available in ratings up to 15 Hp. at 220 volts, and 20 Hp. at 440 and 550 volts. They consist of a rotating drum switch and a resistor. Rotation of the former connects the motor to the line through the entire resistor, and then successively short-circuits sections of the resistor until the motor is connected directly to the line. Neither overload nor under-voltage protection are provided. These controllers are used largely for motors driving machine tools, hoists, small cranes and similar applications where manual control of the torque delivered by the motor is required.

**Magnetic, Reduced-voltage Auto-transformer Controllers** are available in all sizes. These controllers are similar to the manual auto-transformer type, except that in place of the manual switch, magnetic contactors and a relay, usually of the definite time type, are provided to effect the transfer from reduced voltage to full voltage. Overload and under-voltage protection are provided. The latter may be obtained by the use of a time-delay push-button station, or, in the case of contactors held in by a mechanical latch, a time delay under-voltage tripping mechanism may be provided on the contactor itself. General-purpose and dust-tight enclosing cases are available. Completely oil-immersed controllers, both low voltage and high voltage, are available for use in corrosive atmospheres.

**Magnetic Reduced-voltage Resistor Type Controllers** are available in all sizes. They are similar to the auto-transformer type except that a resistor, inserted in series with the motor windings at start, is used, instead of an auto-transformer, to obtain the reduction in voltage. Since the resistor is simply short-circuited as the motor comes up to speed, the circuit to the motor is not opened during the starting period. Hence this type of controller is well adapted for use where the starting current limitations are of the type which do not limit the total current which may be drawn, provided the current is built up in steps of a certain magnitude without opening the circuit between steps. Frequently, a number of accelerating contactors and relays are provided. These short-circuit the starting resistor in steps, in order to limit the current increment on each step to the required value.

Unlike auto-transformer starting, resistor starting results in a reduction in the starting torque efficiency, since the current drawn from the line is reduced in proportion to the voltage impressed on the motor, while the starting torque is reduced in proportion to the square of the voltage.

**Multi-speed Controllers.**—Controllers for multi-speed induction motors are available in both the manual and magnetic type. The former consist of a rotating drum switch, and can be obtained in sizes up to 5 Hp. at 220 volts, and $7^1/_2$ Hp. at 440 and 550 volts for two-, three-, or four-speed motors. They do not provide overload or under-voltage protection. The magnetic type is available in all sizes. They are made up of magnetic contactors, one for each speed, suitably interlocked so only one can be closed at a time. An overload relay for each speed is required to give complete overload protection. With a momentary-contact push-button station (one start button for each speed) under-voltage protection is provided. Controllers for two-speed motors can be provided with a *compelling relay*, which makes it necessary that the motor be started on the low speed only, or with a *sequence relay* which starts the motor on low speed regardless of which start button is pressed, but automatically transfers to the high speed after a definite time delay when the high-speed button is pressed. General-purpose enclosing cases normally are supplied.

A multi-speed motor also may be controlled by a manually-operated switch, which serves to make the proper connections for each speed, plus a line switch, manual or magnetic, for making and breaking the circuit to the motor. Since the former is operated only when the line switch is open, it does not have to open or close on current, and a knife-switch or inexpensive type of drum switch can be employed. Under-voltage protection may be provided in the line switch. In place of the line switch, a reduced voltage controller may be employed to reduce the starting current.

**Reversing Controllers** may be of the manual or magnetic type. The former consist of drum switches, and are available in sizes up to 10 Hp. at 220 volts, and 15 Hp. at 440 or 550 volts. They do not provide overload or under-voltage protection. The magnetic type is made up of two magnetic contactors and is available in all sizes. The contactors usually are interlocked mechanically, so that both cannot be closed at the same time. Overload and under-voltage protection normally are provided. A 3-button push-button station is required. General-purpose, water-tight, and dust-tight enclosing cases are available. Reversing control also may be provided by a manually-operated knife switch which serves to reverse the connections to the motor plus a manual or magnetic, full-voltage or reduced-voltage controller for making and breaking the circuit to the motor.

## Wound-rotor Motors

Wound-rotor induction motors require control for both the primary (stator) winding and the secondary (rotor) winding.

**The Primary Control** serves to connect the primary winding to the source of power. It is similar to squirrel-cage motor control of the full voltage type, and may be either manual or magnetic.

**The Secondary Control** serves to vary the resistance of the secondary circuit of the motor, *i.e.*, to provide a high resistance at the start, and to decrease it as the motor comes up to speed. It consists of a resistor which is connected across the slip rings of the motor and manual or magnetic switches for short-circuiting the resistor in steps as the motor comes up to speed. The resistor usually is designed to limit, to the full-load values, the current drawn and the torque developed by the motor. As the resistor is short-circuited the starting torque increases, up to the maximum torque of the motor, with a corresponding increase in current. If a portion of the resistor is allowed to remain in the circuit, the motor will operate at reduced speed. In this case the resistor must be designed for continuous duty. Before closing the primary switch, the full value of secondary resistance should be inserted in the circuit.

**Manual Primary and Secondary Control** may consist of a single drum switch with contacts for closing the primary circuit, as well as for short-circuiting the secondary resistor. A separate resistor must be provided. These drum switches are available in sizes up to 100 Hp. at 220 volts, and 200 Hp. at 440 or 550 volts. They frequently are arranged to reverse the motor as well by changing the primary connections. Manual primary and secondary control, in all sizes, may be provided by a face-plate type rheostat, or drum switch, with resistor for secondary control, combined with a manual full-voltage controller for primary control. In case an oil-circuit breaker is used for the latter it generally can be provided with a *magnetic lock*, which prevents the breaker from being closed unless the secondary controller is in the starting position, *i.e.*, all the resistance is in the circuit.

**Magnetic Primary and Manual Secondary Control** is similar to the type just described except that a magnetic controller is used for primary control. The secondary controller usually is so arranged that moving the operating handle to the first position closes the control circuit to the primary switch, causing it to close. Thus complete control of the motor is obtained at the secondary controller.

**Magnetic Primary and Secondary Control** employs magnetic contactors for the primary switch and also for short-circuiting the secondary resistor. Definite time relays, or current relays which operate in response to the drop in secondary current as the motor comes up to speed, are used to close the several secondary contactors in proper sequence. Overload and under-voltage protection normally are provided. The controllers may be obtained in the open type, or with general-purpose enclosures. Although more costly than other types, these controllers give complete control from a remotely mounted push-button. They also can be obtained with an extra contactor for automatic reversing.

**COSTS.—Squirrel-cage Induction Motors.**—Controllers for polyphase induction motors, in 1936, ranged approximately as follows:

**Full-voltage, Magnetic Type, 220 volts,** $13 for 2-Hp. controllers, $52 for 25-Hp. and 30-Hp., $232 for 60-Hp. to 100-Hp., $673 for 125-Hp. to 200-Hp. controllers; **440 volts,** $13 for 2-Hp., $52 for 30-Hp. and 50-Hp., $232 for 150-Hp., and 200-Hp., $673 for 250-Hp. to 400-Hp., $799 for 500-Hp. and 600-Hp.; **2200 volts,** $395 for 50-Hp. to 200-Hp., $500 for 250-Hp. to 700-Hp.

**Reduced-voltage, Auto-transformer Magnetic Type, 220 volts,** $155 for 5-Hp. to 15-Hp. controllers, $384 for 40-Hp. and 50-Hp., $604 for 100-Hp., $1137 for 150-Hp., $1405 for 250-Hp.; **440 volts,** $155 for 15-Hp. controllers, $390 for 60-Hp. and 100-Hp., $567 for 150-Hp., $1155 for 250-Hp.; $1416 for 400-Hp., $1633 for 500-Hp.; **2200 volts,** $739 for 50-Hp. controller, $758 for 100-Hp., $782 for 150-Hp., $795 for 200-Hp., $1353 for 300-Hp., $1423 for 500-Hp., $1575 for 800-Hp.

Costs of magnetic type controllers include a separately-mounted start-stop push-button station.

**Reduced-voltage Auto-transformer Manual Type, 220 volts,** $91 for 5-Hp. to 15-Hp. controllers, $173 for 50-Hp., $253 for 100-Hp., $694 for 150-Hp.; **440 volts,** $91 for 15-Hp., $104 for 50-Hp., $186 for 100-Hp., $202 for 150-Hp., $277 for 200-Hp., $695 for 300-Hp.; **2200 volts,** $383 for 50-Hp., $404 for 100-Hp., $428 for 200-Hp., $454 for 300-Hp., $970 for 500-Hp., $1257 for 800-Hp.

**Wound-rotor Induction Motors.**—The costs for full voltage controllers for squirrel-cage motors, given above, apply also to primary control for wound-rotor induction motors. Approximate 1936 costs of manual secondary controllers for wound-rotor induction motors were as follows: For starting duty, $96 for 5-Hp. to 10-Hp., $129 for 25-Hp., $144 for 50-Hp., $163 for 100-Hp., $262 for 200-Hp., $322 for 300-Hp., $531 for 500-Hp.; for speed regulating duty, $103 for 5-Hp., $133 for 15-Hp., $184 for 25-Hp., $236 for 50-Hp., $366 for 100-Hp., $545 for 150-Hp., $676 for 200-Hp.

## 4. CONTROL FOR DIRECT-CURRENT MOTORS

**TYPES.**—Direct-current motor controllers of the following types are in general use: *a*. Manual controllers: 1. Full-voltage, constant speed; 2. Reduced-voltage, constant speed; 3. Reduced-voltage, adjustable speed, by field control; 4. Reduced-voltage, adjustable speed, by armature control; 5. Reduced voltage, adjustable speed, by armature and field control; 6. Drum controllers. *b*. Magnetic controllers: 1. Full-voltage, constant speed; 2. Reduced-voltage, constant speed; 3. Reduced-voltage, adjustable speed, by field control.

**Manual Full-voltage Controllers** are available in ratings up to 1 Hp. at 115 and 230 volts. They consist of a single-pole or double-pole switch and a single-element thermal type overload relay, mounted in a general-purpose or explosion-proof (Class I, Group D, conditions) enclosing case. These simple and low cost controllers are applicable to small motors where full-voltage starting and manual control are acceptable.

**Manual Reduced-voltage, Constant-speed Controllers** are available in ratings up to 100 Hp. at 115 volts, 200 Hp. at 230 volts, and 400 Hp. at 550 volts. They consist essentially of a dial-type switch and a resistor. As the operating handle is turned, the switch first connects the motor armature to the line through the entire resistor, and then short-circuits the resistor in steps until the motor is on full voltage. Full voltage is applied to the field winding throughout. A retaining magnet, which holds the operating arm in the running position, releases the arm and disconnects the motor on failure of voltage. Controllers rated 550 volts usually have a magnetic contactor connected in series with the circuit to the motor. This interrupts the current when the motor is shut down, and avoids the breaking of current on the starting switch contacts. Controllers of the higher horsepower ratings generally are available with a magnetic line contactor for breaking the circuit, and a manually-operated air circuit breaker with an automatic overcurrent trip to provide overload protection. Controllers of the open type and with general-purpose enclosing cases are available.

**ADJUSTABLE SPEED CONTROLLERS. Field Control Speed Adjustment.**—Controllers are available in ratings up to 100 Hp. at 115 volts, 200 Hp. at 230 volts, and 400 Hp. at 550 volts. They are similar to the constant speed controllers described above, with the addition of a resistor, inserted in series with the field winding, and means for short-circuiting the resistor in steps, thereby varying the field current and the speed of the motor. In the smaller sizes, the dial switch has an auxiliary arm and an extra set of segments or buttons by means of which the field current is varied. In the larger sizes, a separate field rheostat is provided. The control is so arranged that all of the resistance in the field circuit must be short-circuited before the motor can be started.

**Armature Control Speed Adjustment.**—Controllers are available in ratings up to 50 Hp. at 115 volts, 60 Hp. at 230 volts, and 5 Hp. at 550 volts. They are similar to the constant speed controllers, except that they have a continuously rated armature resistor, and the operating handle of the dial switch can be left in various positions to leave a certain portion of the resistor in the circuit. Thus the voltage impressed on the armature of the motor can be varied, with consequent variation in speed. Speed reduction down to 50% of full speed is standard. With this type of control the speed of the motor varies considerably with the load. It cannot be used where close speed regulation is required

**Armature and Field Control Speed Adjustment.**—Controllers are available in ratings up to 10 Hp. at 115 volts, 20 Hp. at 230 volts, and 5 Hp. at 550 volts. They consist of a combination of the two types described above. A variable armature resistance, sufficient to give 50% speed reduction, and a variable field resistance to give 25% speed increase, are standard.

**Manual Drum Controllers** of the full-voltage type are available in ratings up to 5 Hp. at 115 and 230 volts. The reduced-voltage type can be obtained in ratings up to 40 Hp. at 115 volts, 75 Hp. at 230 volts, and 100 Hp. at 550 volts. Both types can be obtained for reversing as well as for non-reversing duty. The reduced-voltage type also can be obtained suitable for adjustable speed operation either by field or by armature control. Where field control is provided, the field resistor usually is mounted on the side of the controller, and the drum switch has segments for short circuiting the resistor in steps as the operating handle is turned. The armature resistor usually is mounted separately.

Drum controllers arranged to provide *dynamic braking* also can be obtained. The motor is connected as a generator and a resistor is connected across the armature so that when driven by an overhauling load, power is delivered to the resistor which holds back the load.

Since drum controllers do not provide overload or under-voltage protection, a *protective panel*, consisting of a magnetic contactor and a temperature overload relay mounted in an enclosing case, often is used. The contactor will open on failure of voltage, disconnecting the motor, after which the drum controller must be returned to the off position to reclose the contactor. In the event of injurious overload, the relay will open the contactor. An open-close push-button station is recommended. This may be located at any convenient place, and provides an additional means for stopping the motor. Drum controllers are used with motors driving machine tools, cranes, hoists and similar applications, where a convenient and compact manual controller is required.

**Magnetic Full-voltage Controllers** are available in ratings up to 1/2 Hp. at 115 volts and 3/4 Hp. at 230 volts. They consist of a magnetic contactor mounted in a general-purpose enclosing case. A start-stop push-button station or similar device is required for starting and stopping the motor.

**Magnetic, Reduced-voltage, Constant-speed Controllers** are available in all ratings. They consist of a starting resistor, two or more magnetic contactors, and relays, usually of the definite time type, for closing the contactors in proper sequence to short-circuit the resistor as the motor comes up to speed. In some types, accelerating contactors, incorporating a time-delay feature, are used. These eliminate the necessity for a separate relay. A single-element temperature-overload relay and a general-purpose enclosing case usually are provided. Controllers arranged for dynamic braking with or without reversing features also are available.

**Magnetic, Reduced-voltage, Adjustable-speed Controllers,** with speed adjustment by *field control* are available in all sizes. They are similar to the type above described, with the addition of a field rheostat for varying the field current, and hence the speed, of the motor. A *field accelerating relay* often is added, which prevents high armature current when the motor accelerates rapidly from basic speed (full-field) to a higher speed (weak field). A "field protective" relay may be installed to protect the motor from excessive speed if the field circuit is opened accidentally.

**COSTS** of the more generally used controllers for D.C. motors ranged, in 1936, approximately as follows:

**Manual Controllers, Constant-speed, 115 volts,** $12 for 1-Hp. controllers, $18 for 3-Hp., $22 for 5-Hp., $37 for 7 1/2-Hp., $46 for 10-Hp., $65 for 20-Hp., $85 for 30-Hp., $249 for 50-Hp., $422 for 100-Hp.; **230 volts**, $12 for 1-Hp., $18 for 3-Hp., $22 for 5-Hp., $23 for 7 1/2-Hp., $41 for 10-Hp., $48 for 20-Hp., $65 for 30-Hp., $90 for 50-Hp., $256 for 100-Hp., $266 for 150-Hp., $422 for 200-Hp.; **550 volts,** $22 for 1-Hp. and 3-Hp., $23 for 5-Hp., $28 for 7 1/2-Hp., $72 for 10-Hp., $80 for 20-Hp., $158 for 30-Hp., $164 for 50-Hp., $200 for 100-Hp., $214 for 150-Hp., $323 for 200-Hp., $358 for 300-Hp., $469 for 400-Hp.

**Manual Adjustable-speed Controllers, Field Control.—115 volts,** $36 for 1-Hp., $38 for 3-Hp. and 5-Hp., $46 for 7 1/2-Hp., $77 for 10-Hp., $120 for 20-Hp., $129 for 30-Hp.,

$405 for 50-Hp., $587 for 100-Hp.; **230 volts**, $36 for 1-Hp., $38 for 3-Hp. and 5-Hp., $46 for 7 $^1/_2$-Hp., $77 for 10-Hp and 20-Hp., $120 for 30-Hp., $136 for 50-Hp., $416 for 100-Hp., $428 for 150-Hp., $587 for 200-Hp.; **550 volts**, $317 for 30-Hp. and 50-Hp., $364 for 10-Hp., $405 for 150-Hp., $440 for 300-Hp., $645 for 400-Hp.

**Manual Adjustable-speed Controllers, Armature Control.—115 volts,** $29 for 1-Hp., $47 for 3-Hp., $81 for 5-Hp., $96 for 7 $^1/_2$-Hp., $99 for 10-Hp., $214 for 20-Hp., $319 for 30-Hp., $473 for 50-Hp.; **230 volts,** $29 for 1-Hp., $47 for 3-Hp., $75 for 5-Hp., $81 for 7 $^1/_2$-Hp., $86 for 10-Hp., $144 for 20-Hp., $280 for 30-Hp., $373 for 50-Hp.

**Magnetic Constant-speed Controllers.—115 volts,** $41 for 1-Hp. and 3-Hp., $64 for 5-Hp. and 7 $^1/_2$ Hp., $70 for 10-Hp., $229 for 20-Hp., $364 for 30-Hp., $517 for 50-Hp., $529 for 75-Hp.; **230 volts,** $41 for 1-Hp. and 3-Hp., $47 for 5-Hp. and 7 $^1/_2$-Hp., $64 for 10-Hp., $70 for 20-Hp., $211 for 30-Hp., $375 for 50-Hp., $517 for 100-Hp., $529 for 150-Hp.; **550 volts,** $70 for 1-Hp. and 3-Hp., $76 for 5-Hp. and 7 $^1/_2$-Hp., $105 for 10-Hp., $111 for 20-Hp., $211 for 30-Hp., $252 for 50-Hp., $375 for 100-Hp., $387 for 150-Hp.

**Magnetic Adjustable-speed Controllers, Field Control.—115 volts,** $64 for 1-Hp. and 3-Hp., $88 for 5-Hp. and 7 $^1/_2$-Hp., $94 for 10-Hp., $317 for 20-Hp., $440 for 30-Hp., $739 for 50-Hp., $751 for 75-Hp.; **230 volts,** $64 for 1-Hp. and 3-Hp., $70 for 5-Hp. and 7 $^1/_2$-Hp., $88 for 10-Hp., $94 for 20-Hp., $305 for 30-Hp., $451 for 50-Hp., $745 for 100-Hp., $757 for 150-Hp.; **550 volts,** $99 for 1-Hp. and 3-Hp., $117 for 5-Hp. and 7 $^1/_2$-Hp., $123 for 10-Hp., $129 for 20-Hp., $305 for 30-Hp., $340 for 50-Hp., $451 for 100-Hp., $463 for 150-Hp.

Costs of magnetic controllers include a start-stop push-button station for separate mounting.

## 5. CONTROL FOR SYNCHRONOUS MOTORS

The control for the armature (primary) winding of a synchronous motor is similar to that provided for a squirrel-cage induction motor. See p. 25–54. It may be of the full-voltage or reduced-voltage type, either manual or magnetic. In addition, control for the field winding must be provided. This control usually is of the magnetic type, so that after the primary control has connected the armature winding to the A.C. power lines, direct current automatically is applied to the field winding when the motor has accelerated to the proper speed, *i.e.*, 95 to 98% of synchronous speed. When the magnetic field control is combined with magnetic primary control, the complete controller is designated *magnetic* and when combined with manual primary control it is designated *semi-magnetic.*

A field contactor and *field application relay* are required to apply field excitation automatically. The latter may be of the definite time type, or it may be a slip-frequency relay. Such a relay operates on the change in frequency of the alternating current induced in the field winding as the motor accelerates. In addition, a *discharge resistor* must be connected across the field winding during the starting period in order to limit the voltage induced in that winding. The discharge resistor circuit is opened when field excitation is applied, either by normally closed contacts on the field contactor or by a separate field discharge contactor actuated by the field application relay. Synchronous motor controllers also include a *field removal relay* which operates automatically to open the field contactor and remove excitation, whenever the motor pulls out of synchronism. The latter condition may occur as a result of a high overload or a drop in line voltage. By removing field excitation, after the load or voltage returns to normal, the motor may be able to re-accelerate, and the field application relay will re-apply field so that normal operation is resumed automatically. If field excitation is not removed on pull-out the motor is apt to stall, and, in addition, is subjected to violent torque and current surges. Where the driven machine must be unloaded when starting, in order that the motor will accelerate and synchronize, it often is possible to provide an automatic, electrically-operated unloader, which can be connected to the field contactor. Then on pull-out, the automatic removal of field also will unload the machine, so that the motor can re-synchronize, after which load automatically is re-applied. The field removal relay may operate on the abrupt drop in the power-factor of the current drawn by the motor as it pulls out of step, or on the increase in line current under this condition, or on the current induced in the field winding as the motor speed drops below synchronous speed.

**Means for Adjusting D.C. Excitation** applied to a synchronous motor also should be provided. Where an exciter is used to supply excitation to a single synchronous motor, adequate control of the excitation can be obtained by means of the exciter field rheostat. However, if a single D.C. generator supplies excitation to two or more synchronous motors, a field rheostat must be connected in series with the field winding of each motor. Synchronous motor controllers usually are arranged for mounting either an exciter field rheostat, a motor field rheostat, or both.

Usual practice provides synchronous motor controllers with two ammeters, an A.C. ammeter for reading line current, and a D.C. ammeter for reading field current. Additional instruments, such as an A.C. voltmeter, indicating wattmeter, or watt-hour meter, are often valuable additions.

Like induction motor controllers, synchronous motor controllers normally are provided with a thermal type *overload relay*. Time-delay *under-voltage protection* is standard for controllers rated 2200 volts and above, while for controllers of lower voltage rating, instantaneous under-voltage protection is standard, but the time-delay type can be obtained. Standard controllers are of the open type, but general-purpose enclosing cases are available.

**TYPES.**—Synchronous motor controllers of the following types are commonly used: *a*. Magnetic, full-voltage; *b*. Semi-magnetic, full-voltage; *c*. Magnetic, reduced-voltage, auto-transformer type; *d*. Semi-magnetic, reduced-voltage, auto-transformer type; *e*, Magnetic, reduced-voltage, resistor type; *f*. Magnetic part-winding.

Types *a*, *c*, and *e* are similar to the corresponding types of magnetic squirrel-cage induction motor controllers, with the addition of the automatic field control and ammeters described above. Types *b* and *d* are similar to the corresponding type of manual squirrel-cage induction motor controllers with the same additional features.

**Magnetic Part-winding Controllers** are arranged to start a motor by energizing only a portion of the complete armature winding and then, after the motor has accelerated, energizing the remainder of the winding in one or more steps. The motor must be designed especially for this type of starting, with a *multiple circuit* armature winding and sufficient terminal leads provided to allow energization of the various circuits independently. The effect of this starting method is to reduce the current drawn from the line and the torque developed by the motor at start, to give the effect of reduced-voltage starting without the use of an auto-transformer, resistor, or other voltage reducing device. The amount of reduction in torque and current depends on the percentage of the winding energized, and also on the characteristics of the individual motor. With some motors the torque is reduced in proportion to the reduction in current, while in others a considerably greater reduction in torque occurs. Like reduced-voltage resistor starting, part-winding starting has the advantage that the circuit to the motor is not opened during the starting period, making the method particularly applicable where the starting current limitations are of the increment type. Motors of relatively high voltage in relation to their horsepower cannot be economically designed for part-winding starting. The method is most applicable to low-speed machines of large horsepower and moderate voltage. Most motors and controllers for part-winding starting are arranged to give just one reduced kva. step (two-point starting), but in many cases they can be provided with two, three, or more reduced kva. steps.

**COSTS** of controllers, open type, for synchronous motors, in 1936, ranged approximately as follows:

**Full-voltage Magnetic Type, Unity Power Factor: 220 volts,** $393 for 50-Hp. and 60-Hp. controllers, $514 for 100-Hp. and 125-Hp., $647 for 150-Hp., $750 for 200-Hp.; **440 volts,** $303 for 50-Hp. and 60-Hp. controller, $393 for 100-Hp., $430 for 150-Hp., $514 for 200- and 250-Hp., $647 for 300-Hp., $750 for 400-Hp. and 500 Hp.; **2200 volts,** $647 for 50-Hp. to 350-Hp. controllers, $750 for 400-Hp. to 500-Hp.

**Reduced-voltage, Auto-transformer, Magnetic Type, Unity Power Factor: 220 volts,** $708 for 50-Hp., $998 for 100-Hp., $1513 for 150-Hp., $1694 for 200-Hp.; **440 volts,** $563 for 50-Hp., $702 for 100-Hp., $871 for 150-Hp., $1035 for 200-Hp., $1548 for 300-Hp., $1755 for 400-Hp. to 500-Hp.; **2200 volts,** $1089 for 50-Hp. controller, $1101 for 60-Hp. to 100-Hp., $1131 for 150-Hp., $1143 for 200-Hp., $1258 for 300-Hp., $1621 for 400-Hp. to 500-Hp.

The costs of magnetic type controllers include a start-stop push-button station for separate mounting.

**Full-voltage Semi-magnetic Type, Unity Power Factor: 440 volts,** $532 for 225-Hp. to 350-Hp.; **2200 volts,** $730 for 50-Hp. to 250-Hp., $742 for 300-Hp. to 500-Hp.

**Reduced-voltage, Auto-transformers, Semi-magnetic Type, Unity Power Factor: 220 volts,** $424 for 50- and 60-Hp., $490 for 100-Hp., $1071 for 150-Hp., $1131 for 175-Hp.; **440 volts,** $351 for 50-Hp., $430 for 100-Hp., $448 for 150-Hp., $502 for 200-Hp., $1131 for 300-Hp.; **2200 volts,** $659 for 50-Hp. and 60-Hp., $678 for 100-Hp., $690 for 150-Hp. to 200-Hp., $702 for 250-Hp. to 350-Hp., $1186 for 400-Hp., $1204 for 500-Hp.

# Section 26

# MISCELLANEOUS SHOP EQUIPMENT

## DUST COLLECTION

By M. S. Kice, Jr.

## ILLUMINATION

By Matthew Luckiesh

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# MISCELLANEOUS SHOP EQUIPMENT

## DUST COLLECTION *

By M. S. Kice, Jr.

### 1. CHARACTERISTICS OF DUST

There is no standard definition of industrial dust. Generally it comprises powdered minerals, fine metals, organic dust, chips and shavings. In the following discussion, all of the more or less finely divided particles will be treated as dust.

**COMMON TYPES OF DUST.**—1. Wood sawdust and shavings from planing mill equipment. 2. Metallic and abrasive grindings and lint from metal grinding and polishing machinery. 3. Stone-working dust. 4. Foundry dust. 5. Various types of ore dust at varying temperatures. 6. All pulverized materials such as dried and powdered milk, cement, talcum powder, pulverized coal, etc. 7. Cinder from stoker-fired boilers. 8. Fly ash from pulverized fuel boilers.

**NATURE OF DUST.**—Combinations of various properties of dust are factors affecting the choice of the type of suitable dust collector, as well as its collecting efficiency. The recognized method of proving collector efficiency is the testing of actual samples of a specific dust. Combinations of the many variables involved in dust collecting work would lead to speculative results if the subject were treated mathematically.

**Shape.**—Dust particles formed at high temperature often are hollow or globular. Mineral particles from substances at ordinary temperature usually are solid. Most organic dusts are fuzzy. Rarely is the shape uniform.

**Fineness.**—Metallic and wood particles are coarse. Organic powders, such as flour, are extremely fine. Mineral dusts in industrial use are graded through standard mesh screens. Finer dusts are measured by calibrated microscopic photography and by flotation. The finest mesh screen available has 400 meshes to the inch. The square power of a mesh number represents the number of mesh openings per square inch. Thus, a 400-mesh screen has 160,000 openings per square inch. Each net opening is $2.03 \times 10^{-6}$ sq. in. The wires forming this mesh are $1.075 \times 10^{-3}$ in. diameter. For comparison, the diameter of a No. 40 B. & S. bare wire is $3.441 \times 10^{-3}$ in. Fine dust is measured in microns, which is a linear measure equal to $3.93 \times 10^{-5}$ in.; 1300 particles one micron diameter can be assembled in one plane of one mesh opening of a 400-mesh screen. Table 1 illustrates proportions of common screens.

**Table 1.—Properties of Common Screens**

| Screen mesh | 20 | 40 | 60 | 80 | 100 | 150 | 200 | 325 | 400 |
|---|---|---|---|---|---|---|---|---|---|
| Width of mesh opening, in. | 0.0340 | .0150 | .0087 | .0068 | .0058 | .0041 | .0029 | .0017 | .001425 |
| Approx. microns across net opening | 850 | 375 | 217 | 170 | 147 | 102 | 74 | 43 | 36 |
| Proportional opening, in.* | 240 | 105 | 61 | 48 | 41 | 29 | 20 | 12 | 10 |

* These values represent an equivalent approximate size sphere to the same scale as represented by one micron being enlarged to the size of a circle inscribed in the hexagon of an engineer's common lead pencil, namely, 0.28 in. These values are given to allow visualization of these proportions.

**Uniformness.**—Rarely is dust uniform. Its fineness has to be described in percentage values of that dust that will pass through successive mesh screens. As an illustration of uniformness grading, the following represents the mesh analysis of common Portland cement.

| Mesh | 40 | 60 | 100 | 150 | 200 | 325 |
|---|---|---|---|---|---|---|
| Percentage passing through | 99.9 | 99.8 | 98.5 | 95.0 | 83.0 | 61.0 |

For making this screen test, circular frames about 1 1/2 in. high and 8 in. diam., con-

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*Acknowledgments.—E. J. Tangerman, Managing Editor, *Power*. H. O. Danz, Manager Dust Collector Division, American Blower Corporation. W. O. Vedder, Pangborn Corporation.

taining a stack of screens in the above order and mounted in a vibrating machine are used. A weighed sample is placed on the top screen. After screening, the material on each screen and that which passed through the last screen is weighed. From these values the percentage through each mesh is calculated.

**Specific Gravity** is that of the dust, not of the solid from which the dust is produced.

**Concentration** relates to grains per cubic foot of air. It is a measure of the amount of material possible to extract from a given quantity of air.

**Toxicity.**—In addition to being careful to remove poisonous dusts from an atmosphere, care must be taken to protect those testing such dust samples, and those handling the material collected in the field.

**Adhesiveness.**—Sticky dusts cannot be caught in dust collectors of any type unless provision is made for periodic cleaning and scraping.

**Explosiveness.**—Fine inflammable dusts are extremely hazardous. The air stream must be kept below the flash temperature of the mixture, and provision assured to guard against spark or other ignition. All metallic parts must be grounded. The rotating parts of fans handling such dust should be made of spark-proof material.

**Electrostatic Charge.**—Charges of static electricity that tend to agglomerate dust particles allow a greater collection efficiency of the fine particles because of their adherence to the larger particles.

**GROUPS OF DUST.**—1. Dust hazardous to health, agriculture and machinery. 2. General nuisance dusts, as fly ash. 3. Dusts useful as by-products, products and fuels, as asphalt topping, plaster of Paris and pulverized coal. 4. Valuable dusts, as gold grindings, etc.

## 2. TYPES OF DUST COLLECTORS

**SETTLING CHAMBERS** are the oldest form of dust separators. A large dead air space receives dust-laden air and contains it long enough for the dust to settle. A modified form consists of a pocket trap in an ordinary duct to drop out heavy particles accidentally or purposely admitted to the air stream.

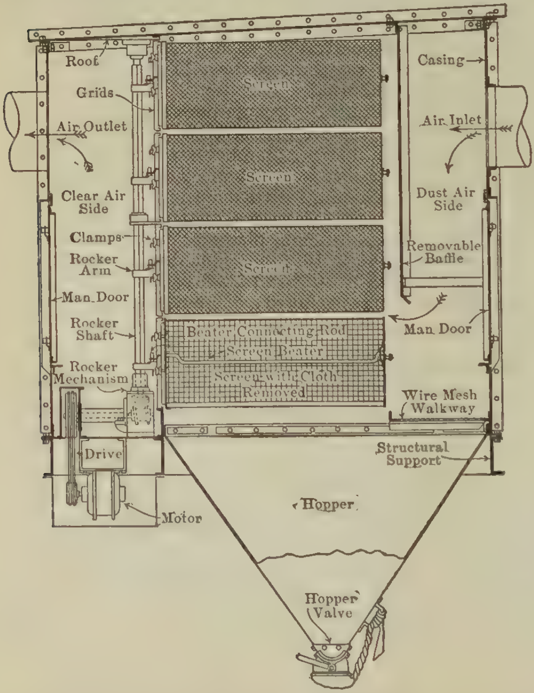

Fig. 1. Section of Cloth Screen Filter

**FILTERS** are a type of collector that appear in various forms. The modern adaption consists of a viscous covered channel, forming tortuous paths for dust-laden air. The dust impinges on the viscous surface. When fully loaded some types are cleaned in an oil bath. Other types are discarded and replaced with new filter elements.

**Bag-type Air Filters** consist of cotton or woolen socks or bags arranged in battery, suspended vertically from rings at the top and secured at the bottom around thimbles which protrude from the floor. Gases enter from a lower plenum. The bags expand as they are under pressure. The bag house is divided into several compartments, to permit one compartment to be isolated from the system and cleaned. This is accomplished by a mechanical shaking system which dislodges the arrested material on the interior of the bag, which material falls to the plenum chamber below and is extracted. A velocity of about 2.5 ft. per min. is figured through the bag surface. Resistance to air flow varies about as the 1.7th power in terms of air velocity, and increases considerably as the bag material becomes loaded with dust. This resistance at constant velocity varies from about 0.5 in. to 3 in. static water gage. Commercially, bag-type filters are used for fumes and gases from smelter furnaces of various kinds for separation of finely divided ores and metals. The gases are cooled to about 180° F.

**Cloth Screen Filter.**—A modern modification of the bag-type filter consists of flat cloth fabric screens applied over a series of rectangular frames and enclosed in a sheet-metal box framework. The cloth covers both sides and three edges of the frame. The fourth edge, left open, forms the flue for the clean air outlet. Hoppers are provided below to catch the dust. The cloth screen frames are connected to a rapping device to dislodge dust packed on the cloth. Coarser dust is settled out prior to encountering the screens, and in some designs a settling chamber is provided ahead of the collector. Fig. 1 shows a typical cross-section of a cloth screen filter.

**Intermittent Type Filters** allow the cloth to be cleaned by periodic shaking, the exhaust fan being shut down. The cleaning frequency is determined by specific conditions. It averages 5 to 10 min. in each 4 to 5 hr. operation. The advantage of the intermittent type is lower first cost. It is suitable where dust concentration is not severe, or where the cleanout time is not disadvantageous.

**Continuous Type Filters** are used for heavy dust concentration or where shut-down periods are prohibitive. They consist of two intermittent types, cross-connected to allow one collector for service operation, while the other is being cleaned. The switch-over can be arranged for manual operation with cross-connecting dampers, or the entire period can be controlled automatically.

**ELECTROSTATIC PRECIPITATORS.**—High voltage direct current fed to electrodes in series gives the suspended dust particles a negative charge, causing them to be attracted by the positively charged electrodes to which the dust adheres. When the current is interrupted and the plates rapped, the dust is precipitated to the hoppers below. Lighter dusts are removed by scraping. The composite plate precipitator has two or more separate banks of electrodes in series, each up to 15 parallel ducts formed by large concrete slabs which act as grounded collecting electrodes. Discharge electrodes hang centrally in the ducts. The curtain rod precipitator is similar, except that collecting electrodes are pipe frameworks into which are threaded many vertical parallel rods. These curtains, hung from the top, can be lifted and dropped against anvils at the bottom, thereby vibrating collected dust loose. Rapping is done continuously or at necessary intervals by a solenoid-actuated mechanism. The vertical pocket electrode precipitator allows gas to enter the hopper and rise through ducts to the breeching. Ducts are formed by parallel electrodes built up of six individual sections or hung from the top supporting member. Pockets in the sides of these boxes provide louvre-like openings which catch the dust when the section is vibrated, dropping it down through the center to the hopper out of the incoming gas path. Fig. 2 shows the composite plate type.

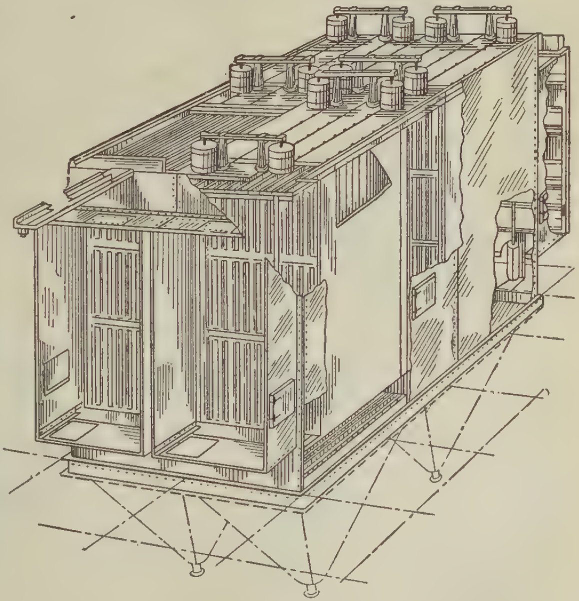

Fig. 2. Composite Plate Type Electrostatic Precipitator

**WET WASHERS.**—The simplest form of gas washer is the common air washer. See Kent's Mechanical Engineers' Handbook—Power, p. 11–52, forming Vol. 2 of this series.

It is efficient for light concentrations of dust soluble or wettable in water. For heavier dust concentrations, the lower tank is sloped and fresh water is used. Particles that will not wet do not puncture the water mist particle. Finely divided, the water surface tension prevents puncturing the water globule. Such dust passes through a common washer. Chemical treatment is possible to reduce the water surface tension. For cleaning products of combustion, the gas scrubber usually consists of plates, rods, links, etc., suspended in a gas stream and sprayed with water. Dust is washed down through the hopper. Sinuous paths through the scrubber increase cleaning efficiency. Materials not attacked by sulphurous and sulphuric acid must be used. The latter acid is formed by the combination of sulphur gases and the spray water.

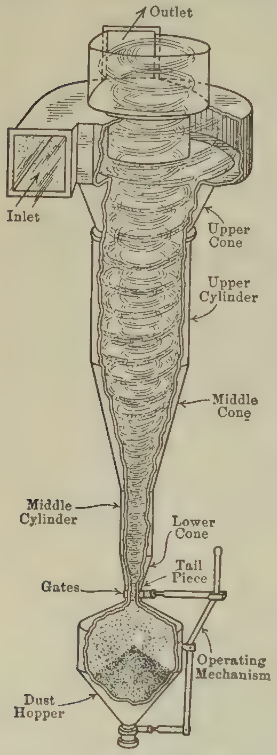

FIG. 3. Action of Vortex Collector and Receptacle

**CINDER TRAPS.**—Rows of diamond and other shaped sections, arranged staggered, are constructed with narrow slots on the side of the entering air. Heavier particles of dust, moving in a straight line, possess too much inertia to work around the staggered rows of baffles, and are trapped inside the hollow section and pass to a hopper below. The air sinuously moves around the staggered rows. Light dust particles move with the air stream.

**CENTRIFUGAL COLLECTORS.**—The same principle of nature that is used in cinder traps is utilized in a centrifugal collector. The dust-laden gas is rotated. The inertia of solid particles prevents them from accelerating toward the center. They strike the outer surface and either are skimmed off or drop down to a chamber below. The efficiency of the collector is directly proportional to the rotation of the air, which, in turn, is proportional to the pressure drop of the collector, and, again in turn, is proportional to the power put into the air stream.

**The Common Cyclone** collector usually consists of a conical lower half and an upper half, which either is likewise conical in the opposite direction or cylindrical. Dust-laden gas enters tangentially in the upper body. The air is rotated inside of the cyclone, and the cleaner air passes out at the top of the center of the whirl. This type is universally used for the collection of shavings, sawdust, etc., from woodworking establishments. Its pressure drop ranges from $^1/_2$ in. to 1 in. water gage.

**The Vortex Collector** is shown in Fig. 3. Dust-laden gas enters a scroll body, which rotates the air stream. Below, and extending downward, are a series of cones and cylinders. The decreasing cross-sectional radius spins the air stream into an extremely high velocity vortex. Dust particles impinge against the outer walls and work downward, passing out of the lower cylinder. A collar extends into the body of the vortex collector to skim off the clean air, which is discharged either through a vertical pipe or an outlet head; the latter is used when it is desired to carry the clean gas in the same plane as the entering gas. The dust outlets must be sealed, either by a rotary dust valve or a lower receptacle. Internal turbulence is eliminated, which prevents the fine particles re-entering the clean air stream. This type is used primarily in industrial process work of furnaces, roasters, calciners, crushers, grinders, etc., and in connection with spray dryers or towers for milk, soap or food products.

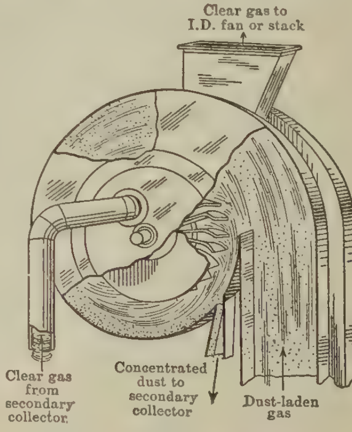
Clear gas to I.D. fan or stack
Clear gas from secondary collector
Concentrated dust to secondary collector
Dust-laden gas

FIG. 4. Spiral Scroll Collector

The pressure drop across the vortex collector depends on the air velocity and the size of the internal collar. The performance characteristics of this type afford unusually high collection efficiency for coarse and fine particles. The efficiency remains practically constant for a wide variation in dust concentration and air velocity.

**Spiral Scroll Collectors** ordinarily are used as a spiral inlet box on a single-inlet fan, or spiral inlet boxes on each side of a double-inlet fan. Fig. 4 shows a cross-section of this type. Dust-laden gases pass around the spiral, and the dust is forced to the outer wall. A scoop opening admits part of the gas to a cyclone collector for final separation. The clean gases pass back into the spiral scroll collector. The pressure drop across the spiral scroll collector operates the vortex collector without the use of additional fan equipment. The primary fan handles the full volume of the air. At the collector outlet, straightening vanes admit the cleaned air into the fan wheel. These vanes prevent internal turbulence in bleeding off clean air. The pressure drop is from 1 1/2 in. to 2 1/2 in. water gage.

**The Spiral Tube Precipitator** consists of a bank of parallel tubes. On the entering side of the dusty air, a streamline spiral rotates the air. The relatively small diameter of each rotating air stream eliminates the necessity for high-velocity rotation, as the dust particle has a long path to travel longitudinally and a short distance to travel axially. In the clean air end, a hollow tube protrudes into the outer tube to receive the clean center of the air stream, the dust passing through a circular orifice into a collecting chamber. From 5% to 10% of the primary air is bled off into a vortex collector for final separation, and the clean air is returned to the entrance of the spiral tube precipitator by means of a secondary fan. The collecting efficiency of this type of precipitator is extremely high for very fine particles. Its most popular application is the collecting of fly ash from boilers fired with pulverized fuel. Fig. 5 is a phantom cross-section of the spiral tube precipitator.

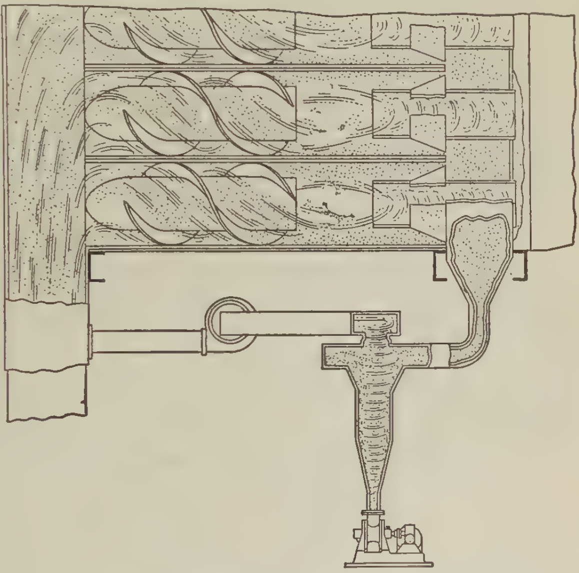

FIG. 5. Spiral Tube Precipitator Elements

## 3. DUST COLLECTING SYSTEMS

**FACTORS INFLUENCING REQUIREMENTS.**—1. Improvement of working conditions. 2. Nuisance elimination. 3. Value of recovered material. 4. Public liability. 5. State laws.

**FACTORS INFLUENCING CHOICE OF TYPE OF SYSTEM.**—1. Relative capital, power and maintenance costs. 2. Nature of dust involved. 3. Dust concentration. 4. Efficiency requirement of dust collection.

**TYPES OF SYSTEMS.**—The best known type of dust collector system is that involving vacuum hoods connected by branches to a main pipe, served by a fan which discharges to a collector.

**The Planing Mill Exhaust System** is an example of the branch pipe and vacuum hood system. Fig. 6 shows a typical arrangement, and Fig. 7 shows typical hoods. In addition to the branches for various hoods, a number of floor sweeps must be supplied and provided with shut-off gates. Ordinarily the size of main is established independently of the number of floor sweeps, as they are used intermittently. Table 2 shows size of hood connections for average woodworking machinery.

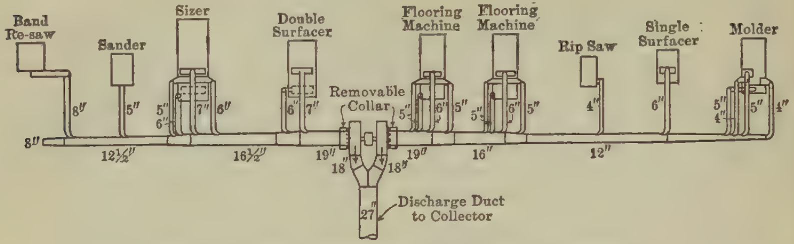

FIG. 6. Plan of Typical Planing Mill Exhaust System

**HOOD DESIGN.**—Requirements of good hood design are: 1. Minimum interference with machine operation. 2. Air to enter hood orifice nearly as possible in the same direction as particles are discharged from machine. 3. Close proximity of hood mouth to source of dust production source. 4. Removable sections in hood for clean-out, inspection and maintenance. 5. Desirable that machine builders incorporate hoods in the original design of the machine. 6. Sufficient entering air velocity into the hood to arrest and carry away generated dust particles.

**PIPING DESIGN.**—Points leading to good piping design are: Conveying-air velocity to be sufficiently high to carry entrained dust. 2. Piping to run in as direct a line as possible from hoods to collector or fan. 3. Non-interference with cranes, machinery and other necessary plant equipment. 4. Bottom of pipe should be on a straight line. The taper should be taken on the sides and top. 5. Branch angles preferably 30 deg. included angle between the branch and the main pipe. 6. Pipe joints lapped in direction of air flow, and soldered or welded. 7. Clean-outs to extend into the lower side of the duct and to be equipped with removable air-tight caps. 8. Floor sweeps provided where required. 9. Large-radius branch-pipe bends into the main pipe. 10. Blast gates to equalize suction. 11. Provision for future additional taps.

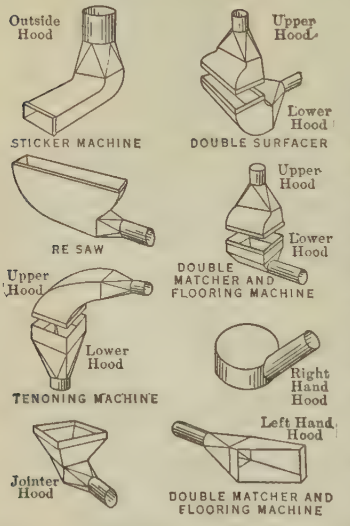

FIG. 7. Typical Hoods for Woodworking Machines

**COMPUTATION OF FAN DUTY.**—The cubic feet per minute (c.f.m.) moved by the exhaust fan is the product of the main duct area in square feet beyond the last branch to the main, and the air velocity in the duct at that point, in feet per minute. The static pressure is the sum of the duct resistance of the longest air path from the farthest machine to the inlet of the collector, plus collector loss, plus suction at the hood. See Kent's Mechanical Engineers' Handbook—Power, p. 1–81, forming Vol. 2 of this series, for duct resistance chart. This static pressure runs from 4 to 8 in. water gage, averaging about 5 1/2 in. The fan brake horsepower, read from the fan capacity table, should be increased 15–25% to allow for the extra weight of material in the air stream which the fan has to move. For data on fan capacity testing, see Kent's Mechanical Engineers' Handbook—Power, p. 1–55, forming Vol. 2 of this series.

**THE MAIN PIPE SIZE** is arrived at by several methods. One method is to add the area of the branches, increasing the main size to the fan outlet, which is 15% to 30% greater than the duct. Another common method is to add the areas of the adjoining pipes and increase the main beyond this junction by 10%; in other words, slowing the velocity down gradually toward the fan. In this case, the last main diameter is the diameter of the fan inlet. A third method consists of using a constant friction per 100 ft. of duct. For details of this method see chart, Kent's Mechanical Engineers' Hand-

**Table 2.—Sizes of Hood Connections for Woodworking Machinery**

| Type of Machine | Diameter of Pipe, in. | Type of Machine | Diameter of Pipe, in. |
|---|---|---|---|
| Matcher heads, each | 5 | Rip saws and resaws | |
| Sash and cabinet shaper, each head | 4 | Up to 14 in. diam | 4 |
| Door tenoner | 5 | 14 to 18 in. diam | 4 1/2 |
| Sash tenoner | 5 | 18 to 24 in. diam | 5 |
| Door and ash sticker, each head | 5 | Cut-off saws not over 16 in. diam | 4 |
| Blind slat sticker | 5 | Band saws | |
| Blind rail router | 4 | Up to 1 1/2 in. wide | 4 |
| Panel raiser, each head | 4 | 1 1/2 to 3 in. wide | 5 |
| Sand drums | | 3 to 4 in. wide | 6 |
| 24 in. long | 5 | 4 to 6 in. wide | 7 |
| 30 in. long | 6 | Buzz planers, moulders, planers and all other machines having knives | |
| 36 in. long | 7 | Up to 6 in. long | 4 |
| 48 in. long | 8 | 6 in. to 8 in. long | 5 |
| 60 in. long | 10 | 8 in. to 12 in. long | 6 |
| Belt sander | | 18 in. to 24 in. long | 8 |
| Up to 4 in. wide | 4 | 24 in. to 30 in. long | 9 |
| 8 to 12 in. wide | 6 | Floor sweep-up | 6 |
| Disc sander | | | |
| Up to 24 in. diam | 5 | | |
| 24 to 36 in. diam | 6 | | |

book—Power, p. 1–81, forming Vol. 2 of this series. The main duct velocities average 3000–3500 ft. per min.

Average sheet-metal gages on blow pipe systems are as follows:

| Diameter, in. | Up to 5 | 6 to 8 | 9 to 15 | 16 to 24 | 25 to 32 | Over 32 |
|---|---|---|---|---|---|---|
| U. S. Gage No. | 26 | 24 | 22 | 20 | 18 | 16 |

The type of collector ordinarily used is the common cyclone. In most cases the fan blows into the collector, inasmuch as wood shavings are not particularly abrasive in action against the fan wheel.

**GRINDING, POLISHING AND BUFFING SYSTEMS** are another type of common blow pipe system. The general principles concerning design of hoods, ducts, etc., apply as in the case of planing mill exhaust systems, except that, ordinarily, higher suctions are required because the greater speed and the heavier particle tends to throw the dust out of the hood. Fig. 8 shows typical hoods used for abrasive exhaust systems.

Customary branch pipe sizes for various sizes of wheels are as follows:

| Wheel size, in. | 6–12 | 13–16 | 17–24 |
|---|---|---|---|
| Pipe diam., grinders, in. | 3 | 4 | 5 |
| Pipe diam., buffers, in. | 4 | 5 | 6 |

**TUMBLING BARREL EXHAUST SYSTEMS.**—Tumbling barrels ordinarily are equipped with dust outlets. Ducts are run to branches and to the fan accordingly. A minimum of 4000 ft. per min. air velocity should be provided from the vent opening. A preference would be 5000 ft. per min. Tumbling barrels are connected together like planing mill exhaust systems. The trunnion opening varies from 4 to 8 in., and the tumbling barrels are equipped with one or two trunnions. The common cyclone or cloth screen filter ordinarily is used to collect the dust.

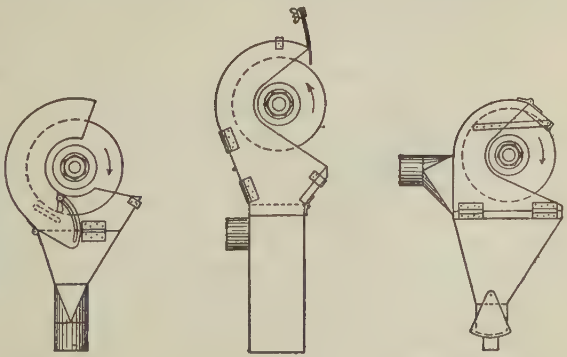

Fig. 8. Typical Hoods for Grinding and Buffing Wheels

**STATE LAWS.**—Particularly the states of Illinois, New York, New Jersey, New Hampshire, Michigan and Ohio have definite laws specifying either the pipe riser diameter, the suction at the hood, or the hood velocity or one or more of these elements. Some states specify no velocity, some specify 2 in. branch pipe suction, and some 5 in., whereas in some cases a flat specification of 9000 ft. per min. velocity is specified in the branches. State laws cover likewise the location of the collector, the size of the dampers and the design and arrangement of explosion relief vents. Some states specify that sheet-metal hoods are to be supplied at the points of dust production in granite and marble dust industries.

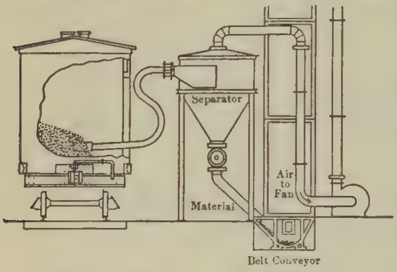

Fig. 9. Grain Unloading System

**PNEUMATIC CONVEYING.**—If a windstorm blows hard enough, solid particles are carried. The velocity of the air is a function of the size and weight of the suspended particle. To suspend a particle, the air thrust must equal the weight of the particle. Ordinary air at 4000 ft. per min. will exert a pressure of 5.2 lb. per sq. ft. The pressure is proportional to the square of the velocity. For conveying, air pressure must be greater than the weight of the material.

The critical air velocity is the vertical flotation velocity. Velocity for conveying the same material should be two to three times the critical velocity. The equation for the critical air velocity is, on the basis of standard air,

$$V = 3280\sqrt{(d \times \text{sp. gr.})}$$

where $V$ = critical velocity, ft. per min.; $d$ = diameter, in.; sp. gr. = specific gravity of the solid material. Critical velocity varies as the square root of the absolute temperature. At 600° F. the coefficient is 4640. Critical air velocities for common grains are as follows:

| Kind of grain | Wheat | Corn | Oats |
|---|---|---|---|
| Critical velocities, ft. per min. | | | |
| Lightest grains lifted | 1150 | 1350 | 800 |
| 50% of grains lifted | 1350 | 1600 | 950 |
| All grains lifted | 1500 | 2200 | 1150 |

**Pipe Carrying Capacity.**—The weight of material transported per pound of air varies directly as the square of the air velocity. Therefore, the carrying capacity of a stream of air increases directly as the cube of the velocity. The horsepower required to move the air varies also as the cube of the velocity; therefore the weight of the material moved is proportional to the fan horsepower. A 10-in. pipe, fed continuously at 5500 ft. per min. air velocity will handle 5 tons per hour. Capacity is proportional to area of the pipe.

Wheat can be transported with an allowance of 3.75 lb. of air per pound of material with a clean air velocity of 5000 ft. per min.

**Material Unloading Systems.**—Fig. 9 shows a common unloading system, such as is used for grain. Generally, such systems are not commercial when designed in connection with centrifugal fans. Since 2 to 3 lb. of air must be moved per pound of material, the efficiency of the system on the basis of the weight of the material moved, is low. The limit of suction of common centrifugal fans is 2 lb. per sq. in., which can be increased to 4 lb. per sq. in. by special design. Positive blowers (See Kent's Mechanical Engineers' Handbook—Power, p. 1–85, forming Vol. 2 of this series) are used at 6 to 10 lb. per sq. in. suction. They must draw through the collector, since the positive blower cannot handle material through its rotor. With such high suction, practically a full stream of material is pumped through the pipe. With high suctions, the pipe size is reduced considerably.

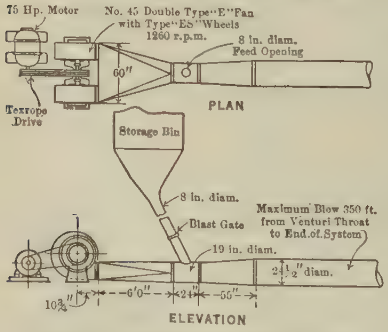

Fig. 10. Material Fed Beyond Fan

**Ducts for Conveying Cotton, Wool and Flax,** and other fibrous textile materials should be smooth inside and free from inward metallic projections. They should be made not less than 10 in. diameter to prevent stoppage. Duct velocity can be approximately 3000 ft. per min. with an allowance of 200 c.f.m. per pound of conveyed material. Stringy material that passes through the exhaust fan will clog up, unless cast-iron impellers are used. This type is cast in one piece and has smooth anticlastic surfaces, affording no points of adherence for stringy substances.

**Materials Subject to Damage by the Fan Impeller** are fed into the duct beyond the fan discharge. Fig. 10 illustrates such a system. The feeding pipe can be admitted at the throat of a Venturi in the fan duct to cause a siphoning from the storage bin.

## 4. CEMENT MILL DUSTS

**PRINCIPAL SOURCES** of dust are: 1. Grinding and handling clay and lime raw materials. 2. Pulverized coal for firing the rotary kilns. 3. Fly ash in the products of combustion from the rotary kilns. 4. Grinding calcined clinker. 5. Handling and bagging finished products. All these, except item 3, are collected by use of fan exhaust hood and piping systems, together with vortex collectors or cloth screen filters. When the air is highly laden with moisture, all piping, etc., should be well insulated to guard against condensation and the consequent plugging of the system. High duct velocities are advisable. Generous use of clean-out doors is necessary.

Fly ash is collected by vortex collectors, or electric, or spiral tube precipitators. Gas temperatures range from 400° to 1300° F., and contain a high percentage of $CO_2$. Materials of construction of the collector system should be chosen accordingly. When fly ash is collected at high temperature, the alkali content passes through the collector as a fume and consequently condenses when the gases strike the atmosphere.

Cement dusts are collected primarily on account of the health and nuisance factor, although they have a good recovery value. The fly ash contains about half calcined material which can be re-burned, unless high alkali content makes this inadvisable. Finished cement dust is especially valuable. Its greater fineness gives it superior strength.

## 5. BOILER FLY ASH

Size and quantity of ash discharged from boiler stacks depends on many factors, as: 1. Kind of firing, as stoker or pulverized fuel. 2. Percentage of ash in coal. 3. Kind of coal. 4. Ash fusion temperature. 5. Type of pulverizers. 6. Design of breeching, and heat exchangers. 7. Design of boiler. 8. Boiler rating, and other factors.

Conditions should be analyzed to determine whether to utilize a well proportioned drop box or settling chamber type collector; or whether to use the high-efficiency vortex collector, the spiral scroll collector, or the electric or spiral tube precipitator.

**Stoker-fired Boilers** may cause a flue gas dust concentration up to about 4 grains per cubic foot of gas. An average screen analysis of a typical sample, in which average combustible is 40%, and average weight is 40 lb. per cu. ft., is

| | | | | |
|---|---|---|---|---|
| Screen mesh ......... | 100 | 200 | 325 | 400 |
| Percent passed ....... | 52.8 | 36.8 | 28.4 | 20.4 |

This ash has a high local nuisance factor. Its relative coarseness causes rapid settling over a near, hence small, area. The nuisance concentration is high.

**Pulverized Fuel Fired Boilers.**—Dust concentration in flue gases varies from about 2 to 10 grains per cubic foot, depending on the percentage of ash in the coal, its fusion temperature, whether dry- or wet-bottom furnaces, design of boiler system, etc. A typical screen analysis is as follows, the sample containing 10% combustible and weighing 52 lb. per cu. ft.:

| | | | | |
|---|---|---|---|---|
| Screen mesh ........ | 100 | 200 | 325 | 400 |
| Percent passed ...... | 98.0 | 89.0 | 78.0 | 65.0 |

This dust can cause a nuisance a considerable distance away from the source, due to dust fineness. A given wind direction concentrates the pattern over an elliptical area. It is extremely abrasive to the induced draft fan wheel and housing. Dust collectors should be placed ahead of the fans. From a *fan maintenance standpoint* the flue gas temperature and fan static pressure are vital factors to determine advisability of the use of dust collectors. The fan wheel tip speed varies as the square root of both factors. Fan erosion varies exponentially with wheel tip speed. The exponent has been quoted as high as 9. With flue gas temperature around 450° F., fan erosion becomes potent above about 10 in. static pressure for stoker-fired boilers, and above 6 in. for pulverized-fuel-fired boilers.

From a *product standpoint*, fly ash or cinders practically are worthless. Aside from the fan maintenance angle, collectors are used to eliminate a nuisance.

**Required Collector Efficiency** depends on: 1. Proximity of nuisance-conscious neighborhood. 2. Height of stack. 3. Screen analysis of ash. 4. Quantity of ash. 5. Fan static pressure. 6. Money available.

Collectors which will remove 90–95% of the nuisance-range coarser particles, which results in an overall efficiency of 75–85%, should handle the problem satisfactorily.

# ILLUMINATION

## THE SCIENCE OF LIGHTING

**By Matthew Luckiesh**

The science and art of lighting may be divided for convenience into three major links: 1. Production of light; light-sources and their modifications. 2. Control of light; lighting equipment and installations. 3. Specification of light and lighting; foot-candle levels and lighting effects.

The first two links are means to certain complex ends. The third includes the efficiency, production, safety, comfort and welfare of human beings. The first link is a well-developed physical science quite beyond the realm of the designer and user of lighting. The second link also is a physical science dealing largely with optical facts and laws. The designer of lighting installations has been chiefly interested in this phase of the art of lighting. He chooses among the light-sources available, combining them and even modifying them for his purpose. However, the third link is the most important one for it not only completes the chain, but it largely determines the first two links. This link is a psychophysiological science, whose development lagged behind the first two. As a consequence, specifications of light and lighting have been largely a matter of guesswork, empirical at best. Besides, owing to the belated development of this link, the first two which are merely means to an end have received much more attention than the end itself.

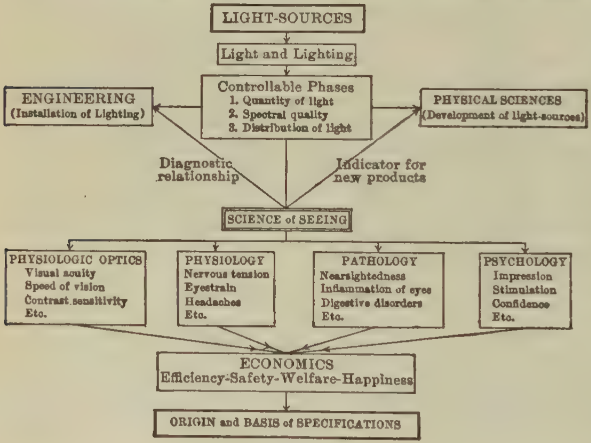

FIG. 1

The ultimate purpose of light and lighting is to make seeing possible. The ideal is to provide light in such amounts and lighting of such character that human beings can see most efficiently, effectively, accurately, quickly and safely. Therefore, specifications of light and lighting must be based upon a science of seeing. This involves a broader viewpoint and different concepts than the science of vision does. Eyes and light are mere tools and seeing is an activity of the entire human being operating as a human seeing-machine. See M. Luckiesh and F. K. Moss, The Human Seeing-Machine, *Jour.* Frank. Inst. ccxv, p. 629, June, 1933. The author and his colleagues, particularly Frank K. Moss, have been evolving this new viewpoint,* the science of seeing, and the researches arising therefrom are revolutionizing lighting considerations, particularly specifications. Fig. 1 illustrates the place of the science of seeing in the science and art of illumination.

**DEFINITIONS.**—A few important definitions (*Trans.* Ill. Eng. Soc., xxviii, p. 265, 1933) are as follows:

**Light.**—For the purposes of illuminating engineering, light is radiant energy evaluated according to its capacity to produce visual sensation.

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* Seeing—A Partnership of Lighting and Vision, M. Luckiesh and F. K. Moss, Williams & Wilkins, Baltimore, 1931.

**Luminous Flux.**—Luminous flux is the time rate of flow of light.

**Lumen.**—The lumen is the unit of luminous flux. It is equal to the flux through a unit solid angle (steradian) from a uniform point source of one candle, or to the flux on a unit surface all points of which are at unit distance from a uniform point source of one candle.

**Luminous Intensity.**—Luminous intensity, of a source of light, in a given direction, is the solid-angular flux density in the direction in question. Hence, it is the luminous flux on a small surface normal to that direction, divided by the solid angle (in steradians) which the surface subtends at the source of light.

**Candle.**—The candle is the unit of luminous intensity. The unit used in the U. S. is a specified fraction of the average horizontal candlepower of a group of 45 carbon-filament lamps preserved at the Bureau of Standards, when the lamps are operated at specified voltages. This unit is identical, within the limits of uncertainty of measurement, with the International Candle established in 1909 by agreement between the national standardizing laboratories of France, Great Britain, and the United States, and adopted in 1921 by the International Commission on Illumination.

**Candlepower.**—Candlepower is luminous intensity expressed in candles.

**Foot-candle.**—The foot-candle is the unit of illumination when the foot is taken as the unit of length. It is the illumination on a surface one square foot in area, on which there is a uniformly distributed flux of one lumen, or the illumination produced at a surface all points of which are at a distance of one foot from a uniform point source of one candle.

**Brightness.**—Brightness is the quotient of the luminous intensity of a surface measured in a given direction divided by the area of this surface projected on a plane perpendicular to the direction considered.

**Reflection-factor (Reflectance).**—The reflection-factor of a body is the ratio of the light reflected by the body to the incident light. It is the sum of the regular and the diffuse reflection-factors.

**EFFICACY OF LIGHTING EQUIPMENT.**—Light-sources are rated in lumens per watt and in total lumens output. The efficiency of lighting equipment or an installation is the ratio of the lumens output to the total lumens produced by the light-source (or sources). The efficacy of lighting equipment or an installation is determined by its contribution to seeing accurately, quickly, safely and comfortably. This depends on many factors, such as the delivery of light to the work-place and to the surroundings, the location of the lighting equipment, the direction of dominant light, and the diffusion of light. The object of light is to produce brightness and the object of lighting is to reveal objects by highlights and shadows. The factors involved in the visual task are *size* of objects or details to be seen, *contrast* in brightness and color with their surroundings or background, *brightness-level* resulting from foot-candles and reflection-factor and *time* available for seeing.

**UNITS OF BRIGHTNESS.**—Inasmuch as brightness is a very important factor in seeing the following relations of Table 1 are presented.

**Table 1.—Units of Brightness**

1 lambert = 0.3183 candle per sq. cm. = 2.054 candles per sq. in.
1 candle per sq. cm. = 3.1416 lamberts.
1 candle per sq. in. = 0.4868 lamberts = 486.8 millilamberts.
1 spherical candlepower emits 12.57 lumens.
1 foot-candle = 1 lumen incident per sq. ft.
1 lux = 1 lumen incident per sq. meter.
*1 lambert = 1 lumen emitted per sq. cm.
*1 lumen emitted per sq. ft. = 1.076 millilambert.
*1 millilambert = 0.929 lumen emitted per sq. ft.

* Perfect diffusion is assumed.

**Surface Brightness** may be due to diffusely or specularly reflected light, or to a combination of both. The brightness of a surface which reflects light solely diffusely (a blotting paper) is due to the foot-candles and reflection-factor. The brightness of a surface which reflects light solely specularly (a perfect mirror) is due to the brightness of the image of something seen in the mirror. Therefore, the brightness of this surface depends on its reflection-factor but not on the foot-candles upon the surface. The brightness of a surface which reflects light both diffusely and specularly (such as common metal surfaces) is complex, and common considerations of foot-candles and reflection-factor are inadequate and erroneous. This is an important matter in lighting, which has been obscured generally by undue and improper attention to foot-candles. It is necessary to distinguish between light (foot-candles) and lighting (distribution of light, brightness, etc.).

**LIGHT-SOURCES.**—During recent years (1936 and prior), tungsten-filament lamps have been the most prevalent light-source in use. Their luminous efficiencies depend on size, wattage and use, ranging from 6 to 30 lumens per watt. The rating of the tungsten lamps most commonly used for general lighting purposes ranges from about 10 to 25 lumens per watt. However, since each year witnesses improvements, it is impracticable to present details.

Low-pressure mercury vapor arcs have been in use for a long time at about 15 to 16 lumens per watt. High-pressure mercury vapor arcs are coming into use (1936) at luminous efficiencies from 25 to 40 lumens per watt. Sodium vapor lamps are already operating at 50 to 70 lumens per watt. There is promise of still higher luminous efficiencies in these new vapor lamps.

Luminous efficiency is important. but the acceptability of a light-source for lighting also depends on other factors, such as physical size, wattage, auxiliary equipment, color and spectral character of light, and obvious economic factors.

**LIGHT-CONTROL.**—In lighting equipment and installations, control of light is achieved by various kinds of media having such optical properties as opacity, refraction, diffuse transmission, and specular and diffuse reflection. Besides the use of optical systems of mirrors, lenses, prisms, etc., for special purposes, simple lighting equipment involves one or more of the foregoing optical characteristics. Polished metal and prismatic glass reflectors and other devices are widely used for light control. Equipment for general lighting may consist of metal reflectors with various degrees of specularly or diffusely reflecting surfaces. Various distributions of light are obtainable by altering the shape and surface finish. Much general lighting equipment consists of diffusely transmitting glass, large enough in relation to the lumens output to prevent the surface brightness of the glass from being too great to be annoying, even in the peripheral field of vision. The brightness of a cloudless sky commonly varies from 1 to 4 candles per square inch, and this range may be taken as the desirable upper limit of brightness in the field of view. A sunlit cloud may have a brightness as high as 20 candles per square inch. This is too bright for comfort indoors under the conditions of low foot-candle levels and brightness levels.

**GENERAL AND LOCALIZED LIGHTING.**—Lighting should be divided into general lighting for general purposes and localized lighting for specific purposes. General lighting is accomplished by indirect and direct systems, by a combination of both, or by an equivalent of such a combination. Indirect lighting requires a reflecting surface, as a ceiling. In a special case the wall or its equivalent may be used. By this method, the bright light-sources and bright areas of the lighting equipment are entirely concealed. However, a wide expanse of bright ceiling or other area may be annoying and may reduce the ability of a worker to see. In large rooms this is not necessarily ideal. It is merely better than a direct-lighting system in which very bright sources or surfaces of lighting equipment are in the field of view.

Resort usually must be made to direct lighting where there is no ceiling available, or where the reflection-factor of the ceiling cannot be maintained properly. Metal or glass reflectors or shades can be of such shape and height that workers cannot conveniently see the surfaces of the equipment which usually are unduly bright.

Between purely indirect and direct systems there are many which are carelessly cataloged as semi-indirect lighting. Glass reflectors in direct-lighting systems permit some light to escape to the upper surroundings and ceiling. Even this low brightness adds much to the satisfactoriness of the lighting. Metal reflectors have been perforated at the top, or are hung on diffusing glass envelopes in such a manner that some light escapes upward. At the same time the light-source is so concealed that the visible brightness of the fixture is reduced to a reasonable limit. This glassteel fixture has been quite satisfactory in many factories. In offices, and in many clean factories with painted walls and ceiling, simple fixtures of diffusely transmitting glass, entirely enclosing the light-source, commonly are used. This is termed direct lighting, notwithstanding the fact that nearly as much light is emitted upward as downward.

Fixtures for general lighting are best appraised by the percentages of light emitted generally upward and downward, respectively. This not only makes it possible to design or choose them wisely for a given interior, but also gives an idea of the lighting effect which they will produce in any interior.

**SPECIFICATION OF LIGHT AND LIGHTING** is the third link in the chain and is the important one. It can be divided into three parts: 1. Quality of light; color and spectral character. 2. Quantity of light; foot-candles and brightness. 3. Quality of lighting; distribution of light, brightness, etc.

**QUALITY OF LIGHT.**—The color of light is not necessarily an indication of its spectral character, although the latter always is an indication of the former. The color of light is important from a psychological viewpoint. Spectral character may be important from the viewpoint of vision in special cases. Daylight is white or bluish-white in color. Psychologically it is associated with work and with daytime. For this reason a white light may seem to be a better light for work-places although there is not yet available any scientific proof of this. Nevertheless human beings, having evolved outdoors, probably are best adapted to white light of a continuous spectrum.

It has long been known that strictly monochromatic light (of a narrow range in wave length) is focused more definitely by a simple lens, such as the eye system, than light of an extended spectral range. Of the unmodified illuminants available at present, sodium light is the only approximately monochromatic light available. Visual acuity, or the ability to distinguish fine detail near the threshold in size visible, is higher under sodium light than it is under the same brightness of light of extended spectral character.[*] This fact can be taken advantage of in special cases, but it should be noted that the size of details to be seen ordinarily is much greater than threshold or near-threshold size. Only near the threshold in size is visual acuity important. Besides, it should be noted that under monochromatic light all color, excepting that of the illuminant, disappears and all the advantages of color-contrast are lost. Extensive studies of this monochromatic light have been made without revealing any other advantages, excepting higher visual acuity at the size threshold, over light of an extended spectral character.

Similar studies have been made with mercury vapor light and no advantages among the complex factors of seeing have been revealed. Even visual acuity is no greater under mercury light than under continuous-spectrum light. Owing to its discontinuous spectrum, colors are distorted under mercury light. A combination of light from high-pressure mercury arcs and tungsten-filament lamps provides a satisfactory psychological white light without much distortion of colors. This combination utilizes the high efficiency of these mercury arcs and may be used extensively in the work-world on account of its whiteness.

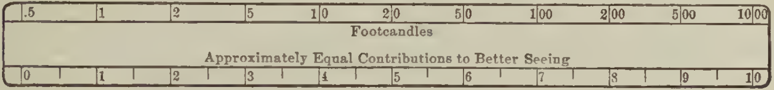

Fig. 2. The Relation of Foot-candles to their Effectiveness

Thus it is seen that excepting from a psychological standpoint and the appearance of colors, scientific knowledge at the present time does not reveal spectral character as of primary interest in lighting. Therefore, the choice of an illuminant should be based on certain other factors, such as luminous efficiency, convenience, cost, and the expense of maintenance.

**FOOT-CANDLES AND FOOT-CANDLE-LEVELS.**—The foot-candle defined as a physical unit is constant in value. In actual use, as an aid to seeing, it is not constant. One foot-candle added to one foot-candle effectively increases the ability to see, but when one foot-candle is added to 10 or 100 foot-candles its contribution to seeing is negligible. In considering the application of foot-candles for their effectiveness in promoting better seeing it is essential to realize that the foot-candle-level must be doubled in order to obtain an obvious and significant improvement in seeing. This leads to the suggestion that foot-candle levels be established in round numbers as shown in Fig. 2. These geometric steps in foot-candles correspond to arithmetic steps in effectiveness upon seeing. Furthermore, this emphasizes the practicability and desirability of establishing a series of foot-candle-levels. Then by experience and actual measurement of visibility of objects and tasks, various tasks of the work-world may be classified in the appropriate foot-candle-level.

There are two fundamental foot-candle-levels. The first and obvious one is that of barely seeing to perform a given visual task. This is the **threshold foot-candle-level.** For example, one possessing normal vision can read a newspaper or even a machinist's scale under a fraction of a foot-candle if given enough time, and if the penalties to eyesight and other human resources are ignored. Studies of vision as a tool reveal only these threshold conditions. Lighting practice has been based largely upon this misconception, and foot-candle specifications have been influenced by this narrow viewpoint.

To escape from this confined view, seeing must be considered instead of vision. Seeing is an activity of the human being operating as a seeing-machine. Seeing is work done by the human seeing-machine. This viewpoint greatly broadens the considerations of aids to seeing such as foot-candles. It reveals another foot-candle-level *i.e.*, that for *easiest* seeing for a given visual task. Investigations designed to ascertain this level for various tasks involve studies of the relations of foot-candles to various physiological and psychological effects resulting from performance of the work of seeing. This fundamental level is that at which the human seeing-machine sees most easily, accurately, and quickly. These are the objectives of the science of seeing. This ideal foot-candle-level already is known to be of a different order of magnitude from that for barely seeing.

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[*] Seeing in Sodium-Vapor Light, M. Luckiesh and Frank K. Moss, *Jour.*, Op. Soc. of Amer. xxiv, p. 5, 1934.

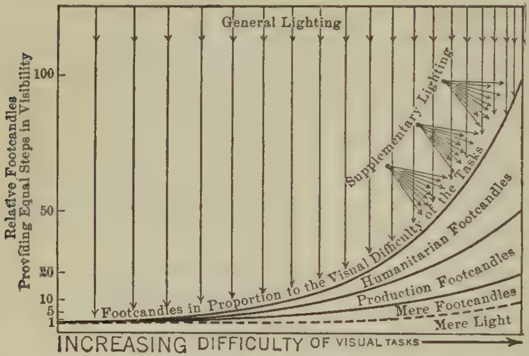

FIG. 3. Relation of Illumination to Difficulty of Visual Tasks

In general, it is indicated to be comparable with daylight levels of illumination. These are hundreds and even thousands of foot-candles. On a sunny day in summer there are commonly 1000 foot-candles in the shade and 8000 to 9500 foot-candles in direct sunlight. Indoors at the window sill there are commonly 100 to 300 foot-candles. At the work-places indoors under natural or artificial light as practiced at present there is an average of less than 5 foot-candles. Artificial light was invented as a competitor of darkness, and the science of vision did not rescue its practice from this basis of barely seeing. The science of seeing reveals that it should be a competitor of daylight. This is so revolutionary that past lighting practice is of little use as a guide.

Fig. 3 illustrates the evolution of the practice from primitive lighting to scientific specification of foot-candles for easy seeing. The vertical geometric scale deals with relative foot-candles from the proper viewpoint of effectiveness. The horizontal scale is one of increasing difficulty of the visual task. For centuries, mere light was available from a light-source near the worker. With the development of gas and electric lighting, mere foot-candles were supplied to entire work-areas. Next came the recognition that the production of the worker was influenced by foot-candles up to a certain level. Finally the science of seeing began to indicate the ideal levels for easiest seeing. These levels of hundreds and even thousands of foot-candles can be obtained economically by general lighting plus supplementary lighting as indicated on the diagram.

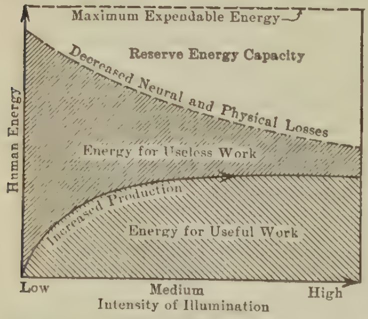

FIG. 4. Relation of Illumination to Production

**Production of Workers.**—Carefully performed studies reveal that the production of a worker ceases to increase appreciably after a level of 20 or 30 foot-candles is reached. Actually in naturally performed work in carefully controlled researches, the rate of work done has been found to increase gradually as the level is increased to at least 100 foot-candles. However, these slight increases usually are not of much practical importance. Before the new concepts and viewpoint of the science of seeing entered into lighting considerations, production was assumed to be an important measure of the beneficence of foot-candles. This is not true, for it has been adequately proved that the production of a worker may be practically the same for two tasks differing greatly in visual difficulty and, therefore, in their drain upon the human resources of the worker.

Fig. 4 illustrates the relation between illumination and the expenditure of human energy. This is necessarily diagrammatic, because it is impossible to give numerical values to the ordinates. As the intensity of illumination increases, production increases along the typical curve indicated. Eventually it reaches a practical maximum. From the viewpoint of narrow economics, the efficiency of the worker is measured in the amount of work done per dollar, or per hour. However, the true efficiency of the worker as a human seeing-machine is the ratio of energy expended for useful work to the total energy expended. As the foot-candles are increased beyond the point where the useful work of the worker ceases to increase appreciably, the work is done more and more easily as indicated. Considerations of this sort lead to the conclusion that humanitarian foot-candle-levels are far higher than production-levels. This diagram is supported by many data pertaining to the costs of seeing to the human seeing-machine.

Light and lighting are always factors in seeing, but there are other factors. Seeing is complex, but it can be divided into at least three parts, all of which are important to the human seeing-machine. These are: 1. The visual tasks or things to be seen; 2. The visual sense as a tool, and as a doorway to the consciousness; 3. The internal realm of psychophysiological effects of seeing. This is a complex chain leading from the visual tasks to the end-products of human efficiency, behavior and welfare. Light and lighting are controllable factors which influence the ease or difficulty of the visual task. They should be fitted to the eyes involved. They generally can aid subnormal vision even more than they aid normal vision. They deeply influence the physiological effects of seeing as far removed from the eyes as the far-reaching effects of food are removed from the stomach. This view reveals the futility of eyes and human beings as capable judges of the effectiveness of foot-candles above the threshold level of barely seeing. The human being needs dependable aids for determining foot-candle requirements for various tasks. And these must be founded upon a science of seeing.

**RECOMMENDED FOOT-CANDLES.**[*]—Foot-candle specifications necessarily have been empirical, and in many cases have been unnecessarily inconsistent. There has been no rational basis until recently (1936). The indicated ideal foot-candle-levels for easiest seeing are so far above those for barely seeing that in many cases they cannot be obtained by prevalent lighting methods at acceptable costs at the present time. For various reasons, compromise is necessary, but this is common in engineering. However, foot-candle specifications can be consistent if based upon equal visibility. In addition to this the Luckiesh-Moss visibility meter [†] makes it possible to specify foot-candles for various visual tasks on a rational basis. This device takes into account important factors of the visual task such as size, contrast and brightness, and also the visual sense of the user.

In Table 2 an attempt has been made to provide by brief description an example of foot-candle specifications for all the common tasks of the work-world. These are ultra-conservative.[‡] They are frankly a compromise between optimum values and prevalent values, or those obtainable by available lighting equipment and present lighting methods. However, these recommendations take into account the geometric scale of foot-candle effectiveness, the effects upon visibility of such factors as size, contrast and brightness, and other new concepts and knowledge embodied in the science of seeing. They are consistent and rational, and, therefore, represent a great improvement over the primitive and unnecessarily poor conditions for seeing which are prevalent throughout the indoor world in the daytime and everywhere at night.

**Table 2.—Foot-candle Specifications for Various Classes of Work**

**100 Foot-candles or More.**—For very severe and prolonged tasks, such as fine needlework, fine engraving, fine penwork, fine assembly, sewing on dark goods and discrimination of fine details of low contrast, as in inspection.

**50 to 100 Foot-candles.**—For severe and prolonged tasks, such as proof-reading, drafting, difficult reading, watch repairing, fine machine work, average sewing and other needlework.

**20 to 50 Foot-candles.**—For moderately critical and prolonged tasks, such as clerical work, ordinary reading, common bench work and average sewing and other needlework on light goods.

**10 to 20 Foot-candles.**—For moderate and prolonged tasks of office and factory and, when not prolonged, ordinary reading and sewing on light goods.

**5 to 10 Foot-candles.**—For visually controlled work in which seeing is important, but more or less interrupted or casual and does not involve discrimination of fine details or low contrasts.

**0 to 5 Foot-candles.**—The danger zone for severe visual tasks, and for quick and certain seeing. Satisfactory for perceiving larger objects and for casual seeing.

**VISIBILITY AND FOOT-CANDLES.**—The difficulty of performing visual tasks arises from such factors as size of critical details, contrast of these with their surroundings or background, and brightness due to the effects of foot-candles and reflection-factor. The threshold size that average normal eyes can see under a few foot-candles if the contrast is high, as black on white, is about 1 minute visual angle. This corresponds to about 0.004

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[*] Recommended Foot-candles, M. Luckiesh and Frank K. Moss, *Trans.*, Ill. Eng. Soc., xxvi, 1931, p. 1061.

[†] Visibility—Its Measurement and Significance in Seeing, M. Luckiesh and Frank K. Moss. *Jour.* Frank. Inst. ccxx, p. 431, 1935.

[‡] Seeing and Human Welfare, M. Luckiesh, Williams & Wilkins, Baltimore, 1934.

in. at a distance of 14 in., or the width of a man a mile distant. Excepting in special cases, human seeing-machines ordinarily are not obliged to see objects so near the size threshold. The science of seeing reveals that for sizes larger than 3 minutes visual angle, contrast becomes the more important factor in visual tasks. With these factors commonly fixed in visual tasks, brightness can be increased, with the result that visibility can be controlled to a large degree. If the object can be viewed against a removable background, the latter can be made of high reflection-factor. This requires fewer foot-candles than a background of low reflection-factor for a given brightness. In relatively rare cases a dark background is expedient. In these the brightness of the object can be increased by increasing the foot-candles.

For example, large telephone directories necessarily are printed on thin, inexpensive paper, whose reflection-factor is lower than that of good white paper. Besides, the contrast between the print and paper is lower than in the case of excellent printing of the same type on thicker white paper. The result is that about three times the foot-candles are necessary for equal visibility of the same text in the telephone directory as in a well-printed book.

Clerical work varies enormously, but, in general, the visual tasks involve high contrast. In official codes of foot-candle specifications it has been the practice to recommend only twice the foot-candles for sewing on dark goods as for office work, notwithstanding the low contrasts between thread and textile in the former case. From the sole consideration of brightness, the garment worker commonly should have at least 10 times the foot-candles of the office worker. This same principle applies through the work-world in fine assembly, inspection, etc.

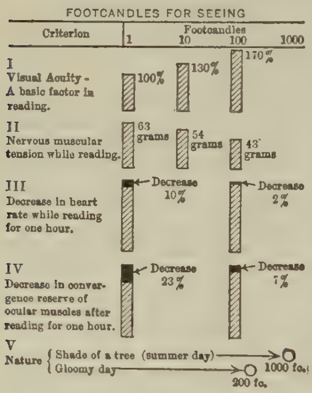

Fig. 5. Criteria for Ascertaining Relation between Foot-candles and Ease of Seeing

The science of seeing includes correlations between visibility and ease of seeing. Such factors as shown in Fig. 5 are taken into account.* It is evident that the task of reading not only is made easier by higher levels of illumination, as indicated by item I, but also the hidden penalties are minimized, as indicated by items II, III, and IV. The decrease in nervous muscular tension with increase in foot-candle level means less energy wasted and less bodily and mental fatigue. It should be noted that this task of reading books printed in large type (10- and 12-point) could be done under a small fraction of a foot-candle (the threshold foot-candle-level). The level for easiest reading is apparently above 100 foot-candles. And this is a relatively easy task compared with those involving very fine details and low contrasts. Even the heart-rate (item III) is affected by this relatively easy task and is more nearly normal under 100 foot-candles than under a few foot-candles. Item IV shows that fatigue of certain eye-muscles decreases with increases in foot-candle-level. These, and other researches and correlations, have definitely established that the optimum visibility of details of a black object on a white background is obtained under a level of illumination well above 100 foot-candles.

**The Luckiesh–Moss Visibility Meter** has two rational scales, viz., 1. Relative visibility; scale range, 1 to 20. 2. Foot-candles recommended; scale range, 1 to 1,000. In all the researches and calibration tests upon which these scales are based, subjects with average normal vision were used. Therefore, other factors being equal, the meter is also a practical test of the part that subnormal vision may play in everyday seeing.

The visibility scale has an absolute rational basis founded upon the maximum ability of persons with average normal vision to recognize the details of a simple describable object of definite size and contrast. In an absolute sense, a setting at unity on the visibility scale represents the limit of average normal vision under favorable conditions for seeing. The scale value 2 indicates that the object or task is twice as visible as the smallest object which can be recognized by persons with average normal vision. These scale values not only represent relative visibility of objects or visual tasks, but also can be interpreted as factors of safety in seeing.

The scale values of recommended foot-candles have no obvious relation to the scale values of relative visibility. The former are rational among themselves in a relative sense. However, the actual values are an arbitrary compromise between the foot-candles necessary for barely seeing and the enormously higher foot-candle levels for easiest seeing, as indicated by researches in seeing. That the scale of recommended foot-candles is conservative is attested by the fact that it is based on the visibility of reading matter printed with 8-point type and black ink on excellent white paper, and illuminated to an intensity of 10 foot-candles without any preventable glare present. This task corresponds to the scale value 10. Researches in several directions have indicated that at least 100 foot-candles are desirable for that particular visual task. A scale value of 20 for another task under the same intensity of illumination as the former basic task indicates that it should have 20 foot-candles in order to be as visible or as easily performed as the foregoing standard task (8-point black print on white paper at 10 foot-candles).

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* The New Science of Lighting, M. Luckiesh and Frank K. Moss, *Trans.* Ill. Eng. Soc., xxix, p. 641, 1934.

If all visual tasks are studied with the visibility meter at the standard level of 10 foot-candles, the scale reads directly the recommended foot-candles whose actual values are an arbitrary compromise between ideals and practical engineering and economic factors. In practice when the actual ($A$) foot-candles on the task being studied differ from 10 foot-candles, the scale reading of recommended foot-candles should be multiplied by the ratio $A/10$ or $0.1\,A$ to obtain, with sufficient accuracy, the true value of recommended foot-candles for that particular task. Wherever practicable in studies of foot-candle requirements for various visual tasks, a standard intensity of illumination of 10 foot-candles should be used so that the recommended foot-candles can be read directly from the scale. The basic principles underlying the scales of the instrument have been described in detail elsewhere.*

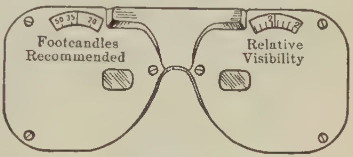

Fig. 6. Visibility Meter

Special gradient filters are turned until the user is barely able to perform various tasks. Properly used, recommended foot-candles can be read from the scale for various tasks in order that the latter may be of the same visibility as the basic task. Table 3 shows the foot-candles necessary to render the various tasks of equal visibility.

**Table 3.—Foot-candles Required for Equal Visibility**

| Visual Task | Foot-candles for Equal Visibility |
|---|---|
| Excellent printing | |
| 8-point Bodoni type | 10 |
| 6-point " " | 17 |
| 10-point " " | 7 |
| 12-point " " | 5 |
| Handwriting with pencil | 25 |
| Newspaper text matter (approximately 8-point) | 35 |
| Telephone directory—yellow pages | 55 |
| Shorthand notes in pencil | 60 |
| Reading micrometer | 70 |
| Typing on dark blue paper | 75 |
| Newspaper—stock quotations | 80 |
| Reading steel scale $^{1}/_{64}$ in. divisions | 150 |
| Distinguishing black thread on dark cloth | 500 |

If a higher or lower basic visibility is desired, the transformation is easy. Table 3 shows the 8-point type to require twice the foot-candles that the 12-point does for equal visibility. If the basic degree of visibility is desired to be that represented by 10 foot-candles on 12-point type, all the foot-candle values in the table are doubled. The visibility meter not only renders foot-candle specifications consistent with the visibility of visual tasks on a definitely rational basis, but it introduces the idea of factors of safety into lighting practice.

There are numberless visual tasks and many fields of lighting. Most tasks are not accurately describable. Therefore, in Table 3, only a few of the describable ones are presented. Foot-candle specifications can be obtained readily for any task. These eventually will appear in the publications which guide lighting practice. In many lighting fields there is a variety of tasks. By means of visibility measurements a compromise foot-candle-level may be recommended. It appears most practicable to place specific tasks and specific lighting fields in one of ten or more foot-candle-levels on a geometric scale as suggested on p. 26–15. This has not yet been done (1936), and therefore, no official foot-candle specifications can be presented for the many work-world tasks and activities.

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* Visibility, Its Measurement and Significance in Seeing, Luckiesh and Moss, *Jour.* Frank. Inst. Vol. 220, 1935, p. 431.

**QUALITY OF LIGHTING** involves the distribution of light and brightness in the visual field. It has much to do with the visibility of three-dimensional objects, with fine detail on specularly reflecting surfaces, with the psychophysiological effects of glare and of brightness of surroundings compared with that of the work to be performed. For fine work, involving the visibility of objects of three dimensions, there can be no question that a dominant source of light, well-screened from the eye, is important. If the objects are mobile, they are manipulated in the hands while the worker produces the lighting effect enabling him to see. This effect may be a high light, or shadow, or a silhouetting of the object against the proper background. In general, any lighting problem may be divided into two parts: 1. Lighting the task or central work-field; 2. Lighting the surroundings. The former requires high intensities for high visibility of the work. In the work-field, physiological factors of seeing predominate. In the surrounding field, the psychological factors predominate. However, both are complexly related.

In Fig. 7 the major factors of the problem of lighting for seeing are shown diagrammatically. Quality of lighting is a complex summary, involving all that is seen in the work-field and in the surroundings. A simple summary is impossible, but Fig. 7 should be helpful. The present designation of lighting equipment and installations is inadequate from the viewpoint of lighting for seeing. Their effect upon seeing and upon human beings depends on the distribution of brightness in the central and peripheral fields. This in turn depends on the size of the interior, reflection-factors, characteristics of the visual task and other factors. Although it has been adequately proved that a human being is a very poor seeing-meter and therefore a poor appraiser of lighting conditions, anyone with experience can recognize the more undesirable characteristics of poor lighting. However, any appraisal of a lighting condition without actually performing the tasks for a long period may be misleading.

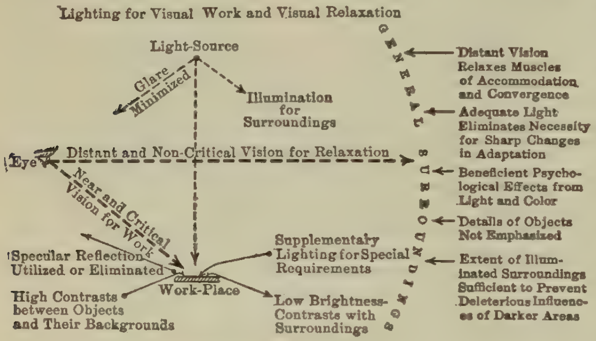
Lighting for Visual Work and Visual Relaxation
Light-Source
Glare Minimized
Illumination for Surroundings
Eye
Distant and Non-Critical Vision for Relaxation
Near and Critical Vision for Work
Specular Reflection Utilized or Eliminated
High Contrasts between Objects and Their Backgrounds
Work-Place
Supplementary Lighting for Special Requirements
Low Brightness-Contrasts with Surroundings
Distant Vision Relaxes Muscles of Accommodation and Convergence
Adequate Light Eliminates Necessity for Sharp Changes in Adaptation
Beneficient Psychological Effects from Light and Color
Details of Objects Not Emphasized
Extent of Illuminated Surroundings Sufficient to Prevent Deleterious Influences of Darker Areas

FIG. 7

Eye-fatigue is measurably greater under purely localized lighting than it is after general lighting has been added. Such an extreme also is annoying in various ways. It is unsafe when workers are exposed to dangers due to poor seeing. On the other hand, in large interiors purely general lighting is the other extreme, which may be annoying and even uncomfortable in the same way as purely localized lighting, but usually not to such a noticeable degree. Safety and comfort demand general lighting but maximal efficiency, generally is realized by adding suitable localized lighting to satisfactory general lighting.

The balance between the general and supplementary lighting not only is important, but is a valuable tool in obtaining adequate light and proper lighting. By properly balancing the two components, high foot-candle-levels can be satisfactory. By adding supplementary lighting to general lighting, various advantages to seeing can be utilized; hundreds of foot-candles can be obtained at low cost; and even 1000 or more foot-candles become economically possible.

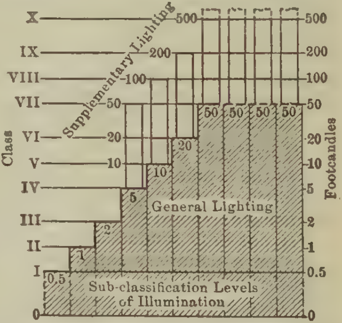

FIG. 8. Combination of General and Supplementary Lighting

Fig. 8 shows safe combinations of general and supplementary lighting for various foot-candle-levels. When and where localized lighting is to be added to general lighting depends upon many factors, too complexly interwoven to discuss here. The foot-candles supplied by supplementary lighting are shown to a limit of ten times the foot-candles supplied by general lighting. The reflection-factors of the work and of the surroundings are important. Here, painting of machinery, walls and ceiling can play an important part.

# Section 27

# SAFETY ENGINEERING

By Sidney J. Williams

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# SAFETY ENGINEERING *

By Sidney J. Williams

## 1. ACCIDENT PREVENTION

The guarding of dangerous machinery and other dangerous places is required by the laws of most states. The absence of such protection generally is penalized, also, by an increase in the rate charged for workmen's compensation insurance.

Many industrial concerns endeavor to eliminate or guard all danger points and to correct all unsafe practices, even where not demanded by state or insurance regulations, because they have found that: 1. Accidents cause, or indicate, an irregularity in production which results in lost time and inefficiency. 2. Accidents cause the temporary or permanent loss of experienced and valuable workmen. 3. Careful attention to safety, by both mechanical and educational means, raises the morale of the working force. Such companies often require that equipment which they purchase be guarded in accordance with certain specifications.

**STANDARDS FOR SAFEGUARDING.**—There has been a considerable lack of uniformity in the requirements of various state and insurance authorities; this is being gradually overcome through the formulation of standard safety codes under the auspices of the American Standards Association (29 West 39th St., New York City). The specifications outlined below are those which generally prevail; the State Department of Labor or State Industrial Board, and the insurance company carrying the risk should be consulted for additional or (in a few cases) more stringent requirements.

**THE ESSENTIALS OF AN EFFICIENT SAFEGUARD,** according to the National Safety Council are: 1. It should be so designed and constructed that it will prevent, on the part guarded, all accidents both to the operator at his regular work and to others who may slip, fall or carelessly touch the machine.

2. It should not interfere with production. If it does, it is liable to be removed. In designing a guard, it generally is wise to consult the man who will use it.

3. In general, it should be attached to the machine. If attached to the floor, use a connection which will interfere as little as possible. See Fig. 1.

4. The guarded part must be easy of access for oiling, inspection and repair, through a door or removable section hinged or fastened to the body of the guard, or to the machine. If not attached, it is likely to be left off permanently.

5. The guard should not interfere with cleaning and sweeping around the machine. It, therefore, should be kept usually about 6 in. above the floor.

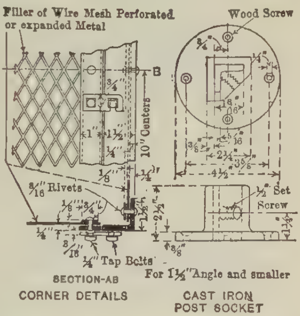

FIG. 1. Details of Machine Guard

6. The guard should be strong enough to resist injury and keep its shape. A light, flimsy guard soon becomes bent and is discarded. A substantial guard is cheaper in the end.

7. Incombustible guards are preferred; except, where subjected to acid fumes, wooden guards may be necessary. Wooden guards, soaked with oil, may become a serious fire hazard. Guards may be made of cast iron, sheet metal, wire mesh, expanded or perforated metal, or slats.

8. The guard and operating mechanism should interlock where possible, to prevent the machine operating unless the guard is in place.

9. A safeguard often can be so designed as to prevent wear on the parts guarded—for example, a solid gear enclosure.

## 2. MECHANICAL POWER TRANSMISSION

**GEARS, FRICTION DRIVES, SPROCKETS.**—"All gears and friction drives, wherever located, shall be completely encased, or, where this is impracticable, shall have a continuous band guard provided with side flanges extending inward beyond the roots of the teeth. All sprockets, wherever located, shall be completely encased. Chains shall be guarded as prescribed for belts. Where there is a spoke hazard, gears and sprockets shall always be enclosed on exposed side."

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* In this section the material included in quotation marks is taken from the Rating Schedule Standards used by most insurance companies and rating bureaus.

Cast-iron guards are preferable, as they fit snugly, present a good appearance and protect the gears from dust and injury. Guards for a variety of machines, and in many sizes, may be more cheaply made of sheet metal. In large guards, an angle iron is used to make the joint between the flat sides and the curved part of the guard. In smaller guards this joint may be made by cutting projections (like saw teeth) on each side piece and bending these over to form a smooth curve. The joint may then be made by spot welding or riveting. Pieces of angle iron $^3/_4 \times {}^3/_4 \times {}^1/_8$ in., 1 in. long, spaced 3 in. to 4 in. apart, may also be used to form the joint.

**FRICTION CLUTCHES.**—"Any revolving part of a friction clutch having dangerous projections within 7 ft. of the floor or platform shall be completely and substantially encased. This also includes any clutch on transmission shafting that is used as the starting and stopping device of a machine, if any part of the clutch has dangerous projections which revolve when the machine is stopped."

**BELTS, CHAINS AND ROPES.**—"If vertical or inclined, 6 ft. or less; if horizontal, with lower run 7 ft. or less above floor or platform, shall be guarded as shown in Figs. 2, 3, 4 and Table 1.

"Where both runs of a horizontal belt are within 6 ft. 6 in. of floor level, the guard shall extend at least 15 in. above the belt, or to a height of 6 ft., but in no case shall it be less than 42 in. high, unless the belt is completely enclosed. Overhead horizontal belts with lower run 7 ft. or less from floor or platform level, shall be guarded on bottom and sides to a height of not less than 6 in. above the lower run of the belt, but not over 7 ft. above the floor. Where pulleys are of such dimensions and so located as to permit passage between the upper and lower runs of horizontal belts, standard railing shall be provided, and a substantial passageway, guarded on both sides and top shall be constructed, or all space completely barred against passage."

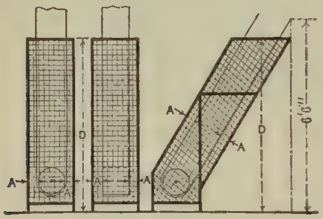


FIG. 2. Belt Guards

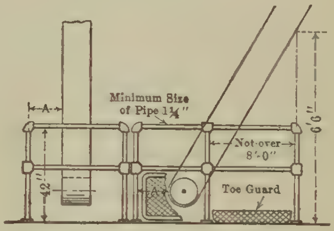


FIG. 3. Belt Guard Using Railings

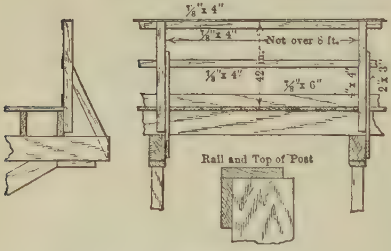


FIG. 4. Wooden Railings

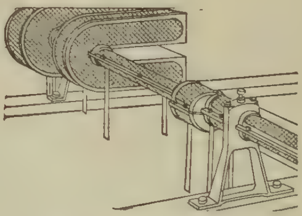


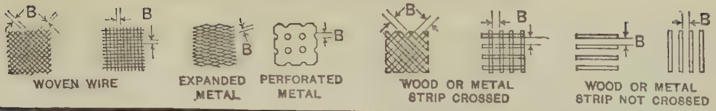
FIG. 5. Shafting Guards

**SHAFTING.**—"Vertical and inclined shafts shall be encased with stationary guards to a height of 6 ft. from floor or platform level. Horizontal shafting, less than 6 ft. from the floor or platform level shall be completely encased . . . or protected by a trough enclosing sides and top or sides and bottom as location requires." See Fig. 5. "Railings are not acceptable as guards for shafting."

**SHAFT ENDS.**—"Any exposed shaft end, which projects beyond a bearing or hub more than 2 in. and is within 6 ft. of the floor or platform, shall be cut off or completely enclosed."

**PROJECTIONS ON REVOLVING SHAFTS.**—"The following are dangerous projections and require protection wherever located: 1. Set-screws, keys, bolts, oil cups or similar projections when not within the plane of the rim of a pulley. 2. Couplings containing protruding bolts or nuts. 3. Broken pulleys and other similar projections.

**Table 1.—Belt Guard Clearances**



| Material | Minimum Clearance of Belt, in. *A* | Largest Mesh or Opening, in. *B* | Minimum Thickness | | Minimum Height of Guard from Floor or Platform, ft. |
|---|---|---|---|---|---|
| | | | In. | U. S. Std. Gage No. | |
| Woven wire | Under 4 | 1/2 | .... | 16 | 6 |
| | 4–15 | 2 | .... | 12 | 5 |
| Expanded metal | Under 4 | 1/2 | .... | 18 | 6 |
| | 4–15 | 2 | .... | 13 | 5 |
| Perforated metal | Under 4 | 1/2 | .... | 20 | 6 |
| | 4–15 | 2 | .... | 14 | 5 |
| Sheet metal | Under 4 | .... | .... | 22 | 6 |
| | 4–15 | .... | .... | 22 | 5 |
| Wood or metal strip, crossed | Under 4 | 1/2 | 3/4 * | 16† | 6 |
| | 4–15 | 2 | 3/4 * | 16† | 5 |
| Wood or metal strip, not crossed | Under 4 | 1/2 | 3/4 * | 16† | 6 |
| | 4–15 | 1 | 3/4 * | 16† | 5 |
| Standard railing | 15–20‡ | See Standard Railing, Figs. 3 and 4 | | | |

* Wood. † Metal. ‡ Maximum clearance.

4. Gears, pinions or sprockets on transmission shafting which are not meshing with other gears or carrying a chain."

"Pulleys, not carrying belts but with spokes or projections, even if these are within the plane of the rim of the pulley, shall be considered dangerous projections when within 6 ft. of floor or working platform."

"All projecting keys in revolving shafts shall be made flush or protected with cylindrical safety sleeves or by stationary enclosures. All revolving objects which project beyond the plane of the rim of pulley, gear or wheel shall be guarded by cylindrical safety sleeves, metal discs or by stationary enclosures. All revolving set-screws exposed to contact shall be made flush or countersunk beneath the surface of the metal part in which they are inserted."

**BELT SHIFTERS.**—"Each set of tight and loose pulleys shall be provided with a belt shifter. This shall be understood to mean a permanent mechanical device by means of which the belt may be shifted from one pulley to the other without touching the belt with the hand."

**INDIVIDUAL AND GROUP DRIVES.**—Certain processes of manufacture lend themselves to individual machine drives, thus eliminating transmission and the dangers of its use and repair. The lighting of a working place is improved by the absence of overhead transmission.

Group drives do not eliminate transmission but they do reduce the danger of running on belts, making repairs and adjustments, etc., since the machines in the group can be shut down easily for the moment needed for such work.

Individual drives, group drives or power control systems make it possible to shut down machines quickly in an emergency.

Also see Safety Code for Mechanical Power-Transmission Apparatus, Am. Stds. Assoc.

## 3. GENERAL MACHINERY

In addition to the guarding of belts, gears, etc., as described above, and the guarding of points of operation described below, safety requires machines to be so located that each workman will have sufficient space in which to handle his material with the least possible interference from other workmen or machines. Machines never should be so located that it is necessary for operators to stand near an aisle, or so that two operators are likely to interfere with each other. Aisles should be maintained of sufficient width to permit the passing of trucks and workmen without crowding.

Proper illumination also should be considered in placing machines. The greatest possible amount of natural light should fall on the work, and yet not shine in the operator's eyes. The same principles should govern the arrangement of artificial lights.

## 4. WOOD-WORKING MACHINERY

Whenever practicable, wood-working machines should be selected for specific operations and guards should be provided suitable for such operations. Each machine then can be more effectively and permanently guarded.

**AUTOMATIC FEEDING** is much safer than hand feeding. It is recommended for every machine where the nature of the work permits.

**HOODS** connected with an exhaust system should be provided on wood-working machines in continuous or frequent use, to remove the sawdust, chips, or shavings. Otherwise these would clog the machine and reduce its efficiency, necessitate frequent cleaning with consequent danger of accidents, make the floors slippery, increase the fire hazard, and often irritate the eyes and injure the health of workmen. The exhaust hood, if made strong enough, may serve as a guard for the cutting tool on trimmers, stickers, molders, matchers, etc.

**SHARP TOOLS** are quite as necessary for safety as the provision of hood guards. They also increase efficiency.

**FLOORS** should be kept in good condition, with a smooth surface to prevent tripping, and kept free from accumulations of waste material, small sticks, blocks, shavings and sawdust. The standing space at each machine should be covered with anti-slip material. Metal plates with abrasive grit surfaces, rubber mats, or roofing paper, or a thick paint or glue applied periodically, with sand sprinkled over it while wet, may be used.

**CIRCULAR SAW GUARDS,** according to the National Safety Council, should: 1. Be of light weight, yet of substantial construction. 2. Be made preferably of aluminum (or of brass, fiber, or wood), so that accidental contact will not break the saw. 3. Permit a clear view of the saw blade at the point of operation, when in use. 4. Automatically and easily adjust themselves to different thicknesses of material. 5. Be rigidly supported to lessen vibration and danger of contact with saw. 6. Be so constructed that they can be swung out of the way when necessary, and return to proper position. 7. Be permanently attached to the machine. 8. The guard supports should not interfere with feeding of machine by operator. 9. Exposed part of saw below table should be completely protected, as by a suction hood for carrying off the sawdust.

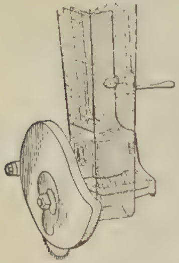


Fig. 6. Swing Saw Guard

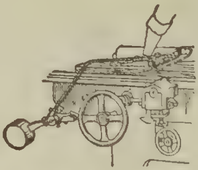


Fig. 7. Wood Planer Guard

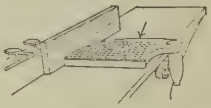


Fig. 8. Wood Planer Guard

**RIP SAWS** also should be equipped with a spreader or splitter to prevent the wood from binding on the saw blade, and to act as a guard for the back side of the saw blade.

**SWING SAWS** should have the counterweight securely fastened in place, and also secured to the ceiling by a safety chain. The travel should be confined within safe limits. The belt should be completely enclosed on the side toward operator for its entire length, and on the back side to at least 6 ft. above the floor. If a saw is located so that workmen can pass behind it, an enclosure should be built to prevent any one coming in contact with the saw blade. See Fig. 6.

**" RAILROAD " OR TRAVELING CUT-OFF SAWS** should have the hood supported from the saw carriage, thus keeping the hood always in position over the saw.

**SAFE RIM SPEED** of circular saws is about 15,000 ft. per min. or 4800 r.p.m. for a 12-in. saw, which may be increased by 25% in the case of special " high-speed " blades.

**BAND SAWS** should have upper and lower wheels completely enclosed, with the side of guard toward operator hinged or movable, for changing saw blades. The uprunning side of the blade should be completely guarded. The working side of the blade should be guarded above guide rolls or gage.

**JOINTERS AND PLANERS** (hand-fed) should be equipped with safety cylinder heads, and a guard as in Figs. 7 or 8. The part of the jointer head back of the guide should always be completely covered.

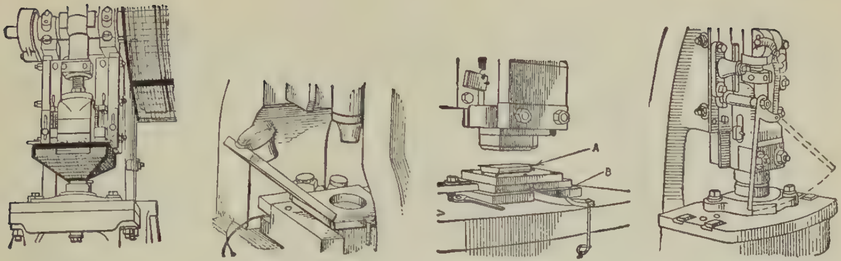


FIG. 9 FIG. 10 FIG. 11 FIG. 12
Methods of Guarding Punch Presses

**SHAPERS, MATCHERS, MOLDERS, STICKERS, ETC.,** should have cutting heads guarded by a cage or by the exhaust hood.

**ROLL FEEDS** should be guarded by a cover, or a bar in front of the intake, to prevent the operator's hands being drawn in.

Also see Safety Code for Wood-working Plants, Am. Stds. Assoc.

## 5. METAL-WORKING MACHINERY

**POWER PRESSES** (punch presses) can be made safe and also more efficient, by so arranging the press, dies, and feed that operator need never place his hand under the ram. Preferred methods are: *a*. Full-automatic feed (roll feed, plunger, chute, revolving or sliding die) with the ram guarded; basket-guard for hand-fed blanking work; see Fig. 9. *b*. Gravity feed for forming; see Fig. 10. *c*. Automatic or foot-operated kick-out; see Fig. 11. *d*. Pneumatic or other handpickers for placing and removing material. *e*. Redesign of dies to leave portion of material projecting so it may be grasped safely. If these methods cannot be used, provide a sweep guard, Fig. 12, attached to ram; a gate guard requiring removal of hand before the press can be tripped; or a two-hand operating device. In the last two cases, also provide non-repeat, and treadle-disconnecting attachments. In all cases provide a treadle guard, Fig. 13, to prevent accidental tripping, and also a friction clutch or tight-and-loose pulleys. Also see Safety Code for Power Presses, Am. Stds. Assoc.

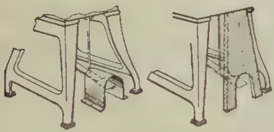


FIG. 13. Foot Treadle Guards

**LATHES.**—Gears should be enclosed. Safety dogs without projecting parts should be used. A guard around or in front of the chuck is advisable. Where the speed is changed frequently, a cone-pulley belt-shifter saves time and prevents accidents, and also permits the belt to be guarded.

**DRILL-PRESSES** should have belt shifter and belt guard the same as lathes.

**PLANERS** should have the openings in the bed covered. A railing or other guard should be provided if the table travels to within 18 in. of a wall or other fixed object.

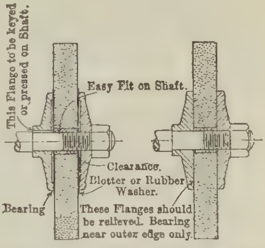

This Flange to be keyed or pressed on Shaft.
Easy Fit on Shaft.
Clearance.
Blotter or Rubber Washer.
Bearing
These Flanges should be relieved. Bearing near outer edge only.

FIG. 14. Safety Flanges for Abrasive Wheels

## 6. MISCELLANEOUS MACHINERY AND EQUIPMENT

**ABRASIVE WHEELS** should have a hood, of sufficient strength to retain the fragments in case the wheel should burst, enclosing at least 270° of bench wheels, or at least the top 180° of swing frame grinders. See Safety Code for Abrasive Wheels, obtainable from most grinding wheel manufacturers or from Amer. Standards Association, 29 W. 39th St., New York. (See also p. 21–81.)

**ROLLS,** including calender or mixing rolls in paper, rubber, and other industries, also feed rolls, should be equipped with an enclosure, plate, or bar to keep the operator's fingers out of intake; or a feed table, for the same purpose; or a quick-stop device operated by a treadle, cord, or bar within easy reach of operator, and with a minimum of lost motion.

**TRAVELING CRANES** (electric overhead) should have a railed footwalk, 18 in. wide, along the bridge; fixed ladder or stair between cab and footwalk; fixed ladder or stair from floor of shop to cab; bumpers at each end of each rail; warning gong or other signal; enclosure or railing on all sides of cab; fenders on bridge wheels; over-hoist limit switch with dynamic braking.

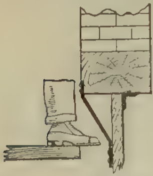


Fig. 15. Elevator Shaft Toe Guard

**ELEVATORS.**—Special safety features include: Bevel guards, Fig. 15, at least 60° from horizontal on all under-projections in shaft; gates or doors interlocking with the power control; substantial screen below overhead sheaves; proper car enclosure, including top; automatic catching device with speed governor; limit stops on machine and in shaft; device to stop machine if cable becomes slack; no-voltage and phase-reversal protection; floor lock for operating cable or shipper rope; non-slip floor in car at door, and on landings. See A.S.M.E. Elevator Code.

**STEAM BOILERS.**—Special safety features include: Location of safety valve and blow-off so as not to injure anyone in discharging; easy access to valves; two exits from boiler room; boiler valve lock, to protect men working in boiler; non-return valve; gage glass guard; furnace door lock. See A.S.M.E. Boiler Code and Section 6, Vol. 2 of this series.

**ENGINE FLY-WHEELS AND PITS** should be guarded as in Fig. 2 or 3, and Table 1. Keys and other projections should be covered. A speed governor should be provided, except for (*a*) reversing engines, and (*b*) engines direct-connected to a constant load. The following engine parts also need protection, according to the insurance rating schedule:

" Cranks, connecting rods, crossheads, tail rods, governor balls, gears, contact point of governor belt and pulley, exposed ends of engine shaft.

" **Cranks, Connecting-rod and Crosshead** shall be guarded by one of the following methods: 1. Engines with overhanging crank shall have a standard railing, Fig. 3, not less than 15 in. nor more than 20 in. from the moving part. 2. Center-crank engines shall have a guard in the form of a plate or wire mesh at each side of the crank at the main bearing. 3. When cranks or crossheads are housed in, the covering shall have an opening large enough to permit examination of the bearing, except in the case of engines with oil splash system, where such opening is not required.

" **Tail Rods** may be enclosed with sheet metal or wire fabric in such a way as to prevent danger of accidental contact with the moving end, or they may be protected by a standard railing placed between 15 in. and 20 in. away from the point of maximum travel.

" **Governor Belt and Pulley** so located that any person who in his regular duties may come in contact with them, shall be guarded by means of a solid metal plate, placed 1 to 2 in. from the pulley and at least 4 in. greater than the diameter of the pulley. It shall be rigidly supported and shall not interfere with inspection, oiling or operation of any part of the equipment.

" **Fly-ball Governors** within 7 ft. of floor or platform shall be protected by a substantial guard or basket of sheet metal, woven wire fabric, etc., so arranged as to prevent danger of accidental contact with the rotating ball.

" If any engine parts are guarded by railings, the railings shall be of standard construction, and covered with wire fabric, sheet metal or other suitable material at all points within 15 in. of moving parts. The clearance between the plane of the railing and the moving parts which it protects shall not exceed 20 in."

**PORTABLE LADDERS** should be equipped with anti-slip bases of a type suited to the floor; for rough wood floors, metal points or lead-coated bases; for concrete floors, lead or carborundum. For some floors, such as iron or wet concrete, it is almost impossible to find a ladder shoe which will not sometimes slip and the only safe practice is to station an attendant at the foot or to lash the ladder at top or bottom or both.

**Stationary Ladders** should have proper clearance on all sides—8 to 12 in. in the back, 24 to 36 in. in front, 15 in. on each side of the center line of the ladder. For vertical ladders, especially if of considerable height, an enclosure or backrest is desirable.

## 7. ELECTRICAL HAZARDS

The construction of high voltage generating equipment and transmission lines is discussed under Electrical Engineering, Section 15, Vol. 2 of this series. See also Pender, Electrical Engineers' Handbook, Vol. 4 of this series. All other phases of electrical safety are covered in detail in the National Electrical Safety Code (U. S. Bureau of Standards, Washington).

**UTILIZATION HAZARDS** are chiefly low-voltage, except the step-down transformer, which should be carefully isolated from all but qualified persons. Plant equipment of 550 volts is admittedly hazardous, and 220 or even 110 volts may kill or injure a person in poor health, standing on or touching a direct ground. Present practice, therefore, favors the enclosure of all switches. Portable lights should have heavily insulated cord; if the user is on a " ground " the socket should be keyless, with an insulating enclosure and handle. Motors, generators, switchboards, transformers, etc., should be enclosed or protected if within 8 ft. of the floor or working platform. Circuits of from 25 to 150 volts to ground should be permanently grounded, except two-wire direct-current systems and circuits exposed to leakage or induction from higher voltages. Motor, generator, transformer, and switch frames, and cases, should be grounded if operating at over 150 volts to ground or if exposed to inflammable gas or explosives, or if within reach of other exposed grounded surfaces. Working space next to live parts carrying over 150 volts to ground should be at least 30 in.; not over 150 volts, 18 in.; space between live parts (over 150 volts) on one side and live or grounded parts on other side, 48 in.; not over 150 volts, 30 in.

## 8. INDUSTRIAL BUILDINGS AND PLANTS

Forethought in the arrangement of buildings and equipment will avoid hazards which later are difficult or impossible to remove. The most common hazards to be considered are:

**CLEARANCE** between railroad track and building—8 1/2 ft. from center of track; between end of crane bridge and wall or column; below present or future overhead pipes, ducts, shafting, etc.; between stairs and floor above.

**ACCESSIBILITY** of valves, switches, oil cups, lamps, etc., should be obtained either by location or by providing runways and stairs or fixed ladders.

**CROSSINGS** of railroad, automobile, hand-truck and pedestrian traffic can often be avoided by careful planning.

**LIGHTING** is necessary for safety as well as efficiency. See Illumination, p. 26–12, and also the specifications of the Illuminating Engineering Society (New York).

**EXITS.**—Provide at least two exits from every point. Preferred types are stairs enclosed with fire-resisting walls and fire-doors and " horizontal exits," *i.e.*, doors through fire walls, or bridges or balconies to other buildings or sections. Outside fire escapes are less safe; if used at all, they should be not steeper than 45°, substantially constructed, well railed, extended to ground, and protected by wire glass windows. See specifications of National Fire Protection Association (Boston).

**STAIR RAILS** 36 in. from center of tread should be provided: On all open sides of stairs; on one side of enclosed stairway 4 ft. wide or less; on both sides of enclosed stairway over 4 ft. wide between walls; on both sides and in center of stairway over 8 ft. wide between walls. (See Figs. 3 and 4.)

**FLOORS** should be of material which, under the usage to which they are exposed, will not splinter nor become slippery, loose, or uneven. Cast-iron or steel door sills, steps, etc., become dangerously slippery unless an anti-slip element is provided.

**FLOOR OPENINGS** should be enclosed or guarded with standard railings (Figs. 3 and 4) or be equally protected with fixed safety covers with bows and rods. All openings through which articles might drop and injure people below should be guarded with standard toe boards. See Figs. 3 and 4. See also Safety Code for Floor and Wall Openings, Am. Stds. Assoc.

## 9. EYE PROTECTION AND SAFE CLOTHING

**THE HEAD AND EYE PROTECTION** Code of the U. S. Bureau of Standards classifies eye hazards as follows (with example of each class):

*a*. Large flying objects—*e.g.*, chipping; *b*. Dust and small flying particles—grinding; *c*. Dust and wind—automobile driving; *d*. Splashing metal—babbitting; *e*. Gases, fumes and liquids—acid handling; *f*. Excessive dust and flying particles—sand-blasting; *g*. Reflected light or glare—snow-covered ground; *h*. Injurious radiant energy of moderate intensity—oxy-acetylene welding; *i*. Injurious radiant energy of high intensity—electric welding. The code contains specifications for protectors for each type of hazard. The lenses of chipping goggles must, when placed horizontal, withstand 10 blows of a 16-gram steel ball falling 54 cm., or one blow from a height of 1 meter. Goggles used only for grinding, may be lighter. Those used for protection against dust or acids must fit the face closely, and in the latter case also must withstand corrosion. Lenses used in welding and similar operations must comply with certain specifications limiting the transmission of invisible as well as visible rays. For all types, strong and durable frames are specified.

Helmets are worn in some cases, instead of goggles. For exposure to poisonous gases, helmets or masks contain a filter like the army gas-mask, or an oxygen supply, or are connected with a hose leading to the outer air.

**Underwriters' Laboratories Label** is placed only on goggles or helmets complying with standard requirements.

**To Persuade Workmen** to wear goggles, provide a pair for each man, fitted to his face and to the spacing of his eyes and (if he regularly wears glasses) ground to his optical prescription. Education and discipline must do the rest.

**LEGGINGS** should be made of material which will prevent the molten metal or corrosive substances that are being handled from coming into contact with the leg of the wearer. They should make a tight enclosure when being worn. Leggings should be so fastened that they can be put on or taken off quickly. They always should be kept in good repair, otherwise a false sense of security may be established which will lead to the taking of unnecessary risks.

**PROTECTIVE SHOES** should be made of material which will not be easily affected by the substances being handled, should closely fit the contour of the foot and ankle, be so designed that liquid or molten material cannot enter the shoe; be easily and quickly removed from the foot and kept in good repair. Metal soles sometimes are used for protection against stepping on sharp objects; and rubber soles for electrical work.

**GLOVES** and arm protectors of proper type are a safeguard against cuts and scratches, in handling material; they should fit loosely so the hand may be withdrawn if the glove becomes caught. In machine work gloves often are a hazard as they may be caught and pull in the hand and arm; if worn at all, they should be hard finished, fit very loosely, and if possible be without fingers. Rubber gloves for electricians must be tested frequently, as a pin prick or worn place destroys their value.

**ALL CLOTHING** worn near machinery or fires should be free from loose ends. Loose or ragged sleeves or other garments, flowing ties or ribbons, jewelry including rings, and long hair unless confined by a cap are dangerous for both men and women machine operators. Aprons, if required, should be lightly secured to the body. One-piece overalls are favored.

## 10. FIRST AID

The immediate and proper care of injuries, both large and small, minimizes the resulting loss of time and often prevents permanent crippling or serious infection. The equipment and operation of a medical and surgical department such as is maintained by many large industries is beyond the scope of this book. First-aid treatment by fellow-employees who have received certain elementary training, is valuable wherever medical attention is not immediately available. Such treatment must be recognized as temporary only, and the injury should in every case be examined and re-dressed as soon as possible by a physician or graduate nurse, acting under medical supervision, either within or outside of the plant.

**INFECTION** of slight injuries causes many cases of prolonged disability, amputation, or death. Workmen should be strictly required to report all injuries, however slight, and have them dressed. First-aid stations at convenient points, with competent attendants, make it possible to enforce such a rule.

**FIRST-AID OUTFITS AND METHODS.**—Standard outfits may be obtained from medical supply dealers. An outfit for general use may contain iodine, aromatic spirits of ammonia, ampoules, toothpicks (as applicators), absorbent cotton, adhesive tape, scissors, roll and triangular bandages, sterile gauze, ointment, and instruction book. A smaller kit for minor injuries may include only iodine, applicators, and finger bandages. First-aid stations in mines and at other points, where a well-trained first-aid team is available, require splints, tourniquets, and other additional equipment. For further information on equipment, methods of treatment, and methods of training, consult the American Red Cross (Washington, D. C.) or U. S. Bureau of Mines (Washington, D. C.).

## 11. SUPERVISION AND EDUCATION

The accomplishments of those industrial concerns which have reduced their accidents by from 50 to 90%, with large money savings, have been brought about not alone by guarding obvious points of danger as described above but by a combination of: 1. Mechanical guarding; 2. Re-design of machinery, and revision of operating methods, which were inherently dangerous; 3. Strict supervision by executives, from manager to foremen, to enforce the use of safe methods of work; 4. Training of workmen, both in general habits of carefulness and in the safe conduct of their particular jobs.

These things are most easily accomplished through an organized effort, or continuous campaign, in which the ten necessary steps are outlined by the National Safety Council as follows: *a.* Secure co-operation of manager; *b.* Secure co-operation of superintendent; *c.* Appoint a safety engineer; *d.* Appoint a central safety committee; *e.* Secure co-operation of foreman and sub-foreman; *f.* Organize foremen's safety meetings; *g.* Appoint foremen's inspection committee; *h.* Hold safety mass meeting for all employees; *i.* Appoint workmen's safety committees; *j.* Start use of safety bulletin boards.

**THE MANAGER AND THE SUPERINTENDENT** must actively support the safety campaign and must be willing to authorize necessary expenditures.

**THE SAFETY ENGINEER OR INSPECTOR** either (1) is held directly responsible for all safety work, or (2) acts as expert adviser to the operating executives if they are held responsible. In a small plant he may give part time to other work. His duties include the following:

*a.* Check up unsafe practices, insanitary conditions, the use and need of safeguards, etc.; *b.* Keep in touch with foremen and secure their co-operation by serving them; *c.* Recommend methods of safeguarding and follow-up installations; *d.* Check all specifications and drawings for new equipment or repairs to see that provision is made for safety; *e.* Supervise bulletin board service; *f.* Supervise safety educational activities; *g.* Arrange programs for safety meetings; *h.* Investigate all serious accidents and near accidents; *i.* Keep records and prepare reports on accident experience.

**THE CENTRAL SAFETY COMMITTEE** composed of the superintendent (chairman), safety engineer (secretary), and such men as chief engineer, master mechanic, employment manager, purchasing agent, and department superintendents should head the safety organization and: *a.* Pass on controversial matters; *b.* Establish standards for safeguards; *c.* Review reports and recommendations of safety engineer, foremen, and workmen; *d.* Formulate safety rules; *e.* Outline educational methods and direct safety campaigns; *f.* Study accident experience tabulated by safety engineer.

**THE FOREMEN** determine the success or failure of safety work. They should: *a.* Instruct their men, especially new men; *b.* Enforce safety rules with wise discipline; *c.* See that tools and machines are kept in proper repair and adjustment; *d.* Inspect their departments and render weekly reports; *e.* Investigate accidents and near accidents and recommend methods of prevention; *f.* Keep closely in touch with and encourage the workmen's safety committees. Monthly foremen's safety meetings should take up inspection and progress reports, recommendations, and accident experience. In large plants, inspection by a foremen's committee will standardize practices in the different departments.

**WORKMEN'S SAFETY COMMITTEES** keep up the interest of workmen. They should: *a.* Make regular inspections to discover unsafe conditions and practices and submit to the foreman a written report of findings; *b.* Instruct and warn fellow workmen regarding dangerous practice; *c.* Investigate all serious accidents and near accidents and submit reports and recommendations.

**SAFETY BULLETIN BOARDS** containing pictures and stories which drive home what the workman can do to protect himself, are recognized as one of the most effective means of reaching the men. Bulletin boards should be placed at convenient points in each department. The secret of a live bulletin board is constant change. In addition to the National Safety Council safety bulletins, home-made bulletins should be used covering accidents or near accidents which have actually occurred in the plant, together with interesting exhibits, such as goggles which prevented eye injuries.

## 12. ORGANIZATIONS

The following is a list of organizations that concern themselves principally with the development of safe practices in industry, or which regard safe practice as an important, although not a major, activity. It is advisable to consult with one or more of these organizations in relation to any special construction or operation that is not covered by an existing safety code.

**National Safety Council** (20 North Wacker Drive, Chicago) is an association of manufacturing and other industrial concerns, insurance companies, engineers, public officials, and others; publishes *National Safety News* (magazine), workmen's bulletins or posters, "Safe Practices" pamphlets on a variety of special subjects; holds annual congress, maintains information bureau; fosters local safety councils; also publications and activities relating to public safety and industrial health.

**Safety Institute of America** (New York) publishes *Safety* (magazine), maintains a safety museum and special service to members.

**American Society of Safety Engineers** (Engineering Section of National Safety Council) is composed of individuals in safety work.

**American Standards Association** (New York) has general supervision of safety code work.

Formulation of specific codes, and other safety investigations have been undertaken by the organizations already named and by the U. S. Bureau of Standards (Washington, D. C.); U. S. Bureau of Mines (Washington, D. C.); Natl. Bureau of Casualty and Surety Underwriters (New York), maintained by stock insurance companies; National Assoc. of Mutual Casualty Companies (New York); American Society of Mechanical Engineers. Underwriters' Laboratories (Chicago) tests, inspects, and labels safety and fire-protection equipment conforming to its standards. Numerous trade associations have safety departments or safety committees.

# Section 28

# MATHEMATICAL TABLES

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# MATHEMATICAL TABLES

## LOGARITHMS

**Logarithms** (abbreviation *log*).—The log of a number is the exponent of the power to which it is necessary to raise a fixed number to produce the given number. The fixed number is called the *base*. Thus if the base is 10, the log of 1000 is 3, for $10^3 = 1000$. There are two systems of logs in general use, the *common*, in which the base is 10, and the Napierian, or *hyperbolic*, in which the base is 2.718281828. . . . The Napierian base is commonly denoted by $e$, as in the equation $e^y = x$, in which $y$ is the Napierian log of $x$. The abbreviation $\log_e$ is commonly used to denote the Napierian log.

In any system of logs, the log of 1 is 0; the log of the base, taken in that system, is 1. In any system the base of which is greater than 1, the logs of all numbers greater than 1 are positive and the logs of all numbers less than 1 are negative.

The modulus of any system is equal to the reciprocal of the Napierian log of the base of that system. The modulus of the Napierian system is 1, that of the common system is 0.4342945. The log of a number in any system equals the modulus of that system × Napierian log of the number. The *hyperbolic* or *Napierian* log of any number equals the common log × 2.3025851.

Every log consists of two parts, an integral part called the *characteristic*, or index, and the decimal part, or *mantissa*. The mantissa only is given in the usual tables of common logs, with the decimal point omitted. The characteristic is one less than the number of figures to the left of the decimal point in the number whose log is to be found. The characteristic of numbers from 1 to 9.99+ is 0, from 10 to 99.99+ is 1, from 100 to 999+ is 2, from 0.1 to 0.99+ is −1, from 0.01 to 0.099+ is −2, etc. Thus,

log of 2000 is 3.30103; log of 0.2 is −1.30103, or 9.30103 − 10
" " 200 " 2.30103; " " 0.02 " −2.30103, " 8.30103 − 10
" " 20 " 1.30103; " " 0.002 " −3.30103, " 7.30103 − 10
" " 2 " 0.30103; " " 0.0002 " −4.30103, " 6.30103 − 10

The minus sign is frequently written above the characteristic thus: log 0.002 = $\bar{3}.30103$. The characteristic only is negative, the decimal part, or mantissa, being always positive.

When a log consists of a negative index and a positive mantissa, it is usual to write the negative sign over the index, or to add 10 to the index, and indicate the subtraction of 10 from the resulting logarithm. Thus log 0.2 = $\bar{1}.30103$, may be written 9.30103 − 10. The difference between a logarithm and 10 is its *arithmetical complement* or **cologarithm.**

In tables of logarithmic sines, etc., the −10 is generally omitted, as being understood.

RULES FOR USE OF THE TABLE OF COMMON LOGARITHMS.—**To Find the Log of a Decimal Fraction or of a Whole Number and a Decimal.**—First find the log of the quantity as if there were no decimal point, then prefix the index according to rule; the index is one less than the number of figures to the left of the decimal point.

EXAMPLE. log of 3.14159. log of 3.141 = 0.497068. Diff. = 138.
From proportional parts 5 = 690
" " " 09 = 1242
log 3.14159 0.4971494

If the number is a decimal less than unity, the index is negative and is one more than the number of zeros to the right of the decimal point. Log of 0.0682 = $\bar{2}.833784$ = 8.833784 − 10.

**To Find the Number Corresponding to a Given Log.**—Find in the table the log nearest to the decimal part of the given log and take the first four digits of the required number from the column N and the top of the column containing the log which is the next less than the given log. To find the 5th and 6th digits subtract the log in the table from the given log, and multiply the difference by 100. and divide by the figure in the Diff. column opposite the log; annex the quotient to the four digits already found, and place the decimal point according to the rule; the number of figures to the left of the decimal point is one greater than the index. The number corresponding to a log is called the anti-logarithm.

Find the anti-log of.............................. 0.497150
Next lowest log in table corresponds to 3141........ $\underline{0.497068}$ Diff. = 82.
Tabular diff. = 138; $82 \div 138 = 0.59+$

The index being 0, the number is therefore 3.14159+.

**Multiplication by Means of Logarithms.**—Add together the logs of the two numbers to be multiplied. The sum is the log of the product.

*a*. Where both factors are greater than unity.

EXAMPLE. $31 \times 1274 = 39{,}494$.

*Solution.* $\log 31 + \log 1274 = 1.491362 + 3.105169 = 4.596531 = \log 39{,}494$.

*b*. Where one or more factors are less than unity, the logs with a negative characteristic can be handled most conveniently by adding and subtracting 10.

EXAMPLE. $.000028961 \times .084507 = .0000024474$.

*Solution.* $\log .000028961 + \log .084507 = \bar{5}.461814 + \bar{2}.926893$.
$= (5.461814 - 10) + (8.926893 - 10) = 14.388707 - 20 = \bar{6}.388707$
$= \log .0000024474$.

**Division by Means of Logarithms.**—Subtract the log of the divisor from the log of the dividend. The remainder is the log of the quotient.

*a*. When the divisor is smaller than the dividend.

EXAMPLE. $2987 \div 63 = 47.41284$.

*Solution.* $\log 2987 - \log 63 = 3.475235 - 1.799341 = 1.675894 = \log 47.41284$.

*b*. When the divisor is larger than the dividend, add and subtract as many tens as may be necessary to the log of the dividend and proceed as before.

EXAMPLE. $.000672 \div 263 = .00000255513$.

*Solution.* $\log .000672 - \log 263 = \bar{4}.827369 - 2.419956$
$= 6.827369 - 2.419956 - 10 = \bar{6}.407413 = \log .00000255513$.

*c*. The log of a fraction is obtained by subtracting the log of the denominator from the log of the numerator. Thus,

$$\log \frac{a}{b} = \log a - \log b.$$

**To Raise a Number to Any Given Power.**—Multiply the log of the number by the exponent of the number, and find the number whose log is the product.

*a*. Where the exponent consists of one figure.

EXAMPLE. $16.2^3 = 4251.528$.

*Solution.* $3 \times \log 16.2 = 3 \times 1.209515 = 3.628545 = \log 4251.528$.

*b*. Where the exponent consists of two or more figures, it is best to multiply the characteristic and mantissa separately.

EXAMPLE. $.005624^{.37} = .147067$.

*Solution.* $.37 \times \log .005624 = .37 \times \bar{3}.750045 = .37 \times (-3) + .37 \times .750045$
$= -1.11 + .277517 = \bar{1}.167517 = \log .147067$.

*c*. Where the number is a fraction, first find the log of the fraction and then multiply it by the exponent.

EXAMPLE. $\left(\frac{276}{.032}\right)^{.72} = 681.9396$.

*Solution.* $.72 (\log 276 - \log .032) = .72\{2.440909 - (8.505150 - 10)\}$
$= .72 \times 3.935759 = 2.833746 = \log 681.9396$.

**To Extract Any Root of a Number.**—Divide the log of the number by the index of the root, and find the number whose log is the quotient.

To extract the root of a decimal: *a*. When the root index is positive and evenly divisible into the negative characteristic of the log of the number, the division may be performed with the negative characteristic written in its usual place.

EXAMPLE. $\sqrt[4]{.0006954} = .16239$.

*Solution.* $\log .0006954 \div 4 = \bar{4}.842235 \div 4 = \bar{1}.210559 = \log .16239$.

*b*. When the root index is positive and not even divisible into the negative characteristic, add to the log of the number, and indicate the subtraction from it, the smallest integral multiple of the root which will eliminate the negative characteristic. Divide the result by the root index and ascertain the number whose log corresponds to the quotient.

EXAMPLE. $\sqrt[3.1]{.00002785} = .03393$.

*Solution.* $\log .00002785 = \bar{5}.444825$; $\bar{5}.444825 \div 3.1 = \{(2 \times 3.1) + \bar{5}.444825 - 2 \times 3.1\} \div 3.1$
$= \{1.644825 - 6.2\} \div 3.1 = \bar{2}.530589 = \log .03393$.

*c*. When the root index is negative, determine the excess of the negative characteristic over the positive mantissa. Divide the result by the root index and ascertain the number whose log corresponds to the quotient.

EXAMPLE. $\sqrt[-4]{.000003976} = 22.394$.

*Solution.* $\log .000003976 = \bar{6}.599446 = .599446 - 6 = -5.400554$;
$-5.400554 \div (-4) = 1.350138 = \log 22.394$.

**Solution of Exponential Equations.**—In an exponential equation, the unknown quantity is the exponent; thus $a^x = b$. This may be transformed to $\log a^x = \log b$, or $x \log a = \log b$, whence $x = \log b \div \log a$.

*a.* When the base is greater than unity, put the equation in the form $x = \log b \div \log a$. Then $\log x = \log (\log b) - \log (\log a)$.

EXAMPLE. $32.6^x = 14.632$.

*Solution.* $\log x = \log (\log 14.632) - \log (\log 32.6) = \log 1.165303 - \log 1.513218$
$= .066439 - .179901 = \bar{1}.886538;\ x = .77008.$

*b.* When both the known quantities are decimal, put the equation in the form $x = \log b \div \log a$. Subtract the positive mantissa from the negative characteristic in both divisor and dividend, obtaining negative remainders. Change the signs of divisor and dividend and proceed as in Case *a*.

EXAMPLE. $.0729^x = .2693$.

*Solution.* $x = \log .2693 \div \log .0729 = \bar{1}.430236 \div \bar{2}.862728$
$= (-.569764) \div (-1.137272) = .569764 \div 1.137272;$
$\log x = \log .569764 - \log 1.137272 = \bar{1}.755695 - .055864 = \bar{1}.699831$
$x = .50099.$

*c.* When only one of the known quantities is a decimal, put the equation in the form $x = \dfrac{\log b}{\log a}$. Subtract the positive mantissa from the negative characteristic of the numerator or denominator as the case may be, and rewrite the fraction with the remainder so obtained as the new numerator or denominator. Make both numerator and denominator positive, but write a minus sign in front of the fraction, to signify that the result will be a negative quantity. Solve the fraction by logarithms and write a minus sign in front of the result.

EXAMPLE. $.726^x = 802.7$.

$$\text{Solution.}\quad x = \frac{\log 802.7}{\log .726} = \frac{2.904553}{\bar{1}.860937} = \frac{2.904553}{-.139063} = -\frac{2.904553}{.139063}.$$

$\log x = -(\log 2.904553 - \log .139063) = (.463079 - \bar{1}.143211) = -(1.319868)$
$x = -20.8867$

*d.* When the exponent is negative and one of the known quantities is less than unity, put the equation in the form $(-x) = \dfrac{\log b}{\log a}$. Subtract the positive mantissa from the negative characteristic, as in Case *c*, and multiply both sides of the resulting equation by $(-1)$. Find the value of $x$ as in Case *c*.

EXAMPLE. $10.78^{-x} = .09431$.

$$\text{Solution.}\quad x = \frac{\log .09431}{\log 10.78} = \frac{\bar{2}.974558}{1.032619};\ x = \frac{1.025442}{1.032619}.$$

$\log x = .010911 - .013940 = \bar{1}.996971.\quad x = .99305.$

**Table 1. Logarithms of Numbers from 1 to 100**

| N | Log | N | Log | N | Log | N | Log | N | Log |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.000000 | 21 | 1.322219 | 41 | 1.612784 | 61 | 1.785330 | 81 | 1.908485 |
| 2 | 0.301030 | 22 | 1.342423 | 42 | 1.623249 | 62 | 1.792392 | 82 | 1.913814 |
| 3 | 0.477121 | 23 | 1.361728 | 43 | 1.633468 | 63 | 1.799341 | 83 | 1.919078 |
| 4 | 0.602060 | 24 | 1.380211 | 44 | 1.643453 | 64 | 1.806180 | 84 | 1.924279 |
| 5 | 0.698970 | 25 | 1.397940 | 45 | 1.653213 | 65 | 1.812913 | 85 | 1.929419 |
| 6 | 0.778151 | 26 | 1.414973 | 46 | 1.662758 | 66 | 1.819544 | 86 | 1.934498 |
| 7 | 0.845098 | 27 | 1.431364 | 47 | 1.672098 | 67 | 1.826075 | 87 | 1.939519 |
| 8 | 0.903090 | 28 | 1.447158 | 48 | 1.681241 | 68 | 1.832509 | 88 | 1.944483 |
| 9 | 0.954243 | 29 | 1.462398 | 49 | 1.690196 | 69 | 1.838849 | 89 | 1.949390 |
| 10 | 1.000000 | 30 | 1.477121 | 50 | 1.698970 | 70 | 1.845098 | 90 | 1.954243 |
| 11 | 1.041393 | 31 | 1.491362 | 51 | 1.707570 | 71 | 1.851258 | 91 | 1.959041 |
| 12 | 1.079181 | 32 | 1.505150 | 52 | 1.716003 | 72 | 1.857332 | 92 | 1.963788 |
| 13 | 1.113943 | 33 | 1.518514 | 53 | 1.724276 | 73 | 1.863323 | 93 | 1.968483 |
| 14 | 1.146128 | 34 | 1.531479 | 54 | 1.732394 | 74 | 1.869232 | 94 | 1.973128 |
| 15 | 1.176091 | 35 | 1.544068 | 55 | 1.740363 | 75 | 1.875061 | 95 | 1.977724 |
| 16 | 1.204120 | 36 | 1.556303 | 56 | 1.748188 | 76 | 1.880814 | 96 | 1.982271 |
| 17 | 1.230449 | 37 | 1.568202 | 57 | 1.755875 | 77 | 1.886491 | 97 | 1.986772 |
| 18 | 1.255273 | 38 | 1.579784 | 58 | 1.763428 | 78 | 1.892095 | 98 | 1.991226 |
| 19 | 1.278754 | 39 | 1.591065 | 59 | 1.770852 | 79 | 1.897627 | 99 | 1.995635 |
| 20 | 1.301030 | 40 | 1.602060 | 60 | 1.778151 | 80 | 1.903090 | 100 | 2.000000 |

See pp. 28–06 to 28–23 for a complete table of six-place logarithms.

### Table 2—Common Logarithms of Numbers

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **100** | **000000** | **000434** | **000868** | **001301** | **001734** | **002166** | **002598** | **003029** | **003461** | **003891** | 432 |
| 1 | 004321 | 004751 | 005181 | 005609 | 006038 | 006466 | 006894 | 007321 | 007748 | 008174 | 428 |
| 2 | 008600 | 009026 | 009451 | 009876 | 010300 | 010724 | 011147 | 011570 | 011993 | 012415 | 424 |
| 3 | 012837 | 013259 | 013680 | 014100 | 014521 | 014940 | 015360 | 015779 | 016197 | 016616 | 420 |
| 4 | 017033 | 017451 | 017868 | 018284 | 018700 | 019116 | 019532 | 019947 | 020361 | 020775 | 416 |
| 5 | 021189 | 021603 | 022016 | 022428 | 022841 | 023252 | 023664 | 024075 | 024486 | 024896 | 412 |
| 6 | 025306 | 025715 | 026125 | 026533 | 026942 | 027350 | 027757 | 028164 | 028571 | 028978 | 408 |
| 7 | 029384 | 029789 | 030195 | 030600 | 031004 | 031408 | 031812 | 032216 | 032619 | 033021 | 404 |
| 8 | 033424 | 033826 | 034227 | 034628 | 035029 | 035430 | 035830 | 036230 | 036629 | 037028 | 400 |
| 9 | 037426 | 037825 | 038223 | 038620 | 039017 | 039414 | 039811 | 040207 | 040602 | 040998 | 397 |
| **110** | **041393** | **041787** | **042182** | **042576** | **042969** | **043362** | **043755** | **044148** | **044540** | **044932** | 393 |
| 1 | 045323 | 045714 | 046105 | 046495 | 046885 | 047275 | 047664 | 048053 | 048442 | 048830 | 390 |
| 2 | 049218 | 049606 | 049993 | 050380 | 050766 | 051153 | 051538 | 051924 | 052309 | 052694 | 386 |
| 3 | 053078 | 053463 | 053846 | 054230 | 054613 | 054996 | 055378 | 055760 | 056142 | 056524 | 383 |
| 4 | 056905 | 057286 | 057666 | 058046 | 058426 | 058805 | 059185 | 059563 | 059942 | 060320 | 379 |
| 5 | 060698 | 061075 | 061452 | 061829 | 062206 | 062582 | 062958 | 063333 | 063709 | 064083 | 376 |
| 6 | 064458 | 064832 | 065206 | 065580 | 065953 | 066326 | 066699 | 067071 | 067443 | 067815 | 373 |
| 7 | 068186 | 068557 | 068928 | 069298 | 069668 | 070038 | 070407 | 070776 | 071145 | 071514 | 370 |
| 8 | 071882 | 072250 | 072617 | 072985 | 073352 | 073718 | 074085 | 074451 | 074816 | 075182 | 366 |
| 9 | 075547 | 075912 | 076276 | 076640 | 077004 | 077368 | 077731 | 078094 | 078457 | 078819 | 363 |
| **120** | **079181** | **079543** | **079904** | **080266** | **080626** | **080987** | **081347** | **081707** | **082067** | **082426** | 360 |
| 1 | 082785 | 083144 | 083503 | 083861 | 084219 | 084576 | 084934 | 085291 | 085647 | 086004 | 357 |
| 2 | 086360 | 086716 | 087071 | 087426 | 087781 | 088136 | 088490 | 088845 | 089198 | 089552 | 355 |
| 3 | 089905 | 090258 | 090611 | 090963 | 091315 | 091667 | 092018 | 092370 | 092721 | 093071 | 352 |
| 4 | 093422 | 093772 | 094122 | 094471 | 094820 | 095169 | 095518 | 095866 | 096215 | 096562 | 349 |
| 5 | 096910 | 097257 | 097604 | 097951 | 098298 | 098644 | 098990 | 099335 | 099681 | 100026 | 346 |
| 6 | 100371 | 100715 | 101059 | 101403 | 101747 | 102091 | 102434 | 102777 | 103119 | 103462 | 343 |
| 7 | 103804 | 104146 | 104487 | 104828 | 105169 | 105510 | 105851 | 106191 | 106531 | 106871 | 341 |
| 8 | 107210 | 107549 | 107888 | 108227 | 108565 | 108903 | 109241 | 109579 | 109916 | 110253 | 338 |
| 9 | 110590 | 110926 | 111263 | 111599 | 111934 | 112270 | 112605 | 112940 | 113275 | 113609 | 335 |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 434 | 43.4 | 86.8 | 130.2 | 173.6 | 217.0 | 260.4 | 303.8 | 347.2 | 390.6 |
| 432 | 43.2 | 86.4 | 129.6 | 172.8 | 216.0 | 259.2 | 302.4 | 345.6 | 388.8 |
| 430 | 43.0 | 86.0 | 129.0 | 172.0 | 215.0 | 258.0 | 301.0 | 344.0 | 387.0 |
| 428 | 42.8 | 85.6 | 128.4 | 171.2 | 214.0 | 256.8 | 299.6 | 342.4 | 385.2 |
| 426 | 42.6 | 85.2 | 127.8 | 170.4 | 213.0 | 255.6 | 298.2 | 340.8 | 383.4 |
| 424 | 42.4 | 84.8 | 127.2 | 169.6 | 212.0 | 254.4 | 296.8 | 339.2 | 381.6 |
| 422 | 42.2 | 84.4 | 126.6 | 168.8 | 211.0 | 253.2 | 295.4 | 337.6 | 379.8 |
| 420 | 42.0 | 84.0 | 126.0 | 168.0 | 210.0 | 252.0 | 294.0 | 336.0 | 378.0 |
| 418 | 41.8 | 83.6 | 125.4 | 167.2 | 209.0 | 250.8 | 292.6 | 334.4 | 376.2 |
| 416 | 41.6 | 83.2 | 124.8 | 166.4 | 208.0 | 249.6 | 291.2 | 332.8 | 374.4 |
| 414 | 41.4 | 82.8 | 124.2 | 165.6 | 207.0 | 248.4 | 289.8 | 331.2 | 372.6 |
| 412 | 41.2 | 82.4 | 123.6 | 164.8 | 206.0 | 247.2 | 288.4 | 329.6 | 370.8 |
| 410 | 41.0 | 82.0 | 123.0 | 164.0 | 205.0 | 246.0 | 287.0 | 328.0 | 369.0 |
| 408 | 40.8 | 81.6 | 122.4 | 163.2 | 204.0 | 244.8 | 285.6 | 326.4 | 367.2 |
| 406 | 40.6 | 81.2 | 121.8 | 162.4 | 203.0 | 243.6 | 284.2 | 324.8 | 365.4 |
| 404 | 40.4 | 80.8 | 121.2 | 161.6 | 202.0 | 242.4 | 282.8 | 323.2 | 363.6 |
| 402 | 40.2 | 80.4 | 120.6 | 160.8 | 201.0 | 241.2 | 281.4 | 321.6 | 361.8 |
| 400 | 40.0 | 80.0 | 120.0 | 160.0 | 200.0 | 240.0 | 280.0 | 320.0 | 360.0 |
| 398 | 39.8 | 79.6 | 119.4 | 159.2 | 199.0 | 238.8 | 278.6 | 318.4 | 358.2 |
| 396 | 39.6 | 79.2 | 118.8 | 158.4 | 198.0 | 237.6 | 277.2 | 316.8 | 356.4 |
| 394 | 39.4 | 78.8 | 118.2 | 157.6 | 197.0 | 236.4 | 275.8 | 315.2 | 354.6 |
| 392 | 39.2 | 78.4 | 117.6 | 156.8 | 196.0 | 235.2 | 274.4 | 313.6 | 352.8 |
| 390 | 39.0 | 78.0 | 117.0 | 156.0 | 195.0 | 234.0 | 273.0 | 312.0 | 351.0 |
| 388 | 38.8 | 77.6 | 116.4 | 155.2 | 194.0 | 232.8 | 271.6 | 310.4 | 349.2 |
| 386 | 38.6 | 77.2 | 115.8 | 154.4 | 193.0 | 231.6 | 270.2 | 308.8 | 347.4 |
| 384 | 38.4 | 76.8 | 115.2 | 153.6 | 192.0 | 230.4 | 268.8 | 307.2 | 345.6 |
| 382 | 38.2 | 76.4 | 114.6 | 152.8 | 191.0 | 229.2 | 267.4 | 305.6 | 343.8 |
| 380 | 38.0 | 76.0 | 114.0 | 152.0 | 190.0 | 228.0 | 266.0 | 304.0 | 342.0 |
| 378 | 37.8 | 75.6 | 113.4 | 151.2 | 189.0 | 226.8 | 264.6 | 302.4 | 340.2 |
| 376 | 37.6 | 75.2 | 112.8 | 150.4 | 188.0 | 225.6 | 263.2 | 300.8 | 338.4 |
| 374 | 37.4 | 74.8 | 112.2 | 149.6 | 187.0 | 224.4 | 261.8 | 299.2 | 336.6 |
| 372 | 37.2 | 74.4 | 111.6 | 148.8 | 186.0 | 223.2 | 260.4 | 297.6 | 334.8 |
| 370 | 37.0 | 74.0 | 111.0 | 148.0 | 185.0 | 222.0 | 259.0 | 296.0 | 333.0 |
| 368 | 36.8 | 73.6 | 110.4 | 147.2 | 184.0 | 220.8 | 257.6 | 294.4 | 331.2 |
| 366 | 36.6 | 73.2 | 109.8 | 146.4 | 183.0 | 219.6 | 256.2 | 292.8 | 329.4 |
| 364 | 36.4 | 72.8 | 109.2 | 145.6 | 182.0 | 218.4 | 254.8 | 291.2 | 327.6 |
| 362 | 36.2 | 72.4 | 108.6 | 144.8 | 181.0 | 217.2 | 253.4 | 289.6 | 325.8 |
| 360 | 36.0 | 72.0 | 108.0 | 144.0 | 180.0 | 216.0 | 252.0 | 288.0 | 324.0 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **130** | **113943** | **114277** | **114611** | **114944** | **115278** | **115611** | **115943** | **116276** | **116608** | **116940** | 333 |
| 1 | 117271 | 117603 | 117934 | 118265 | 118595 | 118926 | 119256 | 119586 | 119915 | 120245 | 330 |
| 2 | 120574 | 120903 | 121231 | 121560 | 121888 | 122216 | 122544 | 122871 | 123198 | 123525 | 328 |
| 3 | 123852 | 124178 | 124504 | 124830 | 125156 | 125481 | 125806 | 126131 | 126456 | 126781 | 325 |
| 4 | 127105 | 127429 | 127753 | 128076 | 128399 | 128722 | 129045 | 129368 | 129690 | 130012 | 323 |
| 5 | 130334 | 130655 | 130977 | 131298 | 131619 | 131939 | 132260 | 132580 | 132900 | 133219 | 321 |
| 6 | 133539 | 133858 | 134177 | 134496 | 134814 | 135133 | 135451 | 135769 | 136086 | 136403 | 318 |
| 7 | 136721 | 137037 | 137354 | 137671 | 137987 | 138303 | 138618 | 138934 | 139249 | 139564 | 316 |
| 8 | 139879 | 140194 | 140508 | 140822 | 141136 | 141450 | 141763 | 142076 | 142389 | 142702 | 314 |
| 9 | 143015 | 143327 | 143639 | 143951 | 144263 | 144574 | 144885 | 145196 | 145507 | 145818 | 311 |
| **140** | **146128** | **146438** | **146748** | **147058** | **147367** | **147676** | **147985** | **148294** | **148603** | **148911** | 309 |
| 1 | 149219 | 149527 | 149835 | 150142 | 150449 | 150756 | 151063 | 151370 | 151676 | 151982 | 307 |
| 2 | 152288 | 152594 | 152900 | 153205 | 153510 | 153815 | 154120 | 154424 | 154728 | 155032 | 305 |
| 3 | 155336 | 155640 | 155943 | 156246 | 156549 | 156852 | 157154 | 157457 | 157759 | 158061 | 303 |
| 4 | 158362 | 158664 | 158965 | 159266 | 159567 | 159868 | 160168 | 160469 | 160769 | 161068 | 301 |
| 5 | 161368 | 161667 | 161967 | 162266 | 162564 | 162863 | 163161 | 163460 | 163758 | 164055 | 299 |
| 6 | 164353 | 164650 | 164947 | 165244 | 165541 | 165838 | 166134 | 166430 | 166726 | 167022 | 297 |
| 7 | 167317 | 167613 | 167908 | 168203 | 168497 | 168792 | 169086 | 169380 | 169674 | 169968 | 295 |
| 8 | 170262 | 170555 | 170848 | 171141 | 171434 | 171726 | 172019 | 172311 | 172603 | 172895 | 293 |
| 9 | 173186 | 173478 | 173769 | 174060 | 174351 | 174641 | 174932 | 175222 | 175512 | 175802 | 291 |
| **150** | **176091** | **176381** | **176670** | **176959** | **177248** | **177536** | **177825** | **178113** | **178401** | **178689** | 289 |
| 1 | 178977 | 179264 | 179552 | 179839 | 180126 | 180413 | 180699 | 180986 | 181272 | 181558 | 287 |
| 2 | 181844 | 182129 | 182415 | 182700 | 182985 | 183270 | 183555 | 183839 | 184123 | 184407 | 285 |
| 3 | 184691 | 184975 | 185259 | 185542 | 185825 | 186108 | 186391 | 186674 | 186956 | 187239 | 283 |
| 4 | 187521 | 187803 | 188084 | 188366 | 188647 | 188928 | 189209 | 189490 | 189771 | 190051 | 281 |
| 5 | 190332 | 190612 | 190892 | 191171 | 191451 | 191730 | 192010 | 192289 | 192567 | 192846 | 279 |
| 6 | 193125 | 193403 | 193681 | 193959 | 194237 | 194514 | 194792 | 195069 | 195346 | 195623 | 278 |
| 7 | 195900 | 196176 | 196453 | 196729 | 197005 | 197281 | 197556 | 197832 | 198107 | 198382 | 276 |
| 8 | 198657 | 198932 | 199206 | 199481 | 199755 | 200029 | 200303 | 200577 | 200850 | 201124 | 274 |
| 9 | 201397 | 201670 | 201943 | 202216 | 202488 | 202761 | 203033 | 203305 | 203577 | 203848 | 272 |

PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 358 | 35.8 | 71.6 | 107.4 | 143.2 | 179.0 | 214.8 | 250.6 | 286.4 | 322.2 |
| 356 | 35.6 | 71.2 | 106.8 | 142.4 | 178.0 | 213.6 | 249.2 | 284.8 | 320.4 |
| 354 | 35.4 | 70.8 | 106.2 | 141.6 | 177.0 | 212.4 | 247.8 | 283.2 | 318.6 |
| 352 | 35.2 | 70.4 | 105.6 | 140.8 | 176.0 | 211.2 | 246.4 | 281.6 | 316.8 |
| 350 | 35.0 | 70.0 | 105.0 | 140.0 | 175.0 | 210.0 | 245.0 | 280.0 | 315.0 |
| 348 | 34.8 | 69.6 | 104.4 | 139.2 | 174.0 | 208.8 | 243.6 | 278.4 | 313.2 |
| 346 | 34.6 | 69.2 | 103.8 | 138.4 | 173.0 | 207.6 | 242.2 | 276.8 | 311.4 |
| 344 | 34.4 | 68.8 | 103.2 | 137.6 | 172.0 | 206.4 | 240.8 | 275.2 | 309.6 |
| 342 | 34.2 | 68.4 | 102.6 | 136.8 | 171.0 | 205.2 | 239.4 | 273.6 | 307.8 |
| 340 | 34.0 | 68.0 | 102.0 | 136.0 | 170.0 | 204.0 | 238.0 | 272.0 | 306.0 |
| 338 | 33.8 | 67.6 | 101.4 | 135.2 | 169.0 | 202.8 | 236.6 | 270.4 | 304.2 |
| 336 | 33.6 | 67.2 | 100.8 | 134.4 | 168.0 | 201.6 | 235.2 | 268.8 | 302.4 |
| 334 | 33.4 | 66.8 | 100.2 | 133.6 | 167.0 | 200.4 | 233.8 | 267.2 | 300.6 |
| 332 | 33.2 | 66.4 | 99.6 | 132.8 | 166.0 | 199.2 | 232.4 | 265.6 | 298.8 |
| 330 | 33.0 | 66.0 | 99.0 | 132.0 | 165.0 | 198.0 | 231.0 | 264.0 | 297.0 |
| 328 | 32.8 | 65.6 | 98.4 | 131.2 | 164.0 | 196.8 | 229.6 | 262.4 | 295.2 |
| 326 | 32.6 | 65.2 | 97.8 | 130.4 | 163.0 | 195.6 | 228.2 | 260.8 | 293.4 |
| 324 | 32.4 | 64.8 | 97.2 | 129.6 | 162.0 | 194.4 | 226.8 | 259.2 | 291.6 |
| 322 | 32.2 | 64.4 | 96.6 | 128.8 | 161.0 | 193.2 | 225.4 | 257.6 | 289.8 |
| 320 | 32.0 | 64.0 | 96.0 | 128.0 | 160.0 | 192.0 | 224.0 | 256.0 | 288.0 |
| 318 | 31.8 | 63.6 | 95.4 | 127.2 | 159.0 | 190.8 | 222.6 | 254.4 | 286.2 |
| 316 | 31.6 | 63.2 | 94.8 | 126.4 | 158.0 | 189.6 | 221.2 | 252.8 | 284.4 |
| 314 | 31.4 | 62.8 | 94.2 | 125.6 | 157.0 | 188.4 | 219.8 | 251.2 | 282.6 |
| 312 | 31.2 | 62.4 | 93.6 | 124.8 | 156.0 | 187.2 | 218.4 | 249.6 | 280.8 |
| 310 | 31.0 | 62.0 | 93.0 | 124.0 | 155.0 | 186.0 | 217.0 | 248.0 | 279.0 |
| 308 | 30.8 | 61.6 | 92.4 | 123.2 | 154.0 | 184.8 | 215.6 | 246.4 | 277.2 |
| 306 | 30.6 | 61.2 | 91.8 | 122.4 | 153.0 | 183.6 | 214.2 | 244.8 | 275.4 |
| 304 | 30.4 | 60.8 | 91.2 | 121.6 | 152.0 | 182.4 | 212.8 | 243.2 | 273.6 |
| 302 | 30.2 | 60.4 | 90.6 | 120.8 | 151.0 | 181.2 | 211.4 | 241.6 | 271.8 |
| 300 | 30.0 | 60.0 | 90.0 | 120.0 | 150.0 | 180.0 | 210.0 | 240.0 | 270.0 |
| 298 | 29.8 | 59.6 | 89.4 | 119.2 | 149.0 | 178.8 | 208.6 | 238.4 | 268.2 |
| 296 | 29.6 | 59.2 | 88.8 | 118.4 | 148.0 | 177.6 | 207.2 | 236.8 | 266.4 |
| 294 | 29.4 | 58.8 | 88.2 | 117.6 | 147.0 | 176.4 | 205.8 | 235.2 | 264.6 |
| 292 | 29.2 | 58.4 | 87.6 | 116.8 | 146.0 | 175.2 | 204.4 | 233.6 | 262.8 |
| 290 | 29.0 | 58.0 | 87.0 | 116.0 | 145.0 | 174.0 | 203.0 | 232.0 | 261.0 |
| 288 | 28.8 | 57.6 | 86.4 | 115.2 | 144.0 | 172.8 | 201.6 | 230.4 | 259.2 |
| 286 | 28.6 | 57.2 | 85.8 | 114.4 | 143.0 | 171.6 | 200.2 | 228.8 | 257.4 |
| 284 | 28.4 | 56.8 | 85.2 | 113.6 | 142.0 | 170.4 | 198.8 | 227.2 | 255.6 |
| 282 | 28.2 | 56.4 | 84.6 | 112.8 | 141.0 | 169.2 | 197.4 | 225.6 | 253.8 |
| 280 | 28.0 | 56.0 | 84.0 | 112.0 | 140.0 | 168.0 | 196.0 | 224.0 | 252.0 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **160** | **204120** | **204391** | **204663** | **204934** | **205204** | **205475** | **205746** | **206016** | **206286** | **206556** | 271 |
| 1 | 206826 | 207096 | 207365 | 207634 | 207904 | 208173 | 208441 | 208710 | 208979 | 209247 | 269 |
| 2 | 209515 | 209783 | 210051 | 210319 | 210586 | 210853 | 211121 | 211388 | 211654 | 211921 | 267 |
| 3 | 212188 | 212454 | 212720 | 212986 | 213252 | 213518 | 213783 | 214049 | 214314 | 214579 | 266 |
| 4 | 214844 | 215109 | 215373 | 215638 | 215902 | 216166 | 216430 | 216694 | 216957 | 217221 | 264 |
| 5 | 217484 | 217747 | 218010 | 218273 | 218536 | 218798 | 219060 | 219323 | 219585 | 219846 | 262 |
| 6 | 220108 | 220370 | 220631 | 220892 | 221153 | 221414 | 221675 | 221936 | 222196 | 222456 | 261 |
| 7 | 222716 | 222976 | 223236 | 223496 | 223755 | 224015 | 224274 | 224533 | 224792 | 225051 | 259 |
| 8 | 225309 | 225568 | 225826 | 226084 | 226342 | 226600 | 226858 | 227115 | 227372 | 227630 | 258 |
| 9 | 227887 | 228144 | 228400 | 228657 | 228913 | 229170 | 229426 | 229682 | 229938 | 230193 | 256 |
| **170** | **230449** | **230704** | **230960** | **231215** | **231470** | **231724** | **231979** | **232234** | **232488** | **232742** | 255 |
| 1 | 232996 | 233250 | 233504 | 233757 | 234011 | 234264 | 234517 | 234770 | 235023 | 235276 | 253 |
| 2 | 235528 | 235781 | 236033 | 236285 | 236537 | 236789 | 237041 | 237292 | 237544 | 237795 | 252 |
| 3 | 238046 | 238297 | 238548 | 238799 | 239049 | 239299 | 239550 | 239800 | 240050 | 240300 | 250 |
| 4 | 240549 | 240799 | 241048 | 241297 | 241546 | 241795 | 242044 | 242293 | 242541 | 242790 | 249 |
| 5 | 243038 | 243286 | 243534 | 243782 | 244030 | 244277 | 244525 | 244772 | 245019 | 245266 | 248 |
| 6 | 245513 | 245759 | 246006 | 246252 | 246499 | 246745 | 246991 | 247237 | 247482 | 247728 | 246 |
| 7 | 247973 | 248219 | 248464 | 248709 | 248954 | 249198 | 249443 | 249687 | 249932 | 250176 | 245 |
| 8 | 250420 | 250664 | 250908 | 251151 | 251395 | 251638 | 251881 | 252125 | 252368 | 252610 | 243 |
| 9 | 252853 | 253096 | 253338 | 253580 | 253822 | 254064 | 254306 | 254548 | 254790 | 255031 | 242 |
| **180** | **255273** | **255514** | **255755** | **255996** | **256237** | **256477** | **256718** | **256958** | **257198** | **257439** | 241 |
| 1 | 257679 | 257918 | 258158 | 258398 | 258637 | 258877 | 259116 | 259355 | 259594 | 259833 | 239 |
| 2 | 260071 | 260310 | 260548 | 260787 | 261025 | 261263 | 261501 | 261739 | 261976 | 262214 | 238 |
| 3 | 262451 | 262688 | 262925 | 263162 | 263399 | 263636 | 263873 | 264109 | 264346 | 264582 | 237 |
| 4 | 264818 | 265054 | 265290 | 265525 | 265761 | 265996 | 266232 | 266467 | 266702 | 266937 | 235 |
| 5 | 267172 | 267406 | 267641 | 267875 | 268110 | 268344 | 268578 | 268812 | 269046 | 269279 | 234 |
| 6 | 269513 | 269746 | 269980 | 270213 | 270446 | 270679 | 270912 | 271144 | 271377 | 271609 | 233 |
| 7 | 271842 | 272074 | 272306 | 272538 | 272770 | 273001 | 273233 | 273464 | 273696 | 273927 | 232 |
| 8 | 274158 | 274389 | 274620 | 274850 | 275081 | 275311 | 275542 | 275772 | 276002 | 276232 | 230 |
| 9 | 276462 | 276692 | 276921 | 277151 | 277380 | 277609 | 277838 | 278067 | 278296 | 278525 | 229 |
| **190** | **278754** | **278982** | **279211** | **279439** | **279667** | **279895** | **280123** | **280351** | **280578** | **280806** | 228 |
| 1 | 281033 | 281261 | 281488 | 281715 | 281942 | 282169 | 282396 | 282622 | 282849 | 283075 | 227 |
| 2 | 283301 | 283527 | 283753 | 283979 | 284205 | 284431 | 284656 | 284882 | 285107 | 285332 | 226 |
| 3 | 285557 | 285782 | 286007 | 286232 | 286456 | 286681 | 286905 | 287130 | 287354 | 287578 | 225 |
| 4 | 287802 | 288026 | 288249 | 288473 | 288696 | 288920 | 289143 | 289366 | 289589 | 289812 | 223 |
| 5 | 290035 | 290257 | 290480 | 290702 | 290925 | 291147 | 291369 | 291591 | 291813 | 292034 | 222 |
| 6 | 292256 | 292478 | 292699 | 292920 | 293141 | 293363 | 293584 | 293804 | 294025 | 294246 | 221 |
| 7 | 294466 | 294687 | 294907 | 295127 | 295347 | 295567 | 295787 | 296007 | 296226 | 296446 | 220 |
| 8 | 296665 | 296884 | 297104 | 297323 | 297542 | 297761 | 297979 | 298198 | 298416 | 298635 | 219 |
| 9 | 298853 | 299071 | 299289 | 299507 | 299725 | 299943 | 300161 | 300378 | 300595 | 300813 | 218 |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 278 | 27.8 | 55.6 | 83.4 | 111.2 | 139.0 | 166.8 | 194.6 | 222.4 | 250.2 |
| 276 | 27.6 | 55.2 | 82.8 | 110.4 | 138.0 | 165.6 | 193.2 | 220.8 | 248.4 |
| 274 | 27.4 | 54.8 | 82.2 | 109.6 | 137.0 | 164.4 | 191.8 | 219.2 | 246.6 |
| 272 | 27.2 | 54.4 | 81.6 | 108.8 | 136.0 | 163.2 | 190.4 | 217.6 | 244.8 |
| 270 | 27.0 | 54.0 | 81.0 | 108.0 | 135.0 | 162.0 | 189.0 | 216.0 | 243.0 |
| 268 | 26.8 | 53.6 | 80.4 | 107.2 | 134.0 | 160.8 | 187.6 | 214.4 | 241.2 |
| 266 | 26.6 | 53.2 | 79.8 | 106.4 | 133.0 | 159.6 | 186.2 | 212.8 | 239.4 |
| 264 | 26.4 | 52.8 | 79.2 | 105.6 | 132.0 | 158.4 | 184.8 | 211.2 | 237.6 |
| 262 | 26.2 | 52.4 | 78.6 | 104.8 | 131.0 | 157.2 | 183.4 | 209.6 | 235.8 |
| 260 | 26.0 | 52.0 | 78.0 | 104.0 | 130.0 | 156.0 | 182.0 | 208.0 | 234.0 |
| 258 | 25.8 | 51.6 | 77.4 | 103.2 | 129.0 | 154.8 | 180.6 | 206.4 | 232.2 |
| 256 | 25.6 | 51.2 | 76.8 | 102.4 | 128.0 | 153.6 | 179.2 | 204.8 | 230.4 |
| 254 | 25.4 | 50.8 | 76.2 | 101.6 | 127.0 | 152.4 | 177.8 | 203.2 | 228.6 |
| 252 | 25.2 | 50.4 | 75.6 | 100.8 | 126.0 | 151.2 | 176.4 | 201.6 | 226.8 |
| 250 | 25.0 | 50.0 | 75.0 | 100.0 | 125.0 | 150.0 | 175.0 | 200.0 | 225.0 |
| 248 | 24.8 | 49.6 | 74.4 | 99.2 | 124.0 | 148.8 | 173.6 | 198.4 | 223.2 |
| 246 | 24.6 | 49.2 | 73.8 | 98.4 | 123.0 | 147.6 | 172.2 | 196.8 | 221.4 |
| 244 | 24.4 | 48.8 | 73.2 | 97.6 | 122.0 | 146.4 | 170.8 | 195.2 | 219.6 |
| 242 | 24.2 | 48.4 | 72.6 | 96.8 | 121.0 | 145.2 | 169.4 | 193.6 | 217.8 |
| 240 | 24.0 | 48.0 | 72.0 | 96.0 | 120.0 | 144.0 | 168.0 | 192.0 | 216.0 |
| 238 | 23.8 | 47.6 | 71.4 | 95.2 | 119.0 | 142.8 | 166.6 | 190.4 | 214.2 |
| 236 | 23.6 | 47.2 | 70.8 | 94.4 | 118.0 | 141.6 | 165.2 | 188.8 | 212.4 |
| 234 | 23.4 | 46.8 | 70.2 | 93.6 | 117.0 | 140.4 | 163.8 | 187.2 | 210.6 |
| 232 | 23.2 | 46.4 | 69.6 | 92.8 | 116.0 | 139.2 | 162.4 | 185.6 | 208.8 |
| 230 | 23.0 | 46.0 | 69.0 | 92.0 | 115.0 | 138.0 | 161.0 | 184.0 | 207.0 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **200** | **301030** | **301247** | **301464** | **301681** | **301898** | **302114** | **302331** | **302547** | **302764** | **302980** | 217 |
| 1 | 303196 | 303412 | 303628 | 303844 | 304059 | 304275 | 304491 | 304706 | 304921 | 305136 | 216 |
| 2 | 305351 | 305566 | 305781 | 305996 | 306211 | 306425 | 306639 | 306854 | 307068 | 307282 | 215 |
| 3 | 307496 | 307710 | 307924 | 308137 | 308351 | 308564 | 308778 | 308991 | 309204 | 309417 | 213 |
| 4 | 309630 | 309843 | 310056 | 310268 | 310481 | 310693 | 310906 | 311118 | 311330 | 311542 | 212 |
| 5 | 311754 | 311966 | 312177 | 312389 | 312600 | 312812 | 313023 | 313234 | 313445 | 313656 | 211 |
| 6 | 313867 | 314078 | 314289 | 314499 | 314710 | 314920 | 315130 | 315340 | 315551 | 315760 | 210 |
| 7 | 315970 | 316180 | 316390 | 316599 | 316809 | 317018 | 317227 | 317436 | 317646 | 317854 | 209 |
| 8 | 318063 | 318272 | 318481 | 318689 | 318898 | 319106 | 319314 | 319522 | 319730 | 319938 | 208 |
| 9 | 320146 | 320354 | 320562 | 320769 | 320977 | 321184 | 321391 | 321598 | 321805 | 322012 | 207 |
| **210** | **322219** | **322426** | **322633** | **322839** | **323046** | **323252** | **323458** | **323665** | **323871** | **324077** | 206 |
| 1 | 324282 | 324488 | 324694 | 324899 | 325105 | 325310 | 325516 | 325721 | 325926 | 326131 | 205 |
| 2 | 326336 | 326541 | 326745 | 326950 | 327155 | 327359 | 327563 | 327767 | 327972 | 328176 | 204 |
| 3 | 328380 | 328583 | 328787 | 328991 | 329194 | 329398 | 329601 | 329805 | 330008 | 330211 | 203 |
| 4 | 330414 | 330617 | 330819 | 331022 | 331225 | 331427 | 331630 | 331832 | 332034 | 332236 | 202 |
| 5 | 332438 | 332640 | 332842 | 333044 | 333246 | 333447 | 333649 | 333850 | 334051 | 334253 | |
| 6 | 334454 | 334655 | 334856 | 335057 | 335257 | 335458 | 335658 | 335859 | 336059 | 336260 | 201 |
| 7 | 336460 | 336660 | 336860 | 337060 | 337260 | 337459 | 337659 | 337858 | 338058 | 338257 | 200 |
| 8 | 338456 | 338656 | 338855 | 339054 | 339253 | 339451 | 339650 | 339849 | 340047 | 340246 | 199 |
| 9 | 340444 | 340642 | 340841 | 341039 | 341237 | 341435 | 341632 | 341830 | 342028 | 342225 | 198 |
| **220** | **342423** | **342620** | **342817** | **343014** | **343212** | **343409** | **343606** | **343802** | **343999** | **344196** | 197 |
| 1 | 344392 | 344589 | 344785 | 344981 | 345178 | 345374 | 345570 | 345766 | 345962 | 346157 | 196 |
| 2 | 346353 | 346549 | 346744 | 346939 | 347135 | 347330 | 347525 | 347720 | 347915 | 348110 | 195 |
| 3 | 348305 | 348500 | 348694 | 348889 | 349083 | 349278 | 349472 | 349666 | 349860 | 350054 | 194 |
| 4 | 350248 | 350442 | 350636 | 350829 | 351023 | 351216 | 351410 | 351603 | 351796 | 351989 | 193 |
| 5 | 352183 | 352375 | 352568 | 352761 | 352954 | 353147 | 353339 | 353532 | 353724 | 353916 | |
| 6 | 354108 | 354301 | 354493 | 354685 | 354876 | 355068 | 355260 | 355452 | 355643 | 355834 | 192 |
| 7 | 356026 | 356217 | 356408 | 356599 | 356790 | 356981 | 357172 | 357363 | 357554 | 357744 | 191 |
| 8 | 357935 | 358125 | 358316 | 358506 | 358696 | 358886 | 359076 | 359266 | 359456 | 359646 | 190 |
| 9 | 359835 | 360025 | 360215 | 360404 | 360593 | 360783 | 360972 | 361161 | 361350 | 361539 | 189 |
| **230** | **361728** | **361917** | **362105** | **362294** | **362482** | **362671** | **362859** | **363048** | **363236** | **363424** | 188 |
| 1 | 363612 | 363800 | 363988 | 364176 | 364363 | 364551 | 364739 | 364926 | 365113 | 365301 | |
| 2 | 365488 | 365675 | 365862 | 366049 | 366236 | 366423 | 366610 | 366796 | 366983 | 367169 | 187 |
| 3 | 367356 | 367542 | 367729 | 367915 | 368101 | 368287 | 368473 | 368659 | 368845 | 369030 | 186 |
| 4 | 369216 | 369401 | 369587 | 369772 | 369958 | 370143 | 370328 | 370513 | 370698 | 370883 | 185 |
| 5 | 371068 | 371253 | 371437 | 371622 | 371806 | 371991 | 372175 | 372360 | 372544 | 372728 | 184 |
| 6 | 372912 | 373096 | 373280 | 373464 | 373647 | 373831 | 374015 | 374198 | 374382 | 374565 | |
| 7 | 374748 | 374932 | 375115 | 375298 | 375481 | 375664 | 375846 | 376029 | 376212 | 376394 | 183 |
| 8 | 376577 | 376759 | 376942 | 377124 | 377306 | 377488 | 377670 | 377852 | 378034 | 378216 | 182 |
| 9 | 378398 | 378580 | 378761 | 378943 | 379124 | 379306 | 379487 | 379668 | 379849 | 380030 | 181 |

## PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 228 | 22.8 | 45.6 | 68.4 | 91.2 | 114.0 | 136.8 | 159.6 | 182.4 | 205.2 |
| 226 | 22.6 | 45.2 | 67.8 | 90.4 | 113.0 | 135.6 | 158.2 | 180.8 | 203.4 |
| 224 | 22.4 | 44.8 | 67.2 | 89.6 | 112.0 | 134.4 | 156.8 | 179.2 | 201.6 |
| 222 | 22.2 | 44.4 | 66.6 | 88.8 | 111.0 | 133.2 | 155.4 | 177.6 | 199.8 |
| 220 | 22.0 | 44.0 | 66.0 | 88.0 | 110.0 | 132.0 | 154.0 | 176.0 | 198.0 |
| 218 | 21.8 | 43.6 | 65.4 | 87.2 | 109.0 | 130.8 | 152.6 | 174.4 | 196.2 |
| 216 | 21.6 | 43.2 | 64.8 | 86.4 | 108.0 | 129.6 | 151.2 | 172.8 | 194.4 |
| 214 | 21.4 | 42.8 | 64.2 | 85.6 | 107.0 | 128.4 | 149.8 | 171.2 | 192.6 |
| 212 | 21.2 | 42.4 | 63.6 | 84.8 | 106.0 | 127.2 | 148.4 | 169.6 | 190.8 |
| 210 | 21.0 | 42.0 | 63.0 | 84.0 | 105.0 | 126.0 | 147.0 | 168.0 | 189.0 |
| 208 | 20.8 | 41.6 | 62.4 | 83.2 | 104.0 | 124.8 | 145.6 | 166.4 | 187.2 |
| 206 | 20.6 | 41.2 | 61.8 | 82.4 | 103.0 | 123.6 | 144.2 | 164.8 | 185.4 |
| 204 | 20.4 | 40.8 | 61.2 | 81.6 | 102.0 | 122.4 | 142.8 | 163.2 | 183.6 |
| 202 | 20.2 | 40.4 | 60.6 | 80.8 | 101.0 | 121.2 | 141.4 | 161.6 | 181.8 |
| 200 | 20.0 | 40.0 | 60.0 | 80.0 | 100.0 | 120.0 | 140.0 | 160.0 | 180.0 |
| 198 | 19.8 | 39.6 | 59.4 | 79.2 | 99.0 | 118.8 | 138.6 | 158.4 | 178.2 |
| 196 | 19.6 | 39.2 | 58.8 | 78.4 | 98.0 | 117.6 | 137.2 | 156.8 | 176.4 |
| 194 | 19.4 | 38.8 | 58.2 | 77.6 | 97.0 | 116.4 | 135.8 | 155.2 | 174.6 |
| 192 | 19.2 | 38.4 | 57.6 | 76.8 | 96.0 | 115.2 | 134.4 | 153.6 | 172.8 |
| 190 | 19.0 | 38.0 | 57.0 | 76.0 | 95.0 | 114.0 | 133.0 | 152.0 | 171.0 |
| 188 | 18.8 | 37.6 | 56.4 | 75.2 | 94.0 | 112.8 | 131.6 | 150.4 | 169.2 |
| 186 | 18.6 | 37.2 | 55.8 | 74.4 | 93.0 | 111.6 | 130.2 | 148.8 | 167.4 |
| 184 | 18.4 | 36.8 | 55.2 | 73.6 | 92.0 | 110.4 | 128.8 | 147.2 | 165.6 |
| 182 | 18.2 | 36.4 | 54.6 | 72.8 | 91.0 | 109.2 | 127.4 | 145.6 | 163.8 |
| 180 | 18.0 | 36.0 | 54.0 | 72.0 | 90.0 | 108.0 | 126.0 | 144.0 | 162.0 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **240** | **380211** | **380392** | **380573** | **380754** | **380934** | **381115** | **381296** | **381476** | **381656** | **381837** | 181 |
| 1 | 382017 | 382197 | 382377 | 382557 | 382737 | 382917 | 383097 | 383277 | 383456 | 383636 | 180 |
| 2 | 383815 | 383995 | 384174 | 384353 | 384533 | 384712 | 384891 | 385070 | 385249 | 385428 | 179 |
| 3 | 385606 | 385785 | 385964 | 386142 | 386321 | 386499 | 386677 | 386856 | 387034 | 387212 | 178 |
| 4 | 387390 | 387568 | 387746 | 387924 | 388101 | 388279 | 388456 | 388634 | 388811 | 388989 | |
| 5 | 389166 | 389343 | 389520 | 389698 | 389875 | 390051 | 390228 | 390405 | 390582 | 390759 | 177 |
| 6 | 390935 | 391112 | 391288 | 391464 | 391641 | 391817 | 391993 | 392169 | 392345 | 392521 | 176 |
| 7 | 392697 | 392873 | 393048 | 393224 | 393400 | 393575 | 393751 | 393926 | 394101 | 394277 | |
| 8 | 394452 | 394627 | 394802 | 394977 | 395152 | 395326 | 395501 | 395676 | 395850 | 396025 | 175 |
| 9 | 396199 | 396374 | 396548 | 396722 | 396896 | 397071 | 397245 | 397419 | 397592 | 397766 | 174 |
| **250** | **397940** | **398114** | **398287** | **398461** | **398634** | **398808** | **398981** | **399154** | **399328** | **399501** | 173 |
| 1 | 399674 | 399847 | 400020 | 400192 | 400365 | 400538 | 400711 | 400883 | 401056 | 401228 | |
| 2 | 401401 | 401573 | 401745 | 401917 | 402089 | 402261 | 402433 | 402605 | 402777 | 402949 | 172 |
| 3 | 403121 | 403292 | 403464 | 403635 | 403807 | 403978 | 404149 | 404320 | 404492 | 404663 | 171 |
| 4 | 404834 | 405005 | 405176 | 405346 | 405517 | 405688 | 405858 | 406029 | 406199 | 406370 | |
| 5 | 406540 | 406710 | 406881 | 407051 | 407221 | 407391 | 407561 | 407731 | 407901 | 408070 | 170 |
| 6 | 408240 | 408410 | 408579 | 408749 | 408918 | 409087 | 409257 | 409426 | 409595 | 409764 | 169 |
| 7 | 409933 | 410102 | 410271 | 410440 | 410609 | 410777 | 410946 | 411114 | 411283 | 411451 | |
| 8 | 411620 | 411788 | 411956 | 412124 | 412293 | 412461 | 412629 | 412796 | 412964 | 413132 | 168 |
| 9 | 413300 | 413467 | 413635 | 413803 | 413970 | 414137 | 414305 | 414472 | 414639 | 414806 | 167 |
| **260** | **414973** | **415140** | **415307** | **415474** | **415641** | **415808** | **415974** | **416141** | **416308** | **416474** | |
| 1 | 416641 | 416807 | 416973 | 417139 | 417306 | 417472 | 417638 | 417804 | 417970 | 418135 | 166 |
| 2 | 418301 | 418467 | 418633 | 418798 | 418964 | 419129 | 419295 | 419460 | 419625 | 419791 | 165 |
| 3 | 419956 | 420121 | 420286 | 420451 | 420616 | 420781 | 420945 | 421110 | 421275 | 421439 | |
| 4 | 421604 | 421768 | 421933 | 422097 | 422261 | 422426 | 422590 | 422754 | 422918 | 423082 | 164 |
| 5 | 423246 | 423410 | 423574 | 423737 | 423901 | 424065 | 424228 | 424392 | 424555 | 424718 | |
| 6 | 424882 | 425045 | 425208 | 425371 | 425534 | 425697 | 425860 | 426023 | 426186 | 426349 | 163 |
| 7 | 426511 | 426674 | 426836 | 426999 | 427161 | 427324 | 427486 | 427648 | 427811 | 427973 | 162 |
| 8 | 428135 | 428297 | 428459 | 428621 | 428783 | 428944 | 429106 | 429268 | 429429 | 429591 | |
| 9 | 429752 | 429914 | 430075 | 430236 | 430398 | 430559 | 430720 | 430881 | 431042 | 431203 | 161 |
| **270** | **431364** | **431525** | **431685** | **431846** | **432007** | **432167** | **432328** | **432488** | **432649** | **432809** | |
| 1 | 432969 | 433130 | 433290 | 433450 | 433610 | 433770 | 433930 | 434090 | 434249 | 434409 | 160 |
| 2 | 434569 | 434729 | 434888 | 435048 | 435207 | 435367 | 435526 | 435685 | 435844 | 436004 | 159 |
| 3 | 436163 | 436322 | 436481 | 436640 | 436799 | 436957 | 437116 | 437275 | 437433 | 437592 | |
| 4 | 437751 | 437909 | 438067 | 438226 | 438384 | 438542 | 438701 | 438859 | 439017 | 439175 | 158 |
| 5 | 439333 | 439491 | 439648 | 439806 | 439964 | 440122 | 440279 | 440437 | 440594 | 440752 | |
| 6 | 440909 | 441066 | 441224 | 441381 | 441538 | 441695 | 441852 | 442009 | 442166 | 442323 | 157 |
| 7 | 442480 | 442637 | 442793 | 442950 | 443106 | 443263 | 443419 | 443576 | 443732 | 443889 | |
| 8 | 444045 | 444201 | 444357 | 444513 | 444669 | 444825 | 444981 | 445137 | 445293 | 445449 | 156 |
| 9 | 445604 | 445760 | 445915 | 446071 | 446226 | 446382 | 446537 | 446692 | 446848 | 447003 | 155 |
| **280** | **447158** | **447313** | **447468** | **447623** | **447778** | **447933** | **448088** | **448242** | **448397** | **448552** | |
| 1 | 448706 | 448861 | 449015 | 449170 | 449324 | 449478 | 449633 | 449787 | 449941 | 450095 | 154 |
| 2 | 450249 | 450403 | 450557 | 450711 | 450865 | 451018 | 451172 | 451326 | 451479 | 451633 | |
| 3 | 451786 | 451940 | 452093 | 452247 | 452400 | 452553 | 452706 | 452859 | 453012 | 453165 | 153 |
| 4 | 453318 | 453471 | 453624 | 453777 | 453930 | 454082 | 454235 | 454387 | 454540 | 454692 | |
| 5 | 454845 | 454997 | 455150 | 455302 | 455454 | 455606 | 455758 | 455910 | 456062 | 456214 | 152 |
| 6 | 456366 | 456518 | 456670 | 456821 | 456973 | 457125 | 457276 | 457428 | 457579 | 457731 | |
| 7 | 457882 | 458033 | 458184 | 458336 | 458487 | 458638 | 458789 | 458940 | 459091 | 459242 | 151 |
| 8 | 459392 | 459543 | 459694 | 459845 | 459995 | 460146 | 460296 | 460447 | 460597 | 460748 | |
| 9 | 460898 | 461048 | 461198 | 461348 | 461499 | 461649 | 461799 | 461948 | 462098 | 462248 | 150 |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 182 | 18.2 | 36.4 | 54.6 | 72.8 | 91.0 | 109.2 | 127.4 | 145.6 | 163.8 |
| 180 | 18.0 | 36.0 | 54.0 | 72.0 | 90.0 | 108.0 | 126.0 | 144.0 | 162.0 |
| 178 | 17.8 | 35.6 | 53.4 | 71.2 | 89.0 | 106.8 | 124.6 | 142.4 | 160.2 |
| 176 | 17.6 | 35.2 | 52.8 | 70.4 | 88.0 | 105.6 | 123.2 | 140.8 | 158.4 |
| 174 | 17.4 | 34.8 | 52.2 | 69.6 | 87.0 | 104.4 | 121.8 | 139.2 | 156.6 |
| 172 | 17.2 | 34.4 | 51.6 | 68.8 | 86.0 | 103.2 | 120.4 | 137.6 | 154.8 |
| 170 | 17.0 | 34.0 | 51.0 | 68.0 | 85.0 | 102.0 | 119.0 | 136.0 | 153.0 |
| 168 | 16.8 | 33.6 | 50.4 | 67.2 | 84.0 | 100.8 | 117.6 | 134.4 | 151.2 |
| 166 | 16.6 | 33.2 | 49.8 | 66.4 | 83.0 | 99.6 | 116.2 | 132.8 | 149.4 |
| 164 | 16.4 | 32.8 | 49.2 | 65.6 | 82.0 | 98.4 | 114.8 | 131.2 | 147.6 |
| 162 | 16.2 | 32.4 | 48.6 | 64.8 | 81.0 | 97.2 | 113.4 | 129.6 | 145.8 |
| 160 | 16.0 | 32.0 | 48.0 | 64.0 | 80.0 | 96.0 | 112.0 | 128.0 | 144.0 |
| 158 | 15.8 | 31.6 | 47.4 | 63.2 | 79.0 | 94.8 | 110.6 | 126.4 | 142.2 |
| 156 | 15.6 | 31.2 | 46.8 | 62.4 | 78.0 | 93.6 | 109.2 | 124.8 | 140.4 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **290** | **462398** | **462548** | **462697** | **462847** | **462997** | **463146** | **463296** | **463445** | **463594** | **463744** | |
| 1 | 463893 | 464042 | 464191 | 464340 | 464490 | 464639 | 464788 | 464936 | 465085 | 465234 | 149 |
| 2 | 465383 | 465532 | 465680 | 465829 | 465977 | 466126 | 466274 | 466423 | 466571 | 466719 | |
| 3 | 466868 | 467016 | 467164 | 467312 | 467460 | 467608 | 467756 | 467904 | 468052 | 468200 | 148 |
| 4 | 468347 | 468495 | 468643 | 468790 | 468938 | 469085 | 469233 | 469380 | 469527 | 469675 | |
| 5 | 469822 | 469969 | 470116 | 470263 | 470410 | 470557 | 470704 | 470851 | 470998 | 471145 | 147 |
| 6 | 471292 | 471438 | 471585 | 471732 | 471878 | 472025 | 472171 | 472318 | 472464 | 472610 | 146 |
| 7 | 472756 | 472903 | 473049 | 473195 | 473341 | 473487 | 473633 | 473779 | 473925 | 474071 | |
| 8 | 474216 | 474362 | 474508 | 474653 | 474799 | 474944 | 475090 | 475235 | 475381 | 475526 | |
| 9 | 475671 | 475816 | 475962 | 476107 | 476252 | 476397 | 476542 | 476687 | 476832 | 476976 | 145 |
| **300** | **477121** | **477266** | **477411** | **477555** | **477700** | **477844** | **477989** | **478133** | **478278** | **478422** | |
| 1 | 478566 | 478711 | 478855 | 478999 | 479143 | 479287 | 479431 | 479575 | 479719 | 479863 | 144 |
| 2 | 480007 | 480151 | 480294 | 480438 | 480582 | 480725 | 480869 | 481012 | 481156 | 481299 | |
| 3 | 481443 | 481586 | 481729 | 481872 | 482016 | 482159 | 482302 | 482445 | 482588 | 482731 | 143 |
| 4 | 482874 | 483016 | 483159 | 483302 | 483445 | 483587 | 483730 | 483872 | 484015 | 484157 | |
| 5 | 484300 | 484442 | 484585 | 484727 | 484869 | 485011 | 485153 | 485295 | 485437 | 485579 | 142 |
| 6 | 485721 | 485863 | 486005 | 486147 | 486289 | 486430 | 486572 | 486714 | 486855 | 486997 | |
| 7 | 487138 | 487280 | 487421 | 487563 | 487704 | 487845 | 487986 | 488127 | 488269 | 488410 | 141 |
| 8 | 488551 | 488692 | 488833 | 488974 | 489114 | 489255 | 489396 | 489537 | 489677 | 489818 | |
| 9 | 489958 | 490099 | 490239 | 490380 | 490520 | 490661 | 490801 | 490941 | 491081 | 491222 | 140 |
| **310** | **491362** | **491502** | **491642** | **491782** | **491922** | **492062** | **492201** | **492341** | **492481** | **492621** | |
| 1 | 492760 | 492900 | 493040 | 493179 | 493319 | 493458 | 493597 | 493737 | 493876 | 494015 | 139 |
| 2 | 494155 | 494294 | 494433 | 494572 | 494711 | 494850 | 494989 | 495128 | 495267 | 495406 | |
| 3 | 495544 | 495683 | 495822 | 495960 | 496099 | 496238 | 496376 | 496515 | 496653 | 496791 | |
| 4 | 496930 | 497068 | 497206 | 497344 | 497483 | 497621 | 497759 | 497897 | 498035 | 498173 | 138 |
| 5 | 498311 | 498448 | 498586 | 498724 | 498862 | 498999 | 499137 | 499275 | 499412 | 499550 | |
| 6 | 499687 | 499824 | 499962 | 500099 | 500236 | 500374 | 500511 | 500648 | 500785 | 500922 | 137 |
| 7 | 501059 | 501196 | 501333 | 501470 | 501607 | 501744 | 501880 | 502017 | 502154 | 502291 | |
| 8 | 502427 | 502564 | 502700 | 502837 | 502973 | 503109 | 503246 | 503382 | 503518 | 503655 | 136 |
| 9 | 503791 | 503927 | 504063 | 504199 | 504335 | 504471 | 504607 | 504743 | 504878 | 505014 | |
| **320** | **505150** | **505286** | **505421** | **505557** | **505693** | **505828** | **505964** | **506099** | **506234** | **506370** | |
| 1 | 506505 | 506640 | 506776 | 506911 | 507046 | 507181 | 507316 | 507451 | 507586 | 507721 | 135 |
| 2 | 507856 | 507991 | 508126 | 508260 | 508395 | 508530 | 508664 | 508799 | 508934 | 509068 | |
| 3 | 509203 | 509337 | 509471 | 509606 | 509740 | 509874 | 510009 | 510143 | 510277 | 510411 | 134 |
| 4 | 510545 | 510679 | 510813 | 510947 | 511081 | 511215 | 511349 | 511482 | 511616 | 511750 | |
| 5 | 511883 | 512017 | 512151 | 512284 | 512418 | 512551 | 512684 | 512818 | 512951 | 513084 | 133 |
| 6 | 513218 | 513351 | 513484 | 513617 | 513750 | 513883 | 514016 | 514149 | 514282 | 514415 | |
| 7 | 514548 | 514681 | 514813 | 514946 | 515079 | 515211 | 515344 | 515476 | 515609 | 515741 | |
| 8 | 515874 | 516006 | 516139 | 516271 | 516403 | 516535 | 516668 | 516800 | 516932 | 517064 | 132 |
| 9 | 517196 | 517328 | 517460 | 517592 | 517724 | 517855 | 517987 | 518119 | 518251 | 518382 | |
| **330** | **518514** | **518646** | **518777** | **518909** | **519040** | **519171** | **519303** | **519434** | **519566** | **519697** | 131 |
| 1 | 519828 | 519959 | 520090 | 520221 | 520353 | 520484 | 520615 | 520745 | 520876 | 521007 | |
| 2 | 521138 | 521269 | 521400 | 521530 | 521661 | 521792 | 521922 | 522053 | 522183 | 522314 | |
| 3 | 522444 | 522575 | 522705 | 522835 | 522966 | 523096 | 523226 | 523356 | 523486 | 523616 | 130 |
| 4 | 523746 | 523876 | 524006 | 524136 | 524266 | 524396 | 524526 | 524656 | 524785 | 524915 | |
| 5 | 525045 | 525174 | 525304 | 525434 | 525563 | 525693 | 525822 | 525951 | 526081 | 526210 | 129 |
| 6 | 526339 | 526469 | 526598 | 526727 | 526856 | 526985 | 527114 | 527243 | 527372 | 527501 | |
| 7 | 527630 | 527759 | 527888 | 528016 | 528145 | 528274 | 528402 | 528531 | 528660 | 528788 | |
| 8 | 528917 | 529045 | 529174 | 529302 | 529430 | 529559 | 529687 | 529815 | 529943 | 530072 | 128 |
| 9 | 530200 | 530328 | 530456 | 530584 | 530712 | 530840 | 530968 | 531096 | 531223 | 531351 | |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 154 | 15.4 | 30.8 | 46.2 | 61.6 | 77.0 | 92.4 | 107.8 | 123.2 | 138.6 |
| 152 | 15.2 | 30.4 | 45.6 | 60.8 | 76.0 | 91.2 | 106.4 | 121.6 | 136.8 |
| 150 | 15.0 | 30.0 | 45.0 | 60.0 | 75.0 | 90.0 | 105.0 | 120.0 | 135.0 |
| 148 | 14.8 | 29.6 | 44.4 | 59.2 | 74.0 | 88.8 | 103.6 | 118.4 | 133.2 |
| 146 | 14.6 | 29.2 | 43.8 | 58.4 | 73.0 | 87.6 | 102.2 | 116.8 | 131.4 |
| 144 | 14.4 | 28.8 | 43.2 | 57.6 | 72.0 | 86.4 | 100.8 | 115.2 | 129.6 |
| 142 | 14.2 | 28.4 | 42.6 | 56.8 | 71.0 | 85.2 | 99.4 | 113.6 | 127.8 |
| 140 | 14.0 | 28.0 | 42.0 | 56.0 | 70.0 | 84.0 | 98.0 | 112.0 | 126.0 |
| 138 | 13.8 | 27.6 | 41.4 | 55.2 | 69.0 | 82.8 | 96.6 | 110.4 | 124.2 |
| 136 | 13.6 | 27.2 | 40.8 | 54.4 | 68.0 | 81.6 | 95.2 | 108.8 | 122.4 |
| 134 | 13.4 | 26.8 | 40.2 | 53.6 | 67.0 | 80.4 | 93.8 | 107.2 | 120.6 |
| 132 | 13.2 | 26.4 | 39.6 | 52.8 | 66.0 | 79.2 | 92.4 | 105.6 | 118.8 |
| 130 | 13.0 | 26.0 | 39.0 | 52.0 | 65.0 | 78.0 | 91.0 | 104.0 | 117.0 |
| 128 | 12.8 | 25.6 | 38.4 | 51.2 | 64.0 | 76.8 | 89.6 | 102.4 | 115.2 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **340** | **531479** | **531607** | **531734** | **531862** | **531990** | **532117** | **532245** | **532372** | **532500** | **532627** | |
| 1 | 532754 | 532882 | 533009 | 533136 | 533264 | 533391 | 533518 | 533645 | 533772 | 533899 | 127 |
| 2 | 534026 | 534153 | 534280 | 534407 | 534534 | 534661 | 534787 | 534914 | 535041 | 535167 | |
| 3 | 535294 | 535421 | 535547 | 535674 | 535800 | 535927 | 536053 | 536180 | 536306 | 536432 | 126 |
| 4 | 536558 | 536685 | 536811 | 536937 | 537063 | 537189 | 537315 | 537441 | 537567 | 537693 | |
| 5 | 537819 | 537945 | 538071 | 538197 | 538322 | 538448 | 538574 | 538699 | 538825 | 538951 | |
| 6 | 539076 | 539202 | 539327 | 539452 | 539578 | 539703 | 539829 | 539954 | 540079 | 540204 | 125 |
| 7 | 540329 | 540455 | 540580 | 540705 | 540830 | 540955 | 541080 | 541205 | 541330 | 541454 | |
| 8 | 541579 | 541704 | 541829 | 541953 | 542078 | 542203 | 542327 | 542452 | 542576 | 542701 | |
| 9 | 542825 | 542950 | 543074 | 543199 | 543323 | 543447 | 543571 | 543696 | 543820 | 543944 | 124 |
| **350** | **544068** | **544192** | **544316** | **544440** | **544564** | **544688** | **544812** | **544936** | **545060** | **545183** | |
| 1 | 545307 | 545431 | 545555 | 545678 | 545802 | 545925 | 546049 | 546172 | 546296 | 546419 | |
| 2 | 546543 | 546666 | 546789 | 546913 | 547036 | 547159 | 547282 | 547405 | 547529 | 547652 | 123 |
| 3 | 547775 | 547898 | 548021 | 548144 | 548267 | 548389 | 548512 | 548635 | 548758 | 548881 | |
| 4 | 549003 | 549126 | 549249 | 549371 | 549494 | 549616 | 549739 | 549861 | 549984 | 550106 | |
| 5 | 550228 | 550351 | 550473 | 550595 | 550717 | 550840 | 550962 | 551084 | 551206 | 551328 | 122 |
| 6 | 551450 | 551572 | 551694 | 551816 | 551938 | 552060 | 552181 | 552303 | 552425 | 552547 | |
| 7 | 552668 | 552790 | 552911 | 553033 | 553155 | 553276 | 553398 | 553519 | 553640 | 553762 | 121 |
| 8 | 553883 | 554004 | 554126 | 554247 | 554368 | 554489 | 554610 | 554731 | 554852 | 554973 | |
| 9 | 555094 | 555215 | 555336 | 555457 | 555578 | 555699 | 555820 | 555940 | 556061 | 556182 | |
| **360** | **556303** | **556423** | **556544** | **556664** | **556785** | **556905** | **557026** | **557146** | **557267** | **557387** | 120 |
| 1 | 557507 | 557627 | 557748 | 557868 | 557988 | 558108 | 558228 | 558349 | 558469 | 558589 | |
| 2 | 558709 | 558829 | 558948 | 559068 | 559188 | 559308 | 559428 | 559548 | 559667 | 559787 | |
| 3 | 559907 | 560026 | 560146 | 560265 | 560385 | 560504 | 560624 | 560743 | 560863 | 560982 | 119 |
| 4 | 561101 | 561221 | 561340 | 561459 | 561578 | 561698 | 561817 | 561936 | 562055 | 562174 | |
| 5 | 562293 | 562412 | 562531 | 562650 | 562769 | 562887 | 563006 | 563125 | 563244 | 563362 | |
| 6 | 563481 | 563600 | 563718 | 563837 | 563955 | 564074 | 564192 | 564311 | 564429 | 564548 | |
| 7 | 564666 | 564784 | 564903 | 565021 | 565139 | 565257 | 565376 | 565494 | 565612 | 565730 | 118 |
| 8 | 565848 | 565966 | 566084 | 566202 | 566320 | 566437 | 566555 | 566673 | 566791 | 566909 | |
| 9 | 567026 | 567144 | 567262 | 567379 | 567497 | 567614 | 567732 | 567849 | 567967 | 568084 | |
| **370** | **568202** | **568319** | **568436** | **568554** | **568671** | **568788** | **568905** | **569023** | **569140** | **569257** | 117 |
| 1 | 569374 | 569491 | 569608 | 569725 | 569842 | 569959 | 570076 | 570193 | 570309 | 570426 | |
| 2 | 570543 | 570660 | 570776 | 570893 | 571010 | 571126 | 571243 | 571359 | 571476 | 571592 | |
| 3 | 571709 | 571825 | 571942 | 572058 | 572174 | 572291 | 572407 | 572523 | 572639 | 572755 | 116 |
| 4 | 572872 | 572988 | 573104 | 573220 | 573336 | 573452 | 573568 | 573684 | 573800 | 573915 | |
| 5 | 574031 | 574147 | 574263 | 574379 | 574494 | 574610 | 574726 | 574841 | 574957 | 575072 | |
| 6 | 575188 | 575303 | 575419 | 575534 | 575650 | 575765 | 575880 | 575996 | 576111 | 576226 | 115 |
| 7 | 576341 | 576457 | 576572 | 576687 | 576802 | 576917 | 577032 | 577147 | 577262 | 577377 | |
| 8 | 577492 | 577607 | 577722 | 577836 | 577951 | 578066 | 578181 | 578295 | 578410 | 578525 | |
| 9 | 578639 | 578754 | 578868 | 578983 | 579097 | 579212 | 579326 | 579441 | 579555 | 579669 | 114 |
| **380** | **579784** | **579898** | **580012** | **580126** | **580241** | **580355** | **580469** | **580583** | **580697** | **580811** | |
| 1 | 580925 | 581039 | 581153 | 581267 | 581381 | 581495 | 581608 | 581722 | 581836 | 581950 | |
| 2 | 582063 | 582177 | 582291 | 582404 | 582518 | 582631 | 582745 | 582858 | 582972 | 583085 | |
| 3 | 583199 | 583312 | 583426 | 583539 | 583652 | 583765 | 583879 | 583992 | 584105 | 584218 | |
| 4 | 584331 | 584444 | 584557 | 584670 | 584783 | 584896 | 585009 | 585122 | 585235 | 585348 | 113 |
| 5 | 585461 | 585574 | 585686 | 585799 | 585912 | 586024 | 586137 | 586250 | 586362 | 586475 | |
| 6 | 586587 | 586700 | 586812 | 586925 | 587037 | 587149 | 587262 | 587374 | 587486 | 587599 | |
| 7 | 587711 | 587823 | 587935 | 588047 | 588160 | 588272 | 588384 | 588496 | 588608 | 588720 | 112 |
| 8 | 588832 | 588944 | 589056 | 589167 | 589279 | 589391 | 589503 | 589615 | 589726 | 589838 | |
| 9 | 589950 | 590061 | 590173 | 590284 | 590396 | 590507 | 590619 | 590730 | 590842 | 590953 | |

## PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 128 | 12.8 | 25.6 | 38.4 | 51.2 | 64.0 | 76.8 | 89.6 | 102.4 | 115.2 |
| 126 | 12.6 | 25.2 | 37.8 | 50.4 | 63.0 | 75.6 | 88.2 | 100.8 | 113.4 |
| 124 | 12.4 | 24.8 | 37.2 | 49.6 | 62.0 | 74.4 | 86.8 | 99.2 | 111.6 |
| 122 | 12.2 | 24.4 | 36.6 | 48.8 | 61.0 | 73.2 | 85.4 | 97.6 | 109.8 |
| 120 | 12.0 | 24.0 | 36.0 | 48.0 | 60.0 | 72.0 | 84.0 | 96.0 | 108.0 |
| 118 | 11.8 | 23.6 | 35.4 | 47.2 | 59.0 | 70.8 | 82.6 | 94.4 | 106.2 |
| 116 | 11.6 | 23.2 | 34.8 | 46.4 | 58.0 | 69.6 | 81.2 | 92.8 | 104.4 |
| 114 | 11.4 | 22.8 | 34.2 | 45.6 | 57.0 | 68.4 | 79.8 | 91.2 | 102.6 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **390** | **591065** | **591176** | **591287** | **591399** | **591510** | **591621** | **591732** | **591843** | **591955** | **592066** | |
| 1 | 592177 | 592288 | 592399 | 592510 | 592621 | 592732 | 592843 | 592954 | 593064 | 593175 | 111 |
| 2 | 593286 | 593397 | 593508 | 593618 | 593729 | 593840 | 593950 | 594061 | 594171 | 594282 | |
| 3 | 594393 | 594503 | 594614 | 594724 | 594834 | 594945 | 595055 | 595165 | 595276 | 595386 | |
| 4 | 595496 | 595606 | 595717 | 595827 | 595937 | 596047 | 596157 | 596267 | 596377 | 596487 | |
| 5 | 596597 | 596707 | 596817 | 596927 | 597037 | 597146 | 597256 | 597366 | 597476 | 597586 | 110 |
| 6 | 597695 | 597805 | 597914 | 598024 | 598134 | 598243 | 598353 | 598462 | 598572 | 598681 | |
| 7 | 598791 | 598900 | 599009 | 599119 | 599228 | 599337 | 599446 | 599556 | 599665 | 599774 | |
| 8 | 599883 | 599992 | 600101 | 600210 | 600319 | 600428 | 600537 | 600646 | 600755 | 600864 | 109 |
| 9 | 600973 | 601082 | 601191 | 601299 | 601408 | 601517 | 601625 | 601734 | 601843 | 601951 | |
| **400** | **602060** | **602169** | **602277** | **602386** | **602494** | **602603** | **602711** | **602819** | **602928** | **603036** | |
| 1 | 603144 | 603253 | 603361 | 603469 | 603577 | 603686 | 603794 | 603902 | 604010 | 604118 | 108 |
| 2 | 604226 | 604334 | 604442 | 604550 | 604658 | 604766 | 604874 | 604982 | 605089 | 605197 | |
| 3 | 605305 | 605413 | 605521 | 605628 | 605736 | 605844 | 605951 | 606059 | 606166 | 606274 | |
| 4 | 606381 | 606489 | 606596 | 606704 | 606811 | 606919 | 607026 | 607133 | 607241 | 607348 | |
| 5 | 607455 | 607562 | 607669 | 607777 | 607884 | 607991 | 608098 | 608205 | 608312 | 608419 | 107 |
| 6 | 608526 | 608633 | 608740 | 608847 | 608954 | 609061 | 609167 | 609274 | 609381 | 609488 | |
| 7 | 609594 | 609701 | 609808 | 609914 | 610021 | 610128 | 610234 | 610341 | 610447 | 610554 | |
| 8 | 610660 | 610767 | 610873 | 610979 | 611086 | 611192 | 611298 | 611405 | 611511 | 611617 | |
| 9 | 611723 | 611829 | 611936 | 612042 | 612148 | 612254 | 612360 | 612466 | 612572 | 612678 | 106 |
| **410** | **612784** | **612890** | **612996** | **613102** | **613207** | **613313** | **613419** | **613525** | **613630** | **613736** | |
| 1 | 613842 | 613947 | 614053 | 614159 | 614264 | 614370 | 614475 | 614581 | 614686 | 614792 | |
| 2 | 614897 | 615003 | 615108 | 615213 | 615319 | 615424 | 615529 | 615634 | 615740 | 615845 | |
| 3 | 615950 | 616055 | 616160 | 616265 | 616370 | 616476 | 616581 | 616686 | 616790 | 616895 | 105 |
| 4 | 617000 | 617105 | 617210 | 617315 | 617420 | 617525 | 617629 | 617734 | 617839 | 617943 | |
| 5 | 618048 | 618153 | 618257 | 618362 | 618466 | 618571 | 618676 | 618780 | 618884 | 618989 | |
| 6 | 619093 | 619198 | 619302 | 619406 | 619511 | 619615 | 619719 | 619824 | 619928 | 620032 | |
| 7 | 620136 | 620240 | 620344 | 620448 | 620552 | 620656 | 620760 | 620864 | 620968 | 621072 | 104 |
| 8 | 621176 | 621280 | 621384 | 621488 | 621592 | 621695 | 621799 | 621903 | 622007 | 622110 | |
| 9 | 622214 | 622318 | 622421 | 622525 | 622628 | 622732 | 622835 | 622939 | 623042 | 623146 | |
| **420** | **623249** | **623353** | **623456** | **623559** | **623663** | **623766** | **623869** | **623973** | **624076** | **624179** | |
| 1 | 624282 | 624385 | 624488 | 624591 | 624695 | 624798 | 624901 | 625004 | 625107 | 625210 | 103 |
| 2 | 625312 | 625415 | 625518 | 625621 | 625724 | 625827 | 625929 | 626032 | 626135 | 626238 | |
| 3 | 626340 | 626443 | 626546 | 626648 | 626751 | 626853 | 626956 | 627058 | 627161 | 627263 | |
| 4 | 627366 | 627468 | 627571 | 627673 | 627775 | 627878 | 627980 | 628082 | 628185 | 628287 | |
| 5 | 628389 | 628491 | 628593 | 628695 | 628797 | 628900 | 629002 | 629104 | 629206 | 629308 | 102 |
| 6 | 629410 | 629512 | 629613 | 629715 | 629817 | 629919 | 630021 | 630123 | 630224 | 630326 | |
| 7 | 630428 | 630530 | 630631 | 630733 | 630835 | 630936 | 631038 | 631139 | 631241 | 631342 | |
| 8 | 631444 | 631545 | 631647 | 631748 | 631849 | 631951 | 632052 | 632153 | 632255 | 632356 | 101 |
| 9 | 632457 | 632559 | 632660 | 632761 | 632862 | 632963 | 633064 | 633165 | 633266 | 633367 | |
| **430** | **633468** | **633569** | **633670** | **633771** | **633872** | **633973** | **634074** | **634175** | **634276** | **634376** | |
| 1 | 634477 | 634578 | 634679 | 634779 | 634880 | 634981 | 635081 | 635182 | 635283 | 635383 | |
| 2 | 635484 | 635584 | 635685 | 635785 | 635886 | 635986 | 636087 | 636187 | 636287 | 636388 | |
| 3 | 636488 | 636588 | 636688 | 636789 | 636889 | 636989 | 637089 | 637189 | 637290 | 637390 | |
| 4 | 637490 | 637590 | 637690 | 637790 | 637890 | 637990 | 638090 | 638190 | 638290 | 638389 | 100 |
| 5 | 638489 | 638589 | 638689 | 638789 | 638888 | 638988 | 639088 | 639188 | 639287 | 639387 | |
| 6 | 639486 | 639586 | 639686 | 639785 | 639885 | 639984 | 640084 | 640183 | 640283 | 640382 | |
| 7 | 640481 | 640581 | 640680 | 640779 | 640879 | 640978 | 641077 | 641177 | 641276 | 641375 | |
| 8 | 641474 | 641573 | 641672 | 641771 | 641871 | 641970 | 642069 | 642168 | 642267 | 642366 | |
| 9 | 642465 | 642563 | 642662 | 642761 | 642860 | 642959 | 643058 | 643156 | 643255 | 643354 | 99 |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 112 | 11.2 | 22.4 | 33.6 | 44.8 | 56.0 | 67.2 | 78.4 | 89.6 | 100.8 |
| 110 | 11.0 | 22.0 | 33.0 | 44.0 | 55.0 | 66.0 | 77.0 | 88.0 | 99.0 |
| 108 | 10.8 | 21.6 | 32.4 | 43.2 | 54.0 | 64.8 | 75.6 | 86.4 | 97.2 |
| 106 | 10.6 | 21.2 | 31.8 | 42.4 | 53.0 | 63.6 | 74.2 | 84.8 | 95.4 |
| 104 | 10.4 | 20.8 | 31.2 | 41.6 | 52.0 | 62.4 | 72.8 | 83.2 | 93.6 |
| 102 | 10.2 | 20.4 | 30.6 | 40.8 | 51.0 | 61.2 | 71.4 | 81.6 | 91.8 |
| 100 | 10.0 | 20.0 | 30.0 | 40.0 | 50.0 | 60.0 | 70.0 | 80.0 | 90.0 |
| 98 | 9.8 | 19.6 | 29.4 | 39.2 | 49.0 | 58.8 | 68.6 | 78.4 | 88.2 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **440** | **643453** | **643551** | **643650** | **643749** | **643847** | **643946** | **644044** | **644143** | **644242** | **644340** | |
| 1 | 644439 | 644537 | 644636 | 644734 | 644832 | 644931 | 645029 | 645127 | 645226 | 645324 | |
| 2 | 645422 | 645521 | 645619 | 645717 | 645815 | 645913 | 646011 | 646110 | 646208 | 646306 | |
| 3 | 646404 | 646502 | 646600 | 646698 | 646796 | 646894 | 646992 | 647089 | 647187 | 647285 | 98 |
| 4 | 647383 | 647481 | 647579 | 647676 | 647774 | 647872 | 647969 | 648067 | 648165 | 648262 | |
| 5 | 648360 | 648458 | 648555 | 648653 | 648750 | 648848 | 648945 | 649043 | 649140 | 649237 | |
| 6 | 649335 | 649432 | 649530 | 649627 | 649724 | 649821 | 649919 | 650016 | 650113 | 650210 | |
| 7 | 650308 | 650405 | 650502 | 650599 | 650696 | 650793 | 650890 | 650987 | 651084 | 651181 | |
| 8 | 651278 | 651375 | 651472 | 651569 | 651666 | 651762 | 651859 | 651956 | 652053 | 652150 | 97 |
| 9 | 652246 | 652343 | 652440 | 652536 | 652633 | 652730 | 652826 | 652923 | 653019 | 653116 | |
| **450** | **653213** | **653309** | **653405** | **653502** | **653598** | **653695** | **653791** | **653888** | **653984** | **654080** | |
| 1 | 654177 | 654273 | 654369 | 654465 | 654562 | 654658 | 654754 | 654850 | 654946 | 655042 | |
| 2 | 655138 | 655235 | 655331 | 655427 | 655523 | 655619 | 655715 | 655810 | 655906 | 656002 | 96 |
| 3 | 656098 | 656194 | 656290 | 656386 | 656482 | 656577 | 656673 | 656769 | 656864 | 656960 | |
| 4 | 657056 | 657152 | 657247 | 657343 | 657438 | 657534 | 657629 | 657725 | 657820 | 657916 | |
| 5 | 658011 | 658107 | 658202 | 658298 | 658393 | 658488 | 658584 | 658679 | 658774 | 658870 | |
| 6 | 658965 | 659060 | 659155 | 659250 | 659346 | 659441 | 659536 | 659631 | 659726 | 659821 | |
| 7 | 659916 | 660011 | 660106 | 660201 | 660296 | 660391 | 660486 | 660581 | 660676 | 660771 | 95 |
| 8 | 660865 | 660960 | 661055 | 661150 | 661245 | 661339 | 661434 | 661529 | 661623 | 661718 | |
| 9 | 661813 | 661907 | 662002 | 662096 | 662191 | 662286 | 662380 | 662475 | 662569 | 662663 | |
| **460** | **662758** | **662852** | **662947** | **663041** | **663135** | **663230** | **663324** | **663418** | **663512** | **663607** | |
| 1 | 663701 | 663795 | 663889 | 663983 | 664078 | 664172 | 664266 | 664360 | 664454 | 664548 | |
| 2 | 664642 | 664736 | 664830 | 664924 | 665018 | 665112 | 665206 | 665299 | 665393 | 665487 | 94 |
| 3 | 665581 | 665675 | 665769 | 665862 | 665956 | 666050 | 666143 | 666237 | 666331 | 666424 | |
| 4 | 666518 | 666612 | 666705 | 666799 | 666892 | 666986 | 667079 | 667173 | 667266 | 667360 | |
| 5 | 667453 | 667546 | 667640 | 667733 | 667826 | 667920 | 668013 | 668106 | 668199 | 668293 | |
| 6 | 668386 | 668479 | 668572 | 668665 | 668759 | 668852 | 668945 | 669038 | 669131 | 669224 | |
| 7 | 669317 | 669410 | 669503 | 669596 | 669689 | 669782 | 669875 | 669967 | 670060 | 670153 | 93 |
| 8 | 670246 | 670339 | 670431 | 670524 | 670617 | 670710 | 670802 | 670895 | 670988 | 671080 | |
| 9 | 671173 | 671265 | 671358 | 671451 | 671543 | 671636 | 671728 | 671821 | 671913 | 672005 | |
| **470** | **672098** | **672190** | **672283** | **672375** | **672467** | **672560** | **672652** | **672744** | **672836** | **672929** | |
| 1 | 673021 | 673113 | 673205 | 673297 | 673390 | 673482 | 673574 | 673666 | 673758 | 673850 | |
| 2 | 673942 | 674034 | 674126 | 674218 | 674310 | 674402 | 674494 | 674586 | 674677 | 674769 | 92 |
| 3 | 674861 | 674953 | 675045 | 675137 | 675228 | 675320 | 675412 | 675503 | 675595 | 675687 | |
| 4 | 675778 | 675870 | 675962 | 676053 | 676145 | 676236 | 676328 | 676419 | 676511 | 676602 | |
| 5 | 676694 | 676785 | 676876 | 676968 | 677059 | 677151 | 677242 | 677333 | 677424 | 677516 | |
| 6 | 677607 | 677698 | 677789 | 677881 | 677972 | 678063 | 678154 | 678245 | 678336 | 678427 | |
| 7 | 678518 | 678609 | 678700 | 678791 | 678882 | 678973 | 679064 | 679155 | 679246 | 679337 | 91 |
| 8 | 679428 | 679519 | 679610 | 679700 | 679791 | 679882 | 679973 | 680063 | 680154 | 680245 | |
| 9 | 680336 | 680426 | 680517 | 680607 | 680698 | 680789 | 680879 | 680970 | 681060 | 681151 | |
| **480** | **681241** | **681332** | **681422** | **681513** | **681603** | **681693** | **681784** | **681874** | **681964** | **682055** | |
| 1 | 682145 | 682235 | 682326 | 682416 | 682506 | 682596 | 682686 | 682777 | 682867 | 682957 | |
| 2 | 683047 | 683137 | 683227 | 683317 | 683407 | 683497 | 683587 | 683677 | 683767 | 683857 | 90 |
| 3 | 683947 | 684037 | 684127 | 684217 | 684307 | 684396 | 684486 | 684576 | 684666 | 684756 | |
| 4 | 684845 | 684935 | 685025 | 685114 | 685204 | 685294 | 685383 | 685473 | 685563 | 685652 | |
| 5 | 685742 | 685831 | 685921 | 686010 | 686100 | 686189 | 686279 | 686368 | 686458 | 686547 | |
| 6 | 686636 | 686726 | 686815 | 686904 | 686994 | 687083 | 687172 | 687261 | 687351 | 687440 | |
| 7 | 687529 | 687618 | 687707 | 687796 | 687886 | 687975 | 688064 | 688153 | 688242 | 688331 | |
| 8 | 688420 | 688509 | 688598 | 688687 | 688776 | 688865 | 688953 | 689042 | 689131 | 689220 | 89 |
| 9 | 689309 | 689398 | 689486 | 689575 | 689664 | 689753 | 689841 | 689930 | 690019 | 690107 | |
| **490** | **690196** | **690285** | **690373** | **690462** | **690550** | **690639** | **690728** | **690816** | **690905** | **690993** | |
| 1 | 691081 | 691170 | 691258 | 691347 | 691435 | 691524 | 691612 | 691700 | 691789 | 691877 | |
| 2 | 691965 | 692053 | 692142 | 692230 | 692318 | 692406 | 692494 | 692583 | 692671 | 692759 | |
| 3 | 692847 | 692935 | 693023 | 693111 | 693199 | 693287 | 693375 | 693463 | 693551 | 693639 | 88 |
| 4 | 693727 | 693815 | 693903 | 693991 | 694078 | 694166 | 694254 | 694342 | 694430 | 694517 | |
| 5 | 694605 | 694693 | 694781 | 694868 | 694956 | 695044 | 695131 | 695219 | 695307 | 695394 | |
| 6 | 695482 | 695569 | 695657 | 695744 | 695832 | 695919 | 696007 | 696094 | 696182 | 696269 | |
| 7 | 696356 | 696444 | 696531 | 696618 | 696706 | 696793 | 696880 | 696968 | 697055 | 697142 | |
| 8 | 697229 | 697317 | 697404 | 697491 | 697578 | 697665 | 697752 | 697839 | 697926 | 698014 | 87 |
| 9 | 698100 | 698188 | 698275 | 698362 | 698449 | 698535 | 698622 | 698709 | 698796 | 698883 | |

Proportional Parts

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 98 | 9.8 | 19.6 | 29.4 | 39.2 | 49.0 | 58.8 | 68.6 | 78.4 | 88.2 |
| 96 | 9.6 | 19.2 | 28.8 | 38.4 | 48.0 | 57.6 | 67.2 | 76.8 | 86.4 |
| 94 | 9.4 | 18.8 | 28.2 | 37.6 | 47.0 | 56.4 | 65.8 | 75.2 | 84.6 |
| 92 | 9.2 | 18.4 | 27.6 | 36.8 | 46.0 | 55.2 | 64.4 | 73.6 | 82.8 |
| 90 | 9.0 | 18.0 | 27.0 | 36.0 | 45.0 | 54.0 | 63.0 | 72.0 | 81.0 |
| 88 | 8.8 | 17.6 | 26.4 | 35.2 | 44.0 | 52.8 | 61.6 | 70.4 | 79.2 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **500** | **698970** | **699057** | **699144** | **699231** | **699317** | **699404** | **699491** | **699578** | **699664** | **699751** | |
| 1 | 699838 | 699924 | 700011 | 700098 | 700184 | 700271 | 700358 | 700444 | 700531 | 700617 | |
| 2 | 700704 | 700790 | 700877 | 700963 | 701050 | 701136 | 701222 | 701309 | 701395 | 701482 | |
| 3 | 701568 | 701654 | 701741 | 701827 | 701913 | 701999 | 702086 | 702172 | 702258 | 702344 | |
| 4 | 702431 | 702517 | 702603 | 702689 | 702775 | 702861 | 702947 | 703033 | 703119 | 703205 | |
| 5 | 703291 | 703377 | 703463 | 703549 | 703635 | 703721 | 703807 | 703893 | 703979 | 704065 | 86 |
| 6 | 704151 | 704236 | 704322 | 704408 | 704494 | 704579 | 704665 | 704751 | 704837 | 704922 | |
| 7 | 705008 | 705094 | 705179 | 705265 | 705350 | 705436 | 705522 | 705607 | 705693 | 705778 | |
| 8 | 705864 | 705949 | 706035 | 706120 | 706206 | 706291 | 706376 | 706462 | 706547 | 706632 | |
| 9 | 706718 | 706803 | 706888 | 706974 | 707059 | 707144 | 707229 | 707315 | 707400 | 707485 | |
| **510** | **707570** | **707655** | **707740** | **707826** | **707911** | **707996** | **708081** | **708166** | **708251** | **708336** | |
| 1 | 708421 | 708506 | 708591 | 708676 | 708761 | 708846 | 708931 | 709015 | 709100 | 709185 | 85 |
| 2 | 709270 | 709355 | 709440 | 709524 | 709609 | 709694 | 709779 | 709863 | 709948 | 710033 | |
| 3 | 710117 | 710202 | 710287 | 710371 | 710456 | 710540 | 710625 | 710710 | 710794 | 710879 | |
| 4 | 710963 | 711048 | 711132 | 711217 | 711301 | 711385 | 711470 | 711554 | 711639 | 711723 | |
| 5 | 711807 | 711892 | 711976 | 712060 | 712144 | 712229 | 712313 | 712397 | 712481 | 712566 | |
| 6 | 712650 | 712734 | 712818 | 712902 | 712986 | 713070 | 713154 | 713238 | 713323 | 713407 | |
| 7 | 713491 | 713575 | 713659 | 713742 | 713826 | 713910 | 713994 | 714078 | 714162 | 714246 | 84 |
| 8 | 714330 | 714414 | 714497 | 714581 | 714665 | 714749 | 714833 | 714916 | 715000 | 715084 | |
| 9 | 715167 | 715251 | 715335 | 715418 | 715502 | 715586 | 715669 | 715753 | 715836 | 715920 | |
| **520** | **716003** | **716087** | **716170** | **716254** | **716337** | **716421** | **716504** | **716588** | **716671** | **716754** | |
| 1 | 716838 | 716921 | 717004 | 717088 | 717171 | 717254 | 717338 | 717421 | 717504 | 717587 | |
| 2 | 717671 | 717754 | 717837 | 717920 | 718003 | 718086 | 718169 | 718253 | 718336 | 718419 | 83 |
| 3 | 718502 | 718585 | 718668 | 718751 | 718834 | 718917 | 719000 | 719083 | 719165 | 719248 | |
| 4 | 719331 | 719414 | 719497 | 719580 | 719663 | 719745 | 719828 | 719911 | 719994 | 720077 | |
| 5 | 720159 | 720242 | 720325 | 720407 | 720490 | 720573 | 720655 | 720738 | 720821 | 720903 | |
| 6 | 720986 | 721068 | 721151 | 721233 | 721316 | 721398 | 721481 | 721563 | 721646 | 721728 | |
| 7 | 721811 | 721893 | 721975 | 722058 | 722140 | 722222 | 722305 | 722387 | 722469 | 722552 | |
| 8 | 722634 | 722716 | 722798 | 722881 | 722963 | 723045 | 723127 | 723209 | 723291 | 723374 | |
| 9 | 723456 | 723538 | 723620 | 723702 | 723784 | 723866 | 723948 | 724030 | 724112 | 724194 | 82 |
| **530** | **724276** | **724358** | **724440** | **724522** | **724604** | **724685** | **724767** | **724849** | **724931** | **725013** | |
| 1 | 725095 | 725176 | 725258 | 725340 | 725422 | 725503 | 725585 | 725667 | 725748 | 725830 | |
| 2 | 725912 | 725993 | 726075 | 726156 | 726238 | 726320 | 726401 | 726483 | 726564 | 726646 | |
| 3 | 726727 | 726809 | 726890 | 726972 | 727053 | 727134 | 727216 | 727297 | 727379 | 727460 | |
| 4 | 727541 | 727623 | 727704 | 727785 | 727866 | 727948 | 728029 | 728110 | 728191 | 728273 | |
| 5 | 728354 | 728435 | 728516 | 728597 | 728678 | 728759 | 728841 | 728922 | 729003 | 729084 | |
| 6 | 729165 | 729246 | 729327 | 729408 | 729489 | 729570 | 729651 | 729732 | 729813 | 729893 | 81 |
| 7 | 729974 | 730055 | 730136 | 730217 | 730298 | 730378 | 730459 | 730540 | 730621 | 730702 | |
| 8 | 730782 | 730863 | 730944 | 731024 | 731105 | 731186 | 731266 | 731347 | 731428 | 731508 | |
| 9 | 731589 | 731669 | 731750 | 731830 | 731911 | 731991 | 732072 | 732152 | 732233 | 732313 | |
| **540** | **732394** | **732474** | **732555** | **732635** | **732715** | **732796** | **732876** | **732956** | **733037** | **733117** | |
| 1 | 733197 | 733278 | 733358 | 733438 | 733518 | 733598 | 733679 | 733759 | 733839 | 733919 | |
| 2 | 733999 | 734079 | 734160 | 734240 | 734320 | 734400 | 734480 | 734560 | 734640 | 734720 | 80 |
| 3 | 734800 | 734880 | 734960 | 735040 | 735120 | 735200 | 735279 | 735359 | 735439 | 735519 | |
| 4 | 735599 | 735679 | 735759 | 735838 | 735918 | 735998 | 736078 | 736157 | 736237 | 736317 | |
| 5 | 736397 | 736476 | 736556 | 736635 | 736715 | 736795 | 736874 | 736954 | 737034 | 737113 | |
| 6 | 737193 | 737272 | 737352 | 737431 | 737511 | 737590 | 737670 | 737749 | 737829 | 737908 | |
| 7 | 737987 | 738067 | 738146 | 738225 | 738305 | 738384 | 738463 | 738543 | 738622 | 738701 | |
| 8 | 738781 | 738860 | 738939 | 739018 | 739097 | 739177 | 739256 | 739335 | 739414 | 739493 | |
| 9 | 739572 | 739651 | 739731 | 739810 | 739889 | 739968 | 740047 | 740126 | 740205 | 740284 | 79 |
| **550** | **740363** | **740442** | **740521** | **740600** | **740678** | **740757** | **740836** | **740915** | **740994** | **741073** | |
| 1 | 741152 | 741230 | 741309 | 741388 | 741467 | 741546 | 741624 | 741703 | 741782 | 741860 | |
| 2 | 741939 | 742018 | 742096 | 742175 | 742254 | 742332 | 742411 | 742489 | 742568 | 742647 | |
| 3 | 742725 | 742804 | 742882 | 742961 | 743039 | 743118 | 743196 | 743275 | 743353 | 743431 | |
| 4 | 743510 | 743588 | 743667 | 743745 | 743823 | 743902 | 743980 | 744058 | 744136 | 744215 | |
| 5 | 744293 | 744371 | 744449 | 744528 | 744606 | 744684 | 744762 | 744840 | 744919 | 744997 | |
| 6 | 745075 | 745153 | 745231 | 745309 | 745387 | 745465 | 745543 | 745621 | 745699 | 745777 | 78 |
| 7 | 745855 | 745933 | 746011 | 746089 | 746167 | 746245 | 746323 | 746401 | 746479 | 746556 | |
| 8 | 746634 | 746712 | 746790 | 746868 | 746945 | 747023 | 747101 | 747179 | 747256 | 747334 | |
| 9 | 747412 | 747489 | 747567 | 747645 | 747722 | 747800 | 747878 | 747955 | 748033 | 748110 | |

## PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 86 | 8.6 | 17.2 | 25.8 | 34.4 | 43.0 | 51.6 | 60.2 | 68.8 | 77.4 |
| 84 | 8.4 | 16.8 | 25.2 | 33.6 | 42.0 | 50.4 | 58.8 | 67.2 | 75.6 |
| 82 | 8.2 | 16.4 | 24.6 | 32.8 | 41.0 | 49.2 | 57.4 | 65.6 | 73.8 |
| 80 | 8.0 | 16.0 | 24.0 | 32.0 | 40.0 | 48.0 | 56.0 | 64.0 | 72.0 |
| 78 | 7.8 | 15.6 | 23.4 | 31.2 | 39.0 | 46.8 | 54.6 | 62.4 | 70.2 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **560** | **748188** | **748266** | **748343** | **748421** | **748498** | **748576** | **748653** | **748731** | **748808** | **748885** | |
| 1 | 748963 | 749040 | 749118 | 749195 | 749272 | 749350 | 749427 | 749504 | 749582 | 749659 | |
| 2 | 749736 | 749814 | 749891 | 749968 | 750045 | 750123 | 750200 | 750277 | 750354 | 750431 | |
| 3 | 750508 | 750586 | 750663 | 750740 | 750817 | 750894 | 750971 | 751048 | 751125 | 751202 | |
| 4 | 751279 | 751356 | 751433 | 751510 | 751587 | 751664 | 751741 | 751818 | 751895 | 751972 | 77 |
| 5 | 752048 | 752125 | 752202 | 752279 | 752356 | 752433 | 752509 | 752586 | 752663 | 752740 | |
| 6 | 752816 | 752893 | 752970 | 753047 | 753123 | 753200 | 753277 | 753353 | 753430 | 753506 | |
| 7 | 753583 | 753660 | 753736 | 753813 | 753889 | 753966 | 754042 | 754119 | 754195 | 754272 | |
| 8 | 754348 | 754425 | 754501 | 754578 | 754654 | 754730 | 754807 | 754883 | 754960 | 755036 | |
| 9 | 755112 | 755189 | 755265 | 755341 | 755417 | 755494 | 755570 | 755646 | 755722 | 755799 | |
| **570** | **755875** | **755951** | **756027** | **756103** | **756180** | **756256** | **756332** | **756408** | **756484** | **756560** | |
| 1 | 756636 | 756712 | 756788 | 756864 | 756940 | 757016 | 757092 | 757168 | 757244 | 757320 | 76 |
| 2 | 757396 | 757472 | 757548 | 757624 | 757700 | 757775 | 757851 | 757927 | 758003 | 758079 | |
| 3 | 758155 | 758230 | 758306 | 758382 | 758458 | 758533 | 758609 | 758685 | 758761 | 758836 | |
| 4 | 758912 | 758988 | 759063 | 759139 | 759214 | 759290 | 759366 | 759441 | 759517 | 759592 | |
| 5 | 759668 | 759743 | 759819 | 759894 | 759970 | 760045 | 760121 | 760196 | 760272 | 760347 | |
| 6 | 760422 | 760498 | 760573 | 760649 | 760724 | 760799 | 760875 | 760950 | 761025 | 761101 | |
| 7 | 761176 | 761251 | 761326 | 761402 | 761477 | 761552 | 761627 | 761702 | 761778 | 761853 | |
| 8 | 761928 | 762003 | 762078 | 762153 | 762228 | 762303 | 762378 | 762453 | 762529 | 762604 | 75 |
| 9 | 762679 | 762754 | 762829 | 762904 | 762978 | 763053 | 763128 | 763203 | 763278 | 763353 | |
| **580** | **763428** | **763503** | **763578** | **763653** | **763727** | **763802** | **763877** | **763952** | **764027** | **764101** | |
| 1 | 764176 | 764251 | 764326 | 764400 | 764475 | 764550 | 764624 | 764699 | 764774 | 764848 | |
| 2 | 764923 | 764998 | 765072 | 765147 | 765221 | 765296 | 765370 | 765445 | 765520 | 765594 | |
| 3 | 765669 | 765743 | 765818 | 765892 | 765966 | 766041 | 766115 | 766190 | 766264 | 766338 | |
| 4 | 766413 | 766487 | 766562 | 766636 | 766710 | 766785 | 766859 | 766933 | 767007 | 767082 | |
| 5 | 767156 | 767230 | 767304 | 767379 | 767453 | 767527 | 767601 | 767675 | 767749 | 767823 | |
| 6 | 767898 | 767972 | 768046 | 768120 | 768194 | 768268 | 768342 | 768416 | 768490 | 768564 | 74 |
| 7 | 768638 | 768712 | 768786 | 768860 | 768934 | 769008 | 769082 | 769156 | 769230 | 769303 | |
| 8 | 769377 | 769451 | 769525 | 769599 | 769673 | 769746 | 769820 | 769894 | 769968 | 770042 | |
| 9 | 770115 | 770189 | 770263 | 770336 | 770410 | 770484 | 770557 | 770631 | 770705 | 770778 | |
| **590** | **770852** | **770926** | **770999** | **771073** | **771146** | **771220** | **771293** | **771367** | **771440** | **771514** | |
| 1 | 771587 | 771661 | 771734 | 771808 | 771881 | 771955 | 772028 | 772102 | 772175 | 772248 | |
| 2 | 772322 | 772395 | 772468 | 772542 | 772615 | 772688 | 772762 | 772835 | 772908 | 772981 | |
| 3 | 773055 | 773128 | 773201 | 773274 | 773348 | 773421 | 773494 | 773567 | 773640 | 773713 | |
| 4 | 773786 | 773860 | 773933 | 774006 | 774079 | 774152 | 774225 | 774298 | 774371 | 774444 | 73 |
| 5 | 774517 | 774590 | 774663 | 774736 | 774809 | 774882 | 774955 | 775028 | 775100 | 775173 | |
| 6 | 775246 | 775319 | 775392 | 775465 | 775538 | 775610 | 775683 | 775756 | 775829 | 775902 | |
| 7 | 775974 | 776047 | 776120 | 776193 | 776265 | 776338 | 776411 | 776483 | 776556 | 776629 | |
| 8 | 776701 | 776774 | 776846 | 776919 | 776992 | 777064 | 777137 | 777209 | 777282 | 777354 | |
| 9 | 777427 | 777499 | 777572 | 777644 | 777717 | 777789 | 777862 | 777934 | 778006 | 778079 | |
| **600** | **778151** | **778224** | **778296** | **778368** | **778441** | **778513** | **778585** | **778658** | **778730** | **778802** | |
| 1 | 778874 | 778947 | 779019 | 779091 | 779163 | 779236 | 779308 | 779380 | 779452 | 779524 | |
| 2 | 779596 | 779669 | 779741 | 779813 | 779885 | 779957 | 780029 | 780101 | 780173 | 780245 | |
| 3 | 780317 | 780389 | 780461 | 780533 | 780605 | 780677 | 780749 | 780821 | 780893 | 780965 | 72 |
| 4 | 781037 | 781109 | 781181 | 781253 | 781324 | 781396 | 781468 | 781540 | 781612 | 781684 | |
| 5 | 781755 | 781827 | 781899 | 781971 | 782042 | 782114 | 782186 | 782258 | 782329 | 782401 | |
| 6 | 782473 | 782544 | 782616 | 782688 | 782759 | 782831 | 782902 | 782974 | 783046 | 783117 | |
| 7 | 783189 | 783260 | 783332 | 783403 | 783475 | 783546 | 783618 | 783689 | 783761 | 783832 | |
| 8 | 783904 | 783975 | 784046 | 784118 | 784189 | 784261 | 784332 | 784403 | 784475 | 784546 | |
| 9 | 784617 | 784689 | 784760 | 784831 | 784902 | 784974 | 785045 | 785116 | 785187 | 785259 | |
| **610** | **785330** | **785401** | **785472** | **785543** | **785615** | **785686** | **785757** | **785828** | **785899** | **785970** | |
| 1 | 786041 | 786112 | 786183 | 786254 | 786325 | 786396 | 786467 | 786538 | 786609 | 786680 | 71 |
| 2 | 786751 | 786822 | 786893 | 786964 | 787035 | 787106 | 787177 | 787248 | 787319 | 787390 | |
| 3 | 787460 | 787531 | 787602 | 787673 | 787744 | 787815 | 787885 | 787956 | 788027 | 788098 | |
| 4 | 788168 | 788239 | 788310 | 788381 | 788451 | 788522 | 788593 | 788663 | 788734 | 788804 | |
| 5 | 788875 | 788946 | 789016 | 789087 | 789157 | 789228 | 789299 | 789369 | 789440 | 789510 | |
| 6 | 789581 | 789651 | 789722 | 789792 | 789863 | 789933 | 790004 | 790074 | 790144 | 790215 | |
| 7 | 790285 | 790356 | 790426 | 790496 | 790567 | 790637 | 790707 | 790778 | 790848 | 790918 | |
| 8 | 790988 | 791059 | 791129 | 791199 | 791269 | 791340 | 791410 | 791480 | 791550 | 791620 | |
| 9 | 791691 | 791761 | 791831 | 791901 | 791971 | 792041 | 792111 | 792181 | 792252 | 792322 | |

PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 78 | 7.8 | 15.6 | 23.4 | 31.2 | 39.0 | 46.8 | 54.6 | 62.4 | 70.2 |
| 76 | 7.6 | 15.2 | 22.8 | 30.4 | 38.0 | 45.6 | 53.2 | 60.8 | 68.4 |
| 74 | 7.4 | 14.8 | 22.2 | 29.6 | 37.0 | 44.4 | 51.8 | 59.2 | 66.6 |
| 72 | 7.2 | 14.4 | 21.6 | 28.8 | 36.0 | 43.2 | 50.4 | 57.6 | 64.8 |
| 70 | 7.0 | 14.0 | 21.0 | 28.0 | 35.0 | 42.0 | 49.0 | 56.0 | 63.0 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **620** | **792392** | **792462** | **792532** | **792602** | **792672** | **792742** | **792812** | **792882** | **792952** | **793022** | 70 |
| 1 | 793092 | 793162 | 793231 | 793301 | 793371 | 793441 | 793511 | 793581 | 793651 | 793721 | |
| 2 | 793790 | 793860 | 793930 | 794000 | 794070 | 794139 | 794209 | 794279 | 794349 | 794418 | |
| 3 | 794488 | 794558 | 794627 | 794697 | 794767 | 794836 | 794906 | 794976 | 795045 | 795115 | |
| 4 | 795185 | 795254 | 795324 | 795393 | 795463 | 795532 | 795602 | 795672 | 795741 | 795811 | |
| 5 | 795880 | 795949 | 796019 | 796088 | 796158 | 796227 | 796297 | 796366 | 796436 | 796505 | |
| 6 | 796574 | 796644 | 796713 | 796782 | 796852 | 796921 | 796990 | 797060 | 797129 | 797198 | |
| 7 | 797268 | 797337 | 797406 | 797475 | 797545 | 797614 | 797683 | 797752 | 797821 | 797890 | |
| 8 | 797960 | 798029 | 798098 | 798167 | 798236 | 798305 | 798374 | 798443 | 798513 | 798582 | |
| 9 | 798651 | 798720 | 798789 | 798858 | 798927 | 798996 | 799065 | 799134 | 799203 | 799272 | 69 |
| **630** | **799341** | **799409** | **799478** | **799547** | **799616** | **799685** | **799754** | **799823** | **799892** | **799961** | |
| 1 | 800029 | 800098 | 800167 | 800236 | 800305 | 800373 | 800442 | 800511 | 800580 | 800648 | |
| 2 | 800717 | 800786 | 800854 | 800923 | 800992 | 801061 | 801129 | 801198 | 801266 | 801335 | |
| 3 | 801404 | 801472 | 801541 | 801609 | 801678 | 801747 | 801815 | 801884 | 801952 | 802021 | |
| 4 | 802089 | 802158 | 802226 | 802295 | 802363 | 802432 | 802500 | 802568 | 802637 | 802705 | |
| 5 | 802774 | 802842 | 802910 | 802979 | 803047 | 803116 | 803184 | 803252 | 803321 | 803389 | |
| 6 | 803457 | 803525 | 803594 | 803662 | 803730 | 803798 | 803867 | 803935 | 804003 | 804071 | |
| 7 | 804139 | 804208 | 804276 | 804344 | 804412 | 804480 | 804548 | 804616 | 804685 | 804753 | |
| 8 | 804821 | 804889 | 804957 | 805025 | 805093 | 805161 | 805229 | 805297 | 805365 | 805433 | 68 |
| 9 | 805501 | 805569 | 805637 | 805705 | 805773 | 805841 | 805908 | 805976 | 806044 | 806112 | |
| **640** | **806180** | **806248** | **806316** | **806384** | **806451** | **806519** | **806587** | **806655** | **806723** | **806790** | |
| 1 | 806858 | 806926 | 806994 | 807061 | 807129 | 807197 | 807264 | 807332 | 807400 | 807467 | |
| 2 | 807535 | 807603 | 807670 | 807738 | 807806 | 807873 | 807941 | 808008 | 808076 | 808143 | |
| 3 | 808211 | 808279 | 808346 | 808414 | 808481 | 808549 | 808616 | 808684 | 808751 | 808818 | |
| 4 | 808886 | 808953 | 809021 | 809088 | 809156 | 809223 | 809290 | 809358 | 809425 | 809492 | |
| 5 | 809560 | 809627 | 809694 | 809762 | 809829 | 809896 | 809964 | 810031 | 810098 | 810165 | |
| 6 | 810233 | 810300 | 810367 | 810434 | 810501 | 810569 | 810636 | 810703 | 810770 | 810837 | |
| 7 | 810904 | 810971 | 811039 | 811106 | 811173 | 811240 | 811307 | 811374 | 811441 | 811508 | 67 |
| 8 | 811575 | 811642 | 811709 | 811776 | 811843 | 811910 | 811977 | 812044 | 812111 | 812178 | |
| 9 | 812245 | 812312 | 812379 | 812445 | 812512 | 812579 | 812646 | 812713 | 812780 | 812847 | |
| **650** | **812913** | **812980** | **813047** | **813114** | **813181** | **813247** | **813314** | **813381** | **813448** | **813514** | |
| 1 | 813581 | 813648 | 813714 | 813781 | 813848 | 813914 | 813981 | 814048 | 814114 | 814181 | |
| 2 | 814248 | 814314 | 814381 | 814447 | 814514 | 814581 | 814647 | 814714 | 814780 | 814847 | |
| 3 | 814913 | 814980 | 815046 | 815113 | 815179 | 815246 | 815312 | 815378 | 815445 | 815511 | |
| 4 | 815578 | 815644 | 815711 | 815777 | 815843 | 815910 | 815976 | 816042 | 816109 | 816175 | |
| 5 | 816241 | 816308 | 816374 | 816440 | 816506 | 816573 | 816639 | 816705 | 816771 | 816838 | |
| 6 | 816904 | 816970 | 817036 | 817102 | 817169 | 817235 | 817301 | 817367 | 817433 | 817499 | |
| 7 | 817565 | 817631 | 817698 | 817764 | 817830 | 817896 | 817962 | 818028 | 818094 | 818160 | |
| 8 | 818226 | 818292 | 818358 | 818424 | 818490 | 818556 | 818622 | 818688 | 818754 | 818820 | 66 |
| 9 | 818885 | 818951 | 819017 | 819083 | 819149 | 819215 | 819281 | 819346 | 819412 | 819478 | |
| **660** | **819544** | **819610** | **819676** | **819741** | **819807** | **819873** | **819939** | **820004** | **820070** | **820136** | |
| 1 | 820201 | 820267 | 820333 | 820399 | 820464 | 820530 | 820595 | 820661 | 820727 | 820792 | |
| 2 | 820858 | 820924 | 820989 | 821055 | 821120 | 821186 | 821251 | 821317 | 821382 | 821448 | |
| 3 | 821514 | 821579 | 821645 | 821710 | 821775 | 821841 | 821906 | 821972 | 822037 | 822103 | |
| 4 | 822168 | 822233 | 822299 | 822364 | 822430 | 822495 | 822560 | 822626 | 822691 | 822756 | |
| 5 | 822822 | 822887 | 822952 | 823018 | 823083 | 823148 | 823213 | 823279 | 823344 | 823409 | |
| 6 | 823474 | 823539 | 823605 | 823670 | 823735 | 823800 | 823865 | 823930 | 823996 | 824061 | |
| 7 | 824126 | 824191 | 824256 | 824321 | 824386 | 824451 | 824516 | 824581 | 824646 | 824711 | |
| 8 | 824776 | 824841 | 824906 | 824971 | 825036 | 825101 | 825166 | 825231 | 825296 | 825361 | 65 |
| 9 | 825426 | 825491 | 825556 | 825621 | 825686 | 825751 | 825815 | 825880 | 825945 | 826010 | |
| **670** | **826075** | **826140** | **826204** | **826269** | **826334** | **826399** | **826464** | **826528** | **826593** | **826658** | |
| 1 | 826723 | 826787 | 826852 | 826917 | 826981 | 827046 | 827111 | 827175 | 827240 | 827305 | |
| 2 | 827369 | 827434 | 827499 | 827563 | 827628 | 827692 | 827757 | 827821 | 827886 | 827951 | |
| 3 | 828015 | 828080 | 828144 | 828209 | 828273 | 828338 | 828402 | 828467 | 828531 | 828595 | |
| 4 | 828660 | 828724 | 828789 | 828853 | 828918 | 828982 | 829046 | 829111 | 829175 | 829239 | |
| 5 | 829304 | 829368 | 829432 | 829497 | 829561 | 829625 | 829690 | 829754 | 829818 | 829882 | |
| 6 | 829947 | 830011 | 830075 | 830139 | 830204 | 830268 | 830332 | 830396 | 830460 | 830525 | |
| 7 | 830589 | 830653 | 830717 | 830781 | 830845 | 830909 | 830973 | 831037 | 831102 | 831166 | |
| 8 | 831230 | 831294 | 831358 | 831422 | 831486 | 831550 | 831614 | 831678 | 831742 | 831806 | 64 |
| 9 | 831870 | 831934 | 831998 | 832062 | 832126 | 832189 | 832253 | 832317 | 832381 | 832445 | |

PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 70 | 7.0 | 14.0 | 21.0 | 28.0 | 35.0 | 42.0 | 49.0 | 56.0 | 63.0 |
| 68 | 6.8 | 13.6 | 20.4 | 27.2 | 34.0 | 40.8 | 47.6 | 54.4 | 61.2 |
| 66 | 6.6 | 13.2 | 19.8 | 26.4 | 33.0 | 39.6 | 46.2 | 52.8 | 59.4 |
| 64 | 6.4 | 12.8 | 19.2 | 25.6 | 32.0 | 38.4 | 44.8 | 51.2 | 57.6 |
| 62 | 6.2 | 12.4 | 18.6 | 24.8 | 31.0 | 37.2 | 43.4 | 49.6 | 55.8 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **680** | **832509** | **832573** | **832637** | **832700** | **832764** | **832828** | **832892** | **832956** | **833020** | **833083** | |
| 1 | 833147 | 833211 | 833275 | 833338 | 833402 | 833466 | 833530 | 833593 | 833657 | 833721 | |
| 2 | 833784 | 833848 | 833912 | 833975 | 834039 | 834103 | 834166 | 834230 | 834294 | 834357 | |
| 3 | 834421 | 834484 | 834548 | 834611 | 834675 | 834739 | 834802 | 834866 | 834929 | 834993 | |
| 4 | 835056 | 835120 | 835183 | 835247 | 835310 | 835373 | 835437 | 835500 | 835564 | 835627 | |
| 5 | 835691 | 835754 | 835817 | 835881 | 835944 | 836007 | 836071 | 836134 | 836197 | 836261 | |
| 6 | 836324 | 836387 | 836451 | 836514 | 836577 | 836641 | 836704 | 836767 | 836830 | 836894 | |
| 7 | 836957 | 837020 | 837083 | 837146 | 837210 | 837273 | 837336 | 837399 | 837462 | 837525 | |
| 8 | 837588 | 837652 | 837715 | 837778 | 837841 | 837904 | 837967 | 838030 | 838093 | 838156 | 63 |
| 9 | 838219 | 838282 | 838345 | 838408 | 838471 | 838534 | 838597 | 838660 | 838723 | 838786 | |
| **690** | **838849** | **838912** | **838975** | **839038** | **839101** | **839164** | **839227** | **839289** | **839352** | **839415** | |
| 1 | 839478 | 839541 | 839604 | 839667 | 839729 | 839792 | 839855 | 839918 | 839981 | 840043 | |
| 2 | 840106 | 840169 | 840232 | 840294 | 840357 | 840420 | 840482 | 840545 | 840608 | 840671 | |
| 3 | 840733 | 840796 | 840859 | 840921 | 840984 | 841046 | 841109 | 841172 | 841234 | 841297 | |
| 4 | 841359 | 841422 | 841485 | 841547 | 841610 | 841672 | 841735 | 841797 | 841860 | 841922 | |
| 5 | 841985 | 842047 | 842110 | 842172 | 842235 | 842297 | 842360 | 842422 | 842484 | 842547 | |
| 6 | 842609 | 842672 | 842734 | 842796 | 842859 | 842921 | 842983 | 843046 | 843108 | 843170 | |
| 7 | 843233 | 843295 | 843357 | 843420 | 843482 | 843544 | 843606 | 843669 | 843731 | 843793 | |
| 8 | 843855 | 843918 | 843980 | 844042 | 844104 | 844166 | 844229 | 844291 | 844353 | 844415 | |
| 9 | 844477 | 844539 | 844601 | 844664 | 844726 | 844788 | 844850 | 844912 | 844974 | 845036 | |
| **700** | **845098** | **845160** | **845222** | **845284** | **845346** | **845408** | **845470** | **845532** | **845594** | **845656** | 62 |
| 1 | 845718 | 845780 | 845842 | 845904 | 845966 | 846028 | 846090 | 846151 | 846213 | 846275 | |
| 2 | 846337 | 846399 | 846461 | 846523 | 846585 | 846646 | 846708 | 846770 | 846832 | 846894 | |
| 3 | 846955 | 847017 | 847079 | 847141 | 847202 | 847264 | 847326 | 847388 | 847449 | 847511 | |
| 4 | 847573 | 847634 | 847696 | 847758 | 847819 | 847881 | 847943 | 848004 | 848066 | 848128 | |
| 5 | 848189 | 848251 | 848312 | 848374 | 848435 | 848497 | 848559 | 848620 | 848682 | 848743 | |
| 6 | 848805 | 848866 | 848928 | 848989 | 849051 | 849112 | 849174 | 849235 | 849297 | 849358 | |
| 7 | 849419 | 849481 | 849542 | 849604 | 849665 | 849726 | 849788 | 849849 | 849911 | 849972 | |
| 8 | 850033 | 850095 | 850156 | 850217 | 850279 | 850340 | 850401 | 850462 | 850524 | 850585 | |
| 9 | 850646 | 850707 | 850769 | 850830 | 850891 | 850952 | 851014 | 851075 | 851136 | 851197 | |
| **710** | **851258** | **851320** | **851381** | **851442** | **851503** | **851564** | **851625** | **851686** | **851747** | **851809** | |
| 1 | 851870 | 851931 | 851992 | 852053 | 852114 | 852175 | 852236 | 852297 | 852358 | 852419 | 61 |
| 2 | 852480 | 852541 | 852602 | 852663 | 852724 | 852785 | 852846 | 852907 | 852968 | 853029 | |
| 3 | 853090 | 853150 | 853211 | 853272 | 853333 | 853394 | 853455 | 853516 | 853577 | 853637 | |
| 4 | 853698 | 853759 | 853820 | 853881 | 853941 | 854002 | 854063 | 854124 | 854185 | 854245 | |
| 5 | 854306 | 854367 | 854428 | 854488 | 854549 | 854610 | 854670 | 854731 | 854792 | 854852 | |
| 6 | 854913 | 854974 | 855034 | 855095 | 855156 | 855216 | 855277 | 855337 | 855398 | 855459 | |
| 7 | 855519 | 855580 | 855640 | 855701 | 855761 | 855822 | 855882 | 855943 | 856003 | 856064 | |
| 8 | 856124 | 856185 | 856245 | 856306 | 856366 | 856427 | 856487 | 856548 | 856608 | 856668 | |
| 9 | 856729 | 856789 | 856850 | 856910 | 856970 | 857031 | 857091 | 857152 | 857212 | 857272 | |
| **720** | **857332** | **857393** | **857453** | **857513** | **857574** | **857634** | **857694** | **857755** | **857815** | **857875** | |
| 1 | 857935 | 857995 | 858056 | 858116 | 858176 | 858236 | 858297 | 858357 | 858417 | 858477 | |
| 2 | 858537 | 858597 | 858657 | 858718 | 858778 | 858838 | 858898 | 858958 | 859018 | 859078 | |
| 3 | 859138 | 859198 | 859258 | 859318 | 859379 | 859439 | 859499 | 859559 | 859619 | 859679 | 60 |
| 4 | 859739 | 859799 | 859859 | 859918 | 859978 | 860038 | 860098 | 860158 | 860218 | 860278 | |
| 5 | 860338 | 860398 | 860458 | 860518 | 860578 | 860637 | 860697 | 860757 | 860817 | 860877 | |
| 6 | 860937 | 860996 | 861056 | 861116 | 861176 | 861236 | 861295 | 861355 | 861415 | 861475 | |
| 7 | 861534 | 861594 | 861654 | 861714 | 861773 | 861833 | 861893 | 861952 | 862012 | 862072 | |
| 8 | 862131 | 862191 | 862251 | 862310 | 862370 | 862430 | 862489 | 862549 | 862608 | 862668 | |
| 9 | 862728 | 862787 | 862847 | 862906 | 862966 | 863025 | 863085 | 863144 | 863204 | 863263 | |
| **730** | **863323** | **863382** | **863442** | **863501** | **863561** | **863620** | **863680** | **863739** | **863799** | **863858** | |
| 1 | 863917 | 863977 | 864036 | 864096 | 864155 | 864214 | 864274 | 864333 | 864392 | 864452 | |
| 2 | 864511 | 864570 | 864630 | 864689 | 864748 | 864808 | 864867 | 864926 | 864985 | 865045 | |
| 3 | 865104 | 865163 | 865222 | 865282 | 865341 | 865400 | 865459 | 865519 | 865578 | 865637 | |
| 4 | 865696 | 865755 | 865814 | 865874 | 865933 | 865992 | 866051 | 866110 | 866169 | 866228 | |
| 5 | 866287 | 866346 | 866405 | 866465 | 866524 | 866583 | 866642 | 866701 | 866760 | 866819 | |
| 6 | 866878 | 866937 | 866996 | 867055 | 867114 | 867173 | 867232 | 867291 | 867350 | 867409 | 59 |
| 7 | 867467 | 867526 | 867585 | 867644 | 867703 | 867762 | 867821 | 867880 | 867939 | 867998 | |
| 8 | 868056 | 868115 | 868174 | 868233 | 868292 | 868350 | 868409 | 868468 | 868527 | 868586 | |
| 9 | 868644 | 868703 | 868762 | 868821 | 868879 | 868938 | 868997 | 869056 | 869114 | 869173 | |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 64 | 6.4 | 12.8 | 19.2 | 25.6 | 32.0 | 38.4 | 44.8 | 51.2 | 57.6 |
| 62 | 6.2 | 12.4 | 18.6 | 24.8 | 31.0 | 37.2 | 43.4 | 49.6 | 55.8 |
| 60 | 6.0 | 12.0 | 18.0 | 24.0 | 30.0 | 36.0 | 42.0 | 48.0 | 54.0 |
| 58 | 5.8 | 11.6 | 17.4 | 23.2 | 29.0 | 34.8 | 40.6 | 46.4 | 52.2 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **740** | **869232** | **869290** | **869349** | **869408** | **869466** | **869525** | **869584** | **869642** | **869701** | **869760** | |
| 1 | 869818 | 869877 | 869935 | 869994 | 870053 | 870111 | 870170 | 870228 | 870287 | 870345 | |
| 2 | 870404 | 870462 | 870521 | 870579 | 870638 | 870696 | 870755 | 870813 | 870872 | 870930 | |
| 3 | 870989 | 871047 | 871106 | 871164 | 871223 | 871281 | 871339 | 871398 | 871456 | 871515 | |
| 4 | 871573 | 871631 | 871690 | 871748 | 871806 | 871865 | 871923 | 871981 | 872040 | 872098 | |
| 5 | 872156 | 872215 | 872273 | 872331 | 872389 | 872448 | 872506 | 872564 | 872622 | 872681 | |
| 6 | 872739 | 872797 | 872855 | 872913 | 872972 | 873030 | 873088 | 873146 | 873204 | 873262 | |
| 7 | 873321 | 873379 | 873437 | 873495 | 873553 | 873611 | 873669 | 873727 | 873785 | 873844 | |
| 8 | 873902 | 873960 | 874018 | 874076 | 874134 | 874192 | 874250 | 874308 | 874366 | 874424 | 58 |
| 9 | 874482 | 874540 | 874598 | 874656 | 874714 | 874772 | 874830 | 874888 | 874945 | 875003 | |
| **750** | **875061** | **875119** | **875177** | **875235** | **875293** | **875351** | **875409** | **875466** | **875524** | **875582** | |
| 1 | 875640 | 875698 | 875756 | 875813 | 875871 | 875929 | 875987 | 876045 | 876102 | 876160 | |
| 2 | 876218 | 876276 | 876333 | 876391 | 876449 | 876507 | 876564 | 876622 | 876680 | 876737 | |
| 3 | 876795 | 876853 | 876910 | 876968 | 877026 | 877083 | 877141 | 877199 | 877256 | 877314 | |
| 4 | 877371 | 877429 | 877487 | 877544 | 877602 | 877659 | 877717 | 877774 | 877832 | 877889 | |
| 5 | 877947 | 878004 | 878062 | 878119 | 878177 | 878234 | 878292 | 878349 | 878407 | 878464 | |
| 6 | 878522 | 878579 | 878637 | 878694 | 878752 | 878809 | 878866 | 878924 | 878981 | 879039 | |
| 7 | 879096 | 879153 | 879211 | 879268 | 879325 | 879383 | 879440 | 879497 | 879555 | 879612 | |
| 8 | 879669 | 879726 | 879784 | 879841 | 879898 | 879956 | 880013 | 880070 | 880127 | 880185 | |
| 9 | 880242 | 880299 | 880356 | 880413 | 880471 | 880528 | 880585 | 880642 | 880699 | 880756 | |
| **760** | **880814** | **880871** | **880928** | **880985** | **881042** | **881099** | **881156** | **881213** | **881271** | **881328** | |
| 1 | 881385 | 881442 | 881499 | 881556 | 881613 | 881670 | 881727 | 881784 | 881841 | 881898 | |
| 2 | 881955 | 882012 | 882069 | 882126 | 882183 | 882240 | 882297 | 882354 | 882411 | 882468 | 57 |
| 3 | 882525 | 882581 | 882638 | 882695 | 882752 | 882809 | 882866 | 882923 | 882980 | 883037 | |
| 4 | 883093 | 883150 | 883207 | 883264 | 883321 | 883377 | 883434 | 883491 | 883548 | 883605 | |
| 5 | 883661 | 883718 | 883775 | 883832 | 883888 | 883945 | 884002 | 884059 | 884115 | 884172 | |
| 6 | 884229 | 884285 | 884342 | 884399 | 884455 | 884512 | 884569 | 884625 | 884682 | 884739 | |
| 7 | 884795 | 884852 | 884909 | 884965 | 885022 | 885078 | 885135 | 885192 | 885248 | 885305 | |
| 8 | 885361 | 885418 | 885474 | 885531 | 885587 | 885644 | 885700 | 885757 | 885813 | 885870 | |
| 9 | 885926 | 885983 | 886039 | 886096 | 886152 | 886209 | 886265 | 886321 | 886378 | 886434 | |
| **770** | **886491** | **886547** | **886604** | **886660** | **886716** | **886773** | **886829** | **886885** | **886942** | **886998** | |
| 1 | 887054 | 887111 | 887167 | 887223 | 887280 | 887336 | 887392 | 887449 | 887505 | 887561 | |
| 2 | 887617 | 887674 | 887730 | 887786 | 887842 | 887898 | 887955 | 888011 | 888067 | 888123 | |
| 3 | 888179 | 888236 | 888292 | 888348 | 888404 | 888460 | 888516 | 888573 | 888629 | 888685 | |
| 4 | 888741 | 888797 | 888853 | 888909 | 888965 | 889021 | 889077 | 889134 | 889190 | 889246 | |
| 5 | 889302 | 889358 | 889414 | 889470 | 889526 | 889582 | 889638 | 889694 | 889750 | 889806 | 56 |
| 6 | 889862 | 889918 | 889974 | 890030 | 890086 | 890141 | 890197 | 890253 | 890309 | 890365 | |
| 7 | 890421 | 890477 | 890533 | 890589 | 890645 | 890700 | 890756 | 890812 | 890868 | 890924 | |
| 8 | 890980 | 891035 | 891091 | 891147 | 891203 | 891259 | 891314 | 891370 | 891426 | 891482 | |
| 9 | 891537 | 891593 | 891649 | 891705 | 891760 | 891816 | 891872 | 891928 | 891983 | 892039 | |
| **780** | **892095** | **892150** | **892206** | **892262** | **892317** | **892373** | **892429** | **892484** | **892540** | **892595** | |
| 1 | 892651 | 892707 | 892762 | 892818 | 892873 | 892929 | 892985 | 893040 | 893096 | 893151 | |
| 2 | 893207 | 893262 | 893318 | 893373 | 893429 | 893484 | 893540 | 893595 | 893651 | 893706 | |
| 3 | 893762 | 893817 | 893873 | 893928 | 893984 | 894039 | 894094 | 894150 | 894205 | 894261 | |
| 4 | 894316 | 894371 | 894427 | 894482 | 894538 | 894593 | 894648 | 894704 | 894759 | 894814 | |
| 5 | 894870 | 894925 | 894980 | 895036 | 895091 | 895146 | 895201 | 895257 | 895312 | 895367 | |
| 6 | 895423 | 895478 | 895533 | 895588 | 895644 | 895699 | 895754 | 895809 | 895864 | 895920 | |
| 7 | 895975 | 896030 | 896085 | 896140 | 896195 | 896251 | 896306 | 896361 | 896416 | 896471 | |
| 8 | 896526 | 896581 | 896636 | 896692 | 896747 | 896802 | 896857 | 896912 | 896967 | 897022 | |
| 9 | 897077 | 897132 | 897187 | 897242 | 897297 | 897352 | 897407 | 897462 | 897517 | 897572 | |
| **790** | **897627** | **897682** | **897737** | **897792** | **897847** | **897902** | **897957** | **898012** | **898067** | **898122** | 55 |
| 1 | 898176 | 898231 | 898286 | 898341 | 898396 | 898451 | 898506 | 898561 | 898615 | 898670 | |
| 2 | 898725 | 898780 | 898835 | 898890 | 898944 | 898999 | 899054 | 899109 | 899164 | 899218 | |
| 3 | 899273 | 899328 | 899383 | 899437 | 899492 | 899547 | 899602 | 899656 | 899711 | 899766 | |
| 4 | 899821 | 899875 | 899930 | 899985 | 900039 | 900094 | 900149 | 900203 | 900258 | 900312 | |
| 5 | 900367 | 900422 | 900476 | 900531 | 900586 | 900640 | 900695 | 900749 | 900804 | 900859 | |
| 6 | 900913 | 900968 | 901022 | 901077 | 901131 | 901186 | 901240 | 901295 | 901349 | 901404 | |
| 7 | 901458 | 901513 | 901567 | 901622 | 901676 | 901731 | 901785 | 901840 | 901894 | 901948 | |
| 8 | 902003 | 902057 | 902112 | 902166 | 902221 | 902275 | 902329 | 902384 | 902438 | 902492 | |
| 9 | 902547 | 902601 | 902655 | 902710 | 902764 | 902818 | 902873 | 902927 | 902981 | 903036 | |

### Proportional Parts

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 60 | 6.0 | 12.0 | 18.0 | 24.0 | 30.0 | 36.0 | 42.0 | 48.0 | 54.0 |
| 58 | 5.8 | 11.6 | 17.4 | 23.2 | 29.0 | 34.8 | 40.6 | 46.4 | 52.2 |
| 56 | 5.6 | 11.2 | 16.8 | 22.4 | 28.0 | 33.6 | 39.2 | 44.8 | 50.4 |
| 54 | 5.4 | 10.8 | 16.2 | 21.6 | 27.0 | 32.4 | 37.8 | 43.2 | 48.6 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **800** | **903090** | **903144** | **903199** | **903253** | **903307** | **903361** | **903416** | **903470** | **903524** | **903578** | |
| 1 | 903633 | 903687 | 903741 | 903795 | 903849 | 903904 | 903958 | 904012 | 904066 | 904120 | |
| 2 | 904174 | 904229 | 904283 | 904337 | 904391 | 904445 | 904499 | 904553 | 904607 | 904661 | |
| 3 | 904716 | 904770 | 904824 | 904878 | 904932 | 904986 | 905040 | 905094 | 905148 | 905202 | |
| 4 | 905256 | 905310 | 905364 | 905418 | 905472 | 905526 | 905580 | 905634 | 905688 | 905742 | 54 |
| 5 | 905796 | 905850 | 905904 | 905958 | 906012 | 906066 | 906119 | 906173 | 906227 | 906281 | |
| 6 | 906335 | 906389 | 906443 | 906497 | 906551 | 906604 | 906658 | 906712 | 906766 | 906820 | |
| 7 | 906874 | 906927 | 906981 | 907035 | 907089 | 907143 | 907196 | 907250 | 907304 | 907358 | |
| 8 | 907411 | 907465 | 907519 | 907573 | 907626 | 907680 | 907734 | 907787 | 907841 | 907895 | |
| 9 | 907949 | 908002 | 908056 | 908110 | 908163 | 908217 | 908270 | 908324 | 908378 | 908431 | |
| **810** | **908485** | **908539** | **908592** | **908646** | **908699** | **908753** | **908807** | **908860** | **908914** | **908967** | |
| 1 | 909021 | 909074 | 909128 | 909181 | 909235 | 909289 | 909342 | 909396 | 909449 | 909503 | |
| 2 | 909556 | 909610 | 909663 | 909716 | 909770 | 909823 | 909877 | 909930 | 909984 | 910037 | |
| 3 | 910091 | 910144 | 910197 | 910251 | 910304 | 910358 | 910411 | 910464 | 910518 | 910571 | |
| 4 | 910624 | 910678 | 910731 | 910784 | 910838 | 910891 | 910944 | 910998 | 911051 | 911104 | |
| 5 | 911158 | 911211 | 911264 | 911317 | 911371 | 911424 | 911477 | 911530 | 911584 | 911637 | |
| 6 | 911690 | 911743 | 911797 | 911850 | 911903 | 911956 | 912009 | 912063 | 912116 | 912169 | |
| 7 | 912222 | 912275 | 912328 | 912381 | 912435 | 912488 | 912541 | 912594 | 912647 | 912700 | |
| 8 | 912753 | 912806 | 912859 | 912913 | 912966 | 913019 | 913072 | 913125 | 913178 | 913231 | |
| 9 | 913284 | 913337 | 913390 | 913443 | 913496 | 913549 | 913602 | 913655 | 913708 | 913761 | 53 |
| **820** | **913814** | **913867** | **913920** | **913973** | **914026** | **914079** | **914132** | **914184** | **914237** | **914290** | |
| 1 | 914343 | 914396 | 914449 | 914502 | 914555 | 914608 | 914660 | 914713 | 914766 | 914819 | |
| 2 | 914872 | 914925 | 914977 | 915030 | 915083 | 915136 | 915189 | 915241 | 915294 | 915347 | |
| 3 | 915400 | 915453 | 915505 | 915558 | 915611 | 915664 | 915716 | 915769 | 915822 | 915875 | |
| 4 | 915927 | 915980 | 916033 | 916085 | 916138 | 916191 | 916243 | 916296 | 916349 | 916401 | |
| 5 | 916454 | 916507 | 916559 | 916612 | 916664 | 916717 | 916770 | 916822 | 916875 | 916927 | |
| 6 | 916980 | 917033 | 917085 | 917138 | 917190 | 917243 | 917295 | 917348 | 917400 | 917453 | |
| 7 | 917506 | 917558 | 917611 | 917663 | 917716 | 917768 | 917820 | 917873 | 917925 | 917978 | |
| 8 | 918030 | 918083 | 918135 | 918188 | 918240 | 918293 | 918345 | 918397 | 918450 | 918502 | |
| 9 | 918555 | 918607 | 918659 | 918712 | 918764 | 918816 | 918869 | 918921 | 918973 | 919026 | |
| **830** | **919078** | **919130** | **919183** | **919235** | **919287** | **919340** | **919392** | **919444** | **919496** | **919549** | |
| 1 | 919601 | 919653 | 919706 | 919758 | 919810 | 919862 | 919914 | 919967 | 920019 | 920071 | |
| 2 | 920123 | 920176 | 920228 | 920280 | 920332 | 920384 | 920436 | 920489 | 920541 | 920593 | |
| 3 | 920645 | 920697 | 920749 | 920801 | 920853 | 920906 | 920958 | 921010 | 921062 | 921114 | 52 |
| 4 | 921166 | 921218 | 921270 | 921322 | 921374 | 921426 | 921478 | 921530 | 921582 | 921634 | |
| 5 | 921686 | 921738 | 921790 | 921842 | 921894 | 921946 | 921998 | 922050 | 922102 | 922154 | |
| 6 | 922206 | 922258 | 922310 | 922362 | 922414 | 922466 | 922518 | 922570 | 922622 | 922674 | |
| 7 | 922725 | 922777 | 922829 | 922881 | 922933 | 922985 | 923037 | 923089 | 923140 | 923192 | |
| 8 | 923244 | 923296 | 923348 | 923399 | 923451 | 923503 | 923555 | 923607 | 923658 | 923710 | |
| 9 | 923762 | 923814 | 923865 | 923917 | 923969 | 924021 | 924072 | 924124 | 924176 | 924228 | |
| **840** | **924279** | **924331** | **924383** | **924434** | **924486** | **924538** | **924589** | **924641** | **924693** | **924744** | |
| 1 | 924796 | 924848 | 924899 | 924951 | 925003 | 925054 | 925106 | 925157 | 925209 | 925261 | |
| 2 | 925312 | 925364 | 925415 | 925467 | 925518 | 925570 | 925621 | 925673 | 925725 | 925776 | |
| 3 | 925828 | 925879 | 925931 | 925982 | 926034 | 926085 | 926137 | 926188 | 926240 | 926291 | |
| 4 | 926342 | 926394 | 926445 | 926497 | 926548 | 926600 | 926651 | 926702 | 926754 | 926805 | |
| 5 | 926857 | 926908 | 926959 | 927011 | 927062 | 927114 | 927165 | 927216 | 927268 | 927319 | |
| 6 | 927370 | 927422 | 927473 | 927524 | 927576 | 927627 | 927678 | 927730 | 927781 | 927832 | |
| 7 | 927883 | 927935 | 927986 | 928037 | 928088 | 928140 | 928191 | 928242 | 928293 | 928345 | |
| 8 | 928396 | 928447 | 928498 | 928549 | 928601 | 928652 | 928703 | 928754 | 928805 | 928857 | |
| 9 | 928908 | 928959 | 929010 | 929061 | 929112 | 929163 | 929215 | 929266 | 929317 | 929368 | |
| **850** | **929419** | **929470** | **929521** | **929572** | **929623** | **929674** | **929725** | **929776** | **929827** | **929879** | |
| 1 | 929930 | 929981 | 930032 | 930083 | 930134 | 930185 | 930236 | 930287 | 930338 | 930389 | 51 |
| 2 | 930440 | 930491 | 930542 | 930592 | 930643 | 930694 | 930745 | 930796 | 930847 | 930898 | |
| 3 | 930949 | 931000 | 931051 | 931102 | 931153 | 931204 | 931254 | 931305 | 931356 | 931407 | |
| 4 | 931458 | 931509 | 931560 | 931610 | 931661 | 931712 | 931763 | 931814 | 931865 | 931915 | |
| 5 | 931966 | 932017 | 932068 | 932118 | 932169 | 932220 | 932271 | 932322 | 932372 | 932423 | |
| 6 | 932474 | 932524 | 932575 | 932626 | 932677 | 932727 | 932778 | 932829 | 932879 | 932930 | |
| 7 | 932981 | 933031 | 933082 | 933133 | 933183 | 933234 | 933285 | 933335 | 933386 | 933437 | |
| 8 | 933487 | 933538 | 933589 | 933639 | 933690 | 933740 | 933791 | 933841 | 933892 | 933943 | |
| 9 | 933993 | 934044 | 934094 | 934145 | 934195 | 934246 | 934296 | 934347 | 934397 | 934448 | |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 56 | 5.6 | 11.2 | 16.8 | 22.4 | 28.0 | 33.6 | 39.2 | 44.8 | 50.4 |
| 54 | 5.4 | 10.8 | 16.2 | 21.6 | 27.0 | 32.4 | 37.8 | 43.2 | 48.6 |
| 52 | 5.2 | 10.4 | 15.6 | 20.8 | 26.0 | 31.2 | 36.4 | 41.6 | 46.8 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **860** | **934498** | **934549** | **934599** | **934650** | **934700** | **934751** | **934801** | **934852** | **934902** | **934953** | |
| 1 | 935003 | 935054 | 935104 | 935154 | 935205 | 935255 | 935306 | 935356 | 935406 | 935457 | |
| 2 | 935507 | 935558 | 935608 | 935658 | 935709 | 935759 | 935809 | 935860 | 935910 | 935960 | |
| 3 | 936011 | 936061 | 936111 | 936162 | 936212 | 936262 | 936313 | 936363 | 936413 | 936463 | |
| 4 | 936514 | 936564 | 936614 | 936665 | 936715 | 936765 | 936815 | 936865 | 936916 | 936966 | |
| 5 | 937016 | 937066 | 937116 | 937167 | 937217 | 937267 | 937317 | 937367 | 937418 | 937468 | |
| 6 | 937518 | 937568 | 937618 | 937668 | 937718 | 937769 | 937819 | 937869 | 937919 | 937969 | |
| 7 | 938019 | 938069 | 938119 | 938169 | 938219 | 938269 | 938320 | 938370 | 938420 | 938470 | |
| 8 | 938520 | 938570 | 938620 | 938670 | 938720 | 938770 | 938820 | 938870 | 938920 | 938970 | 50 |
| 9 | 939020 | 939070 | 939120 | 939170 | 939220 | 939270 | 939320 | 939369 | 939419 | 939469 | |
| **870** | **939519** | **939569** | **939619** | **939669** | **939719** | **939769** | **939819** | **939869** | **939918** | **939968** | |
| 1 | 940018 | 940068 | 940118 | 940168 | 940218 | 940267 | 940317 | 940367 | 940417 | 940467 | |
| 2 | 940516 | 940566 | 940616 | 940666 | 940716 | 940765 | 940815 | 940865 | 940915 | 940964 | |
| 3 | 941014 | 941064 | 941114 | 941163 | 941213 | 941263 | 941313 | 941362 | 941412 | 941462 | |
| 4 | 941511 | 941561 | 941611 | 941660 | 941710 | 941760 | 941809 | 941859 | 941909 | 941958 | |
| 5 | 942008 | 942058 | 942107 | 942157 | 942207 | 942256 | 942306 | 942355 | 942405 | 942455 | |
| 6 | 942504 | 942554 | 942603 | 942653 | 942702 | 942752 | 942801 | 942851 | 942901 | 942950 | |
| 7 | 943000 | 943049 | 943099 | 943148 | 943198 | 943247 | 943297 | 943346 | 943396 | 943445 | |
| 8 | 943495 | 943544 | 943593 | 943643 | 943692 | 943742 | 943791 | 943841 | 943890 | 943939 | |
| 9 | 943989 | 944038 | 944088 | 944137 | 944186 | 944236 | 944285 | 944335 | 944384 | 944433 | |
| **880** | **944483** | **944532** | **944581** | **944631** | **944680** | **944729** | **944779** | **944828** | **944877** | **944927** | |
| 1 | 944976 | 945025 | 945074 | 945124 | 945173 | 945222 | 945272 | 945321 | 945370 | 945419 | |
| 2 | 945469 | 945518 | 945567 | 945616 | 945665 | 945715 | 945764 | 945813 | 945862 | 945912 | |
| 3 | 945961 | 946010 | 946059 | 946108 | 946157 | 946207 | 946256 | 946305 | 946354 | 946403 | |
| 4 | 946452 | 946501 | 946551 | 946600 | 946649 | 946698 | 946747 | 946796 | 946845 | 946894 | |
| 5 | 946943 | 946992 | 947041 | 947090 | 947140 | 947189 | 947238 | 947287 | 947336 | 947385 | |
| 6 | 947434 | 947483 | 947532 | 947581 | 947630 | 947679 | 947728 | 947777 | 947826 | 947875 | 49 |
| 7 | 947924 | 947973 | 948022 | 948070 | 948119 | 948168 | 948217 | 948266 | 948315 | 948364 | |
| 8 | 948413 | 948462 | 948511 | 948560 | 948608 | 948657 | 948706 | 948755 | 948804 | 948853 | |
| 9 | 948902 | 948951 | 948999 | 949048 | 949097 | 949146 | 949195 | 949244 | 949292 | 949341 | |
| **890** | **949390** | **949439** | **949488** | **949536** | **949585** | **949634** | **949683** | **949731** | **949780** | **949829** | |
| 1 | 949878 | 949926 | 949975 | 950024 | 950073 | 950121 | 950170 | 950219 | 950267 | 950316 | |
| 2 | 950365 | 950414 | 950462 | 950511 | 950560 | 950608 | 950657 | 950706 | 950754 | 950803 | |
| 3 | 950851 | 950900 | 950949 | 950997 | 951046 | 951095 | 951143 | 951192 | 951240 | 951289 | |
| 4 | 951338 | 951386 | 951435 | 951483 | 951532 | 951580 | 951629 | 951677 | 951726 | 951775 | |
| 5 | 951823 | 951872 | 951920 | 951969 | 952017 | 952066 | 952114 | 952163 | 952211 | 952260 | |
| 6 | 952308 | 952356 | 952405 | 952453 | 952502 | 952550 | 952599 | 952647 | 952696 | 952744 | |
| 7 | 952792 | 952841 | 952889 | 952938 | 952986 | 953034 | 953083 | 953131 | 953180 | 953228 | |
| 8 | 953276 | 953325 | 953373 | 953421 | 953470 | 953518 | 953566 | 953615 | 953663 | 953711 | |
| 9 | 953760 | 953808 | 953856 | 953905 | 953953 | 954001 | 954049 | 954098 | 954146 | 954194 | |
| **900** | **954243** | **954291** | **954339** | **954387** | **954435** | **954484** | **954532** | **954580** | **954628** | **954677** | |
| 1 | 954725 | 954773 | 954821 | 954869 | 954918 | 954966 | 955014 | 955062 | 955110 | 955158 | |
| 2 | 955207 | 955255 | 955303 | 955351 | 955399 | 955447 | 955495 | 955543 | 955592 | 955640 | |
| 3 | 955688 | 955736 | 955784 | 955832 | 955880 | 955928 | 955976 | 956024 | 956072 | 956120 | |
| 4 | 956168 | 956216 | 956265 | 956313 | 956361 | 956409 | 956457 | 956505 | 956553 | 956601 | |
| 5 | 956649 | 956697 | 956745 | 956793 | 956840 | 956888 | 956936 | 956984 | 957032 | 957080 | 48 |
| 6 | 957128 | 957176 | 957224 | 957272 | 957320 | 957368 | 957416 | 957464 | 957512 | 957559 | |
| 7 | 957607 | 957655 | 957703 | 957751 | 957799 | 957847 | 957894 | 957942 | 957990 | 958038 | |
| 8 | 958086 | 958134 | 958181 | 958229 | 958277 | 958325 | 958373 | 958421 | 958468 | 958516 | |
| 9 | 958564 | 958612 | 958659 | 958707 | 958755 | 958803 | 958850 | 958898 | 958946 | 958994 | |
| **910** | **959041** | **959089** | **959137** | **959185** | **959232** | **959280** | **959328** | **959375** | **959423** | **959471** | |
| 1 | 959518 | 959566 | 959614 | 959661 | 959709 | 959757 | 959804 | 959852 | 959900 | 959947 | |
| 2 | 959995 | 960042 | 960090 | 960138 | 960185 | 960233 | 960280 | 960328 | 960376 | 960423 | |
| 3 | 960471 | 960518 | 960566 | 960613 | 960661 | 960709 | 960756 | 960804 | 960851 | 960899 | |
| 4 | 960946 | 960994 | 961041 | 961089 | 961136 | 961184 | 961231 | 961279 | 961326 | 961374 | |
| 5 | 961421 | 961469 | 961516 | 961563 | 961611 | 961658 | 961706 | 961753 | 961801 | 961848 | |
| 6 | 961895 | 961943 | 961990 | 962038 | 962085 | 962132 | 962180 | 962227 | 962275 | 962322 | |
| 7 | 962369 | 962417 | 962464 | 962511 | 962559 | 962606 | 962653 | 962701 | 962748 | 962795 | |
| 8 | 962843 | 962890 | 962937 | 962985 | 963032 | 963079 | 963126 | 963174 | 963221 | 963268 | |
| 9 | 963316 | 963363 | 963410 | 963457 | 963504 | 963552 | 963599 | 963646 | 963693 | 963741 | |

### PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 52 | 5.2 | 10.4 | 15.6 | 20.8 | 26.0 | 31.2 | 36.4 | 41.6 | 46.8 |
| 50 | 5.0 | 10.0 | 15.0 | 20.0 | 25.0 | 30.0 | 35.0 | 40.0 | 45.0 |
| 48 | 4.8 | 9.6 | 14.4 | 19.2 | 24.0 | 28.8 | 33.6 | 38.4 | 43.2 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **920** | **963788** | **963835** | **963882** | **963929** | **963977** | **964024** | **964071** | **964118** | **964165** | **964212** | |
| 1 | 964260 | 964307 | 964354 | 964401 | 964448 | 964495 | 964542 | 964590 | 964637 | 964684 | |
| 2 | 964731 | 964778 | 964825 | 964872 | 964919 | 964966 | 965013 | 965061 | 965108 | 965155 | |
| 3 | 965202 | 965249 | 965296 | 965343 | 965390 | 965437 | 965484 | 965531 | 965578 | 965625 | |
| 4 | 965672 | 965719 | 965766 | 965813 | 965860 | 965907 | 965954 | 966001 | 966048 | 966095 | **47** |
| 5 | 966142 | 966189 | 966236 | 966283 | 966329 | 966376 | 966423 | 966470 | 966517 | 966564 | |
| 6 | 966611 | 966658 | 966705 | 966752 | 966799 | 966845 | 966892 | 966939 | 966986 | 967033 | |
| 7 | 967080 | 967127 | 967173 | 967220 | 967267 | 967314 | 967361 | 967408 | 967454 | 967501 | |
| 8 | 967548 | 967595 | 967642 | 967688 | 967735 | 967782 | 967829 | 967875 | 967922 | 967969 | |
| 9 | 968016 | 968062 | 968109 | 968156 | 968203 | 968249 | 968296 | 968343 | 968390 | 968436 | |
| **930** | **968483** | **968530** | **968576** | **968623** | **968670** | **968716** | **968763** | **968810** | **968856** | **968903** | |
| 1 | 968950 | 968996 | 969043 | 969090 | 969136 | 969183 | 969229 | 969276 | 969323 | 969369 | |
| 2 | 969416 | 969463 | 969509 | 969556 | 969602 | 969649 | 969695 | 969742 | 969789 | 969835 | |
| 3 | 969882 | 969928 | 969975 | 970021 | 970068 | 970114 | 970161 | 970207 | 970254 | 970300 | |
| 4 | 970347 | 970393 | 970440 | 970486 | 970533 | 970579 | 970626 | 970672 | 970719 | 970765 | |
| 5 | 970812 | 970858 | 970904 | 970951 | 970997 | 971044 | 971090 | 971137 | 971183 | 971229 | |
| 6 | 971276 | 971322 | 971369 | 971415 | 971461 | 971508 | 971554 | 971601 | 971647 | 971693 | |
| 7 | 971740 | 971786 | 971832 | 971879 | 971925 | 971971 | 972018 | 972064 | 972110 | 972157 | |
| 8 | 972203 | 972249 | 972295 | 972342 | 972388 | 972434 | 972481 | 972527 | 972573 | 972619 | |
| 9 | 972666 | 972712 | 972758 | 972804 | 972851 | 972897 | 972943 | 972989 | 973035 | 973082 | |
| **940** | **973128** | **973174** | **973220** | **973266** | **973313** | **973359** | **973405** | **973451** | **973497** | **973543** | |
| 1 | 973590 | 973636 | 973682 | 973728 | 973774 | 973820 | 973866 | 973913 | 973959 | 974005 | |
| 2 | 974051 | 974097 | 974143 | 974189 | 974235 | 974281 | 974327 | 974374 | 974420 | 974466 | |
| 3 | 974512 | 974558 | 974604 | 974650 | 974696 | 974742 | 974788 | 974834 | 974880 | 974926 | |
| 4 | 974972 | 975018 | 975064 | 975110 | 975156 | 975202 | 975248 | 975294 | 975340 | 975386 | **46** |
| 5 | 975432 | 975478 | 975524 | 975570 | 975616 | 975662 | 975707 | 975753 | 975799 | 975845 | |
| 6 | 975891 | 975937 | 975983 | 976029 | 976075 | 976121 | 976167 | 976212 | 976258 | 976304 | |
| 7 | 976350 | 976396 | 976442 | 976488 | 976533 | 976579 | 976625 | 976671 | 976717 | 976763 | |
| 8 | 976808 | 976854 | 976900 | 976946 | 976992 | 977037 | 977083 | 977129 | 977175 | 977220 | |
| 9 | 977266 | 977312 | 977358 | 977403 | 977449 | 977495 | 977541 | 977586 | 977632 | 977678 | |
| **950** | **977724** | **977769** | **977815** | **977661** | **977906** | **977952** | **977998** | **978043** | **978089** | **978135** | |
| 1 | 978181 | 978226 | 978272 | 978317 | 978363 | 978409 | 978454 | 978500 | 978546 | 978591 | |
| 2 | 978637 | 978683 | 978728 | 978774 | 978819 | 978865 | 978911 | 978956 | 979002 | 979047 | |
| 3 | 979093 | 979138 | 979184 | 979230 | 979275 | 979321 | 979366 | 979412 | 979457 | 979503 | |
| 4 | 979548 | 979594 | 979639 | 979685 | 979730 | 979776 | 979821 | 979867 | 979912 | 979958 | |
| 5 | 980003 | 980049 | 980094 | 980140 | 980185 | 980231 | 980276 | 980322 | 980367 | 980412 | |
| 6 | 980458 | 980503 | 980549 | 980594 | 980640 | 980685 | 980730 | 980776 | 980821 | 980867 | |
| 7 | 980912 | 980957 | 981003 | 981048 | 981093 | 981139 | 981184 | 981229 | 981275 | 981320 | |
| 8 | 981366 | 981411 | 981456 | 981501 | 981547 | 981592 | 981637 | 981683 | 981728 | 981773 | |
| 9 | 981819 | 981864 | 981909 | 981954 | 982000 | 982045 | 982090 | 982135 | 982181 | 982226 | |
| **960** | **982271** | **982316** | **982362** | **982407** | **982452** | **982497** | **982543** | **982588** | **982633** | **982678** | |
| 1 | 982723 | 982769 | 982814 | 982859 | 982904 | 982949 | 982994 | 983040 | 983085 | 983130 | |
| 2 | 983175 | 983220 | 983265 | 983310 | 983356 | 983401 | 983446 | 983491 | 983536 | 983581 | |
| 3 | 983626 | 983671 | 983716 | 983762 | 983807 | 983852 | 983897 | 983942 | 983987 | 984032 | |
| 4 | 984077 | 984122 | 984167 | 984212 | 984257 | 984302 | 984347 | 984392 | 984437 | 984482 | |
| 5 | 984527 | 984572 | 984617 | 984662 | 984707 | 984752 | 984797 | 984842 | 984887 | 984932 | **45** |
| 6 | 984977 | 985022 | 985067 | 985112 | 985157 | 985202 | 985247 | 985292 | 985337 | 985382 | |
| 7 | 985426 | 985471 | 985516 | 985561 | 985606 | 985651 | 985696 | 985741 | 985786 | 985830 | |
| 8 | 985875 | 985920 | 985965 | 986010 | 986055 | 986100 | 986144 | 986189 | 986234 | 986279 | |
| 9 | 986324 | 986369 | 986413 | 986458 | 986503 | 986548 | 986593 | 986637 | 986682 | 986727 | |
| **970** | **986772** | **986817** | **986861** | **986906** | **986951** | **986996** | **987040** | **987085** | **987130** | **987175** | |
| 1 | 987219 | 987264 | 987309 | 987353 | 987398 | 987443 | 987488 | 987532 | 987577 | 987622 | |
| 2 | 987666 | 987711 | 987756 | 987800 | 987845 | 987890 | 987934 | 987979 | 988024 | 988068 | |
| 3 | 988113 | 988157 | 988202 | 988247 | 988291 | 988336 | 988381 | 988425 | 988470 | 988514 | |
| 4 | 988559 | 988604 | 988648 | 988693 | 988737 | 988782 | 988826 | 988871 | 988916 | 988960 | |
| 5 | 989005 | 989049 | 989094 | 989138 | 989183 | 989227 | 989272 | 989316 | 989361 | 989405 | |
| 6 | 989450 | 989494 | 989539 | 989583 | 989628 | 989672 | 989717 | 989761 | 989806 | 989850 | |
| 7 | 989895 | 989939 | 989983 | 990028 | 990072 | 990117 | 990161 | 990206 | 990250 | 990294 | |
| 8 | 990339 | 990383 | 990428 | 990472 | 990516 | 990561 | 990605 | 990650 | 990694 | 990738 | |
| 9 | 990783 | 990827 | 990871 | 990916 | 990960 | 991004 | 991049 | 991093 | 991137 | 991182 | |

PROPORTIONAL PARTS

| Diff. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| 48 | 4.8 | 9.6 | 14.4 | 19.2 | 24.0 | 28.8 | 33.6 | 38.4 | 43.2 |
| 46 | 4.6 | 9.2 | 13.8 | 18.4 | 23.0 | 27.6 | 32.2 | 36.8 | 41.4 |
| 44 | 4.4 | 8.8 | 13.2 | 17.6 | 22.0 | 26.4 | 30.8 | 35.2 | 39.6 |
| 42 | 4.2 | 8.4 | 12.6 | 16.8 | 21.0 | 25.2 | 29.4 | 33.6 | 37.8 |

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | Diff. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **980** | **991226** | **991270** | **991315** | **991359** | **991403** | **991448** | **991492** | **991536** | **991580** | **991625** | |
| 1 | 991669 | 991713 | 991758 | 991802 | 991846 | 991890 | 991935 | 991979 | 992023 | 992067 | |
| 2 | 992111 | 992156 | 992200 | 992244 | 992288 | 992333 | 992377 | 992421 | 992465 | 992509 | |
| 3 | 992554 | 992598 | 992642 | 992686 | 992730 | 992774 | 992819 | 992863 | 992907 | 992951 | |
| 4 | 992995 | 993039 | 993083 | 993127 | 993172 | 993216 | 993260 | 993304 | 993348 | 993392 | |
| 5 | 993436 | 993480 | 993524 | 993568 | 993613 | 993657 | 993701 | 993745 | 993789 | 993833 | |
| 6 | 993877 | 993921 | 993965 | 994009 | 994053 | 994097 | 994141 | 994185 | 994229 | 994273 | |
| 7 | 994317 | 994361 | 994405 | 994449 | 994493 | 994537 | 994581 | 994625 | 994669 | 994713 | **44** |
| 8 | 994757 | 994801 | 994845 | 994889 | 994933 | 994977 | 995021 | 995065 | 995108 | 995152 | |
| 9 | 995196 | 995240 | 995284 | 995328 | 995372 | 995416 | 995460 | 995504 | 995547 | 995591 | |
| **990** | **995635** | **995679** | **995723** | **995767** | **995811** | **995854** | **995898** | **995942** | **995986** | **996030** | |
| 1 | 996074 | 996117 | 996161 | 996205 | 996249 | 996293 | 996337 | 996380 | 996424 | 996468 | |
| 2 | 996512 | 996555 | 996599 | 996643 | 996687 | 996731 | 996774 | 996818 | 996862 | 996906 | |
| 3 | 996949 | 996993 | 997037 | 997080 | 997124 | 997168 | 997212 | 997255 | 997299 | 997343 | |
| 4 | 997386 | 997430 | 997474 | 997517 | 997561 | 997605 | 997648 | 997692 | 997736 | 997779 | |
| 5 | 997823 | 997867 | 997910 | 997954 | 997998 | 998041 | 998085 | 998129 | 998172 | 998216 | |
| 6 | 998259 | 998303 | 998347 | 998390 | 998434 | 998477 | 998521 | 998564 | 998608 | 998652 | |
| 7 | 998695 | 998739 | 998782 | 998826 | 998869 | 998913 | 998956 | 999000 | 999043 | 999087 | |
| 8 | 999131 | 999174 | 999218 | 999261 | 999305 | 999348 | 999392 | 999435 | 999479 | 999522 | |
| 9 | 999565 | 999609 | 999652 | 999696 | 999739 | 999783 | 999826 | 999870 | 999913 | 999957 | |
| **1000** | **000000** | **000043** | **000087** | **000130** | **000174** | **000217** | **000260** | **000304** | **000347** | **000391** | 43 |

## Table 3.—Natural (Napierian or Hyperbolic) Logarithms of Numbers

Table gives natural logarithms of numbers from 1.0 to 9.99 directly. To find logarithms of numbers outside that range add or subtract natural logarithm of the powers of 10 as follows:

$\log_e 10 = 2.302585$ $\log_e 10^4 = 9.210340$ $\log_e 10^7 = 16.118096$
$\log_e 10^2 = 4.605170$ $\log_e 10^5 = 11.512925$ $\log_e 10^8 = 18.420681$
$\log_e 10^3 = 6.907755$ $\log_e 10^6 = 13.815511$ $\log_e 10^9 = 20.703266$

**Example.**—$\log_e 679. = \log_e 6.79 + \log_e 10^2 = 1.9155 + 4.6052 = 6.5207$
$\log_e .0679 = \log_e 6.79 - \log_e 10^2 = 1.9155 - 4.6052 = -2.6897$

The common logarithm is the natural logarithm multiplied by the modulus of $\log_{10}$: $\log_{10} N = 0.434294 \log_e N$.

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|---|
| **1.0** | **0.0000** | **0.0100** | **0.0198** | **0.0296** | **0.0392** | **0.0488** | **0.0583** | **0.0677** | **0.0770** | **0.0862** |
| 1.1 | 0.0953 | 0.1044 | 0.1133 | 0.1222 | 0.1310 | 0.1398 | 0.1484 | 0.1570 | 0.1655 | 0.1740 |
| 1.2 | 0.1823 | 0.1906 | 0.1989 | 0.2070 | 0.2151 | 0.2231 | 0.2311 | 0.2390 | 0.2469 | 0.2546 |
| 1.3 | 0.2624 | 0.2700 | 0.2776 | 0.2852 | 0.2927 | 0.3001 | 0.3075 | 0.3148 | 0.3221 | 0.3293 |
| 1.4 | 0.3365 | 0.3436 | 0.3507 | 0.3577 | 0.3646 | 0.3716 | 0.3784 | 0.3853 | 0.3920 | 0.3988 |
| 1.5 | 0.4055 | 0.4121 | 0.4187 | 0.4253 | 0.4318 | 0.4383 | 0.4447 | 0.4511 | 0.4574 | 0.4637 |
| 1.6 | 0.4700 | 0.4762 | 0.4824 | 0.4886 | 0.4947 | 0.5008 | 0.5068 | 0.5128 | 0.5188 | 0.5247 |
| 1.7 | 0.5306 | 0.5365 | 0.5423 | 0.5481 | 0.5539 | 0.5596 | 0.5653 | 0.5710 | 0.5766 | 0.5822 |
| 1.8 | 0.5878 | 0.5933 | 0.5988 | 0.6043 | 0.6098 | 0.6152 | 0.6206 | 0.6259 | 0.6313 | 0.6366 |
| 1.9 | 0.6419 | 0.6471 | 0.6523 | 0.6575 | 0.6627 | 0.6678 | 0.6729 | 0.6780 | 0.6831 | 0.6881 |
| **2.0** | **0.6931** | **0.6981** | **0.7031** | **0.7080** | **0.7129** | **0.7178** | **0.7227** | **0.7275** | **0.7324** | **0.7372** |
| 2.1 | 0.7419 | 0.7467 | 0.7514 | 0.7561 | 0.7608 | 0.7655 | 0.7701 | 0.7747 | 0.7793 | 0.7839 |
| 2.2 | 0.7885 | 0.7930 | 0.7975 | 0.8020 | 0.8065 | 0.8109 | 0.8154 | 0.8198 | 0.8242 | 0.8286 |
| 2.3 | 0.8329 | 0.8372 | 0.8416 | 0.8459 | 0.8502 | 0.8544 | 0.8587 | 0.8629 | 0.8671 | 0.8713 |
| 2.4 | 0.8755 | 0.8796 | 0.8838 | 0.8879 | 0.8920 | 0.8961 | 0.9002 | 0.9042 | 0.9083 | 0.9123 |
| 2.5 | 0.9163 | 0.9203 | 0.9243 | 0.9282 | 0.9322 | 0.9361 | 0.9400 | 0.9439 | 0.9478 | 0.9517 |
| 2.6 | 0.9555 | 0.9594 | 0.9632 | 0.9670 | 0.9708 | 0.9746 | 0.9783 | 0.9821 | 0.9858 | 0.9895 |
| 2.7 | 0.9933 | 0.9969 | 1.0006 | 1.0043 | 1.0080 | 1.0116 | 1.0152 | 1.0188 | 1.0225 | 1.0260 |
| 2.8 | 1.0296 | 1.0332 | 1.0367 | 1.0403 | 1.0438 | 1.0473 | 1.0508 | 1.0543 | 1.0578 | 1.0613 |
| 2.9 | 1.0647 | 1.0682 | 1.0716 | 1.0750 | 1.0784 | 1.0818 | 1.0852 | 1.0886 | 1.0919 | 1.0953 |
| **3.0** | **1.0986** | **1.1019** | **1.1053** | **1.1086** | **1.1119** | **1.1151** | **1.1184** | **1.1217** | **1.1249** | **1.1282** |
| 3.1 | 1.1314 | 1.1346 | 1.1378 | 1.1410 | 1.1442 | 1.1474 | 1.1506 | 1.1537 | 1.1569 | 1.1600 |
| 3.2 | 1.1632 | 1.1663 | 1.1694 | 1.1725 | 1.1756 | 1.1787 | 1.1817 | 1.1848 | 1.1878 | 1.1909 |
| 3.3 | 1.1939 | 1.1969 | 1.2000 | 1.2030 | 1.2060 | 1.2090 | 1.2119 | 1.2149 | 1.2179 | 1.2208 |
| 3.4 | 1.2238 | 1.2267 | 1.2296 | 1.2326 | 1.2355 | 1.2384 | 1.2413 | 1.2442 | 1.2470 | 1.2499 |
| 3.5 | 1.2528 | 1.2556 | 1.2585 | 1.2613 | 1.2641 | 1.2669 | 1.2698 | 1.2726 | 1.2754 | 1.2782 |
| 3.6 | 1.2809 | 1.2837 | 1.2865 | 1.2892 | 1.2920 | 1.2947 | 1.2975 | 1.3002 | 1.3029 | 1.3056 |
| 3.7 | 1.3083 | 1.3110 | 1.3137 | 1.3164 | 1.3191 | 1.3218 | 1.3244 | 1.3271 | 1.3297 | 1.3324 |
| 3.8 | 1.3350 | 1.3376 | 1.3403 | 1.3429 | 1.3455 | 1.3481 | 1.3507 | 1.3533 | 1.3558 | 1.3584 |
| 3.9 | 1.3610 | 1.3635 | 1.3661 | 1.3686 | 1.3712 | 1.3737 | 1.3762 | 1.3788 | 1.3813 | 1.3838 |

*(Table continued on following page)*

**Table 3.—Natural (Napierian or Hyperbolic) Logarithms of Numbers** (*Continued*)

| N | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|---|
| **4.0** | **1.3863** | **1.3888** | **1.3913** | **1.3938** | **1.3962** | **1.3987** | **1.4012** | **1.4036** | **1.4061** | **1.4085** |
| 4.1 | 1.4110 | 1.4134 | 1.4159 | 1.4183 | 1.4207 | 1.4231 | 1.4255 | 1.4279 | 1.4303 | 1.4327 |
| 4.2 | 1.4351 | 1.4375 | 1.4398 | 1.4422 | 1.4446 | 1.4469 | 1.4493 | 1.4516 | 1.4540 | 1.4563 |
| 4.3 | 1.4586 | 1.4609 | 1.4633 | 1.4656 | 1.4679 | 1.4702 | 1.4725 | 1.4748 | 1.4770 | 1.4793 |
| 4.4 | 1.4816 | 1.4839 | 1.4861 | 1.4884 | 1.4907 | 1.4929 | 1.4951 | 1.4974 | 1.4996 | 1.5019 |
| 4.5 | 1.5041 | 1.5063 | 1.5085 | 1.5107 | 1.5129 | 1.5151 | 1.5173 | 1.5195 | 1.5217 | 1.5239 |
| 4.6 | 1.5261 | 1.5282 | 1.5304 | 1.5326 | 1.5347 | 1.5369 | 1.5390 | 1.5412 | 1.5433 | 1.5454 |
| 4.7 | 1.5476 | 1.5497 | 1.5518 | 1.5539 | 1.5560 | 1.5581 | 1.5602 | 1.5623 | 1.5644 | 1.5665 |
| 4.8 | 1.5686 | 1.5707 | 1.5728 | 1.5748 | 1.5769 | 1.5790 | 1.5810 | 1.5831 | 1.5851 | 1.5872 |
| 4.9 | 1.5892 | 1.5913 | 1.5933 | 1.5953 | 1.5974 | 1.5994 | 1.6014 | 1.6034 | 1.6054 | 1.6074 |
| **5.0** | **1.6094** | **1.6114** | **1.6134** | **1.6154** | **1.6174** | **1.6194** | **1.6214** | **1.6233** | **1.6253** | **1.6273** |
| 5.1 | 1.6292 | 1.6312 | 1.6332 | 1.6351 | 1.6371 | 1.6390 | 1.6409 | 1.6429 | 1.6448 | 1.6467 |
| 5.2 | 1.6487 | 1.6506 | 1.6525 | 1.6544 | 1.6563 | 1.6582 | 1.6601 | 1.6620 | 1.6639 | 1.6658 |
| 5.3 | 1.6677 | 1.6696 | 1.6715 | 1.6734 | 1.6752 | 1.6771 | 1.6790 | 1.6808 | 1.6827 | 1.6845 |
| 5.4 | 1.6864 | 1.6882 | 1.6901 | 1.6919 | 1.6938 | 1.6956 | 1.6974 | 1.6993 | 1.7011 | 1.7029 |
| 5.5 | 1.7047 | 1.7066 | 1.7084 | 1.7102 | 1.7120 | 1.7138 | 1.7156 | 1.7174 | 1.7192 | 1.7210 |
| 5.6 | 1.7228 | 1.7246 | 1.7263 | 1.7281 | 1.7299 | 1.7317 | 1.7334 | 1.7352 | 1.7370 | 1.7387 |
| 5.7 | 1.7405 | 1.7422 | 1.7440 | 1.7457 | 1.7475 | 1.7492 | 1.7509 | 1.7527 | 1.7544 | 1.7561 |
| 5.8 | 1.7579 | 1.7596 | 1.7613 | 1.7630 | 1.7647 | 1.7664 | 1.7681 | 1.7699 | 1.7716 | 1.7733 |
| 5.9 | 1.7750 | 1.7766 | 1.7783 | 1.7800 | 1.7817 | 1.7834 | 1.7851 | 1.7867 | 1.7884 | 1.7901 |
| **6.0** | **1.7918** | **1.7934** | **1.7951** | **1.7967** | **1.7984** | **1.8001** | **1.8017** | **1.8034** | **1.8050** | **1.8066** |
| 6.1 | 1.8083 | 1.8099 | 1.8116 | 1.8132 | 1.8148 | 1.8165 | 1.8181 | 1.8197 | 1.8213 | 1.8229 |
| 6.2 | 1.8245 | 1.8262 | 1.8278 | 1.8294 | 1.8310 | 1.8326 | 1.8342 | 1.8358 | 1.8374 | 1.8390 |
| 6.3 | 1.8405 | 1.8421 | 1.8437 | 1.8453 | 1.8469 | 1.8485 | 1.8500 | 1.8516 | 1.8532 | 1.8547 |
| 6.4 | 1.8563 | 1.8579 | 1.8594 | 1.8610 | 1.8625 | 1.8641 | 1.8656 | 1.8672 | 1.8687 | 1.8703 |
| 6.5 | 1.8718 | 1.8733 | 1.8749 | 1.8764 | 1.8779 | 1.8795 | 1.8810 | 1.8825 | 1.8840 | 1.8856 |
| 6.6 | 1.8871 | 1.8886 | 1.8901 | 1.8916 | 1.8931 | 1.8946 | 1.8961 | 1.8976 | 1.8991 | 1.9006 |
| 6.7 | 1.9021 | 1.9036 | 1.9051 | 1.9066 | 1.9081 | 1.9095 | 1.9110 | 1.9125 | 1.9140 | 1.9155 |
| 6.8 | 1.9169 | 1.9184 | 1.9199 | 1.9213 | 1.9228 | 1.9242 | 1.9257 | 1.9272 | 1.9286 | 1.9301 |
| 6.9 | 1.9315 | 1.9330 | 1.9344 | 1.9359 | 1.9373 | 1.9387 | 1.9402 | 1.9416 | 1.9430 | 1.9445 |
| **7.0** | **1.9459** | **1.9473** | **1.9488** | **1.9502** | **1.9516** | **1.9530** | **1.9544** | **1.9559** | **1.9573** | **1.9587** |
| 7.1 | 1.9601 | 1.9615 | 1.9629 | 1.9643 | 1.9657 | 1.9671 | 1.9685 | 1.9699 | 1.9713 | 1.9727 |
| 7.2 | 1.9741 | 1.9755 | 1.9769 | 1.9782 | 1.9796 | 1.9810 | 1.9824 | 1.9838 | 1.9851 | 1.9865 |
| 7.3 | 1.9879 | 1.9892 | 1.9906 | 1.9920 | 1.9933 | 1.9947 | 1.9961 | 1.9974 | 1.9988 | 2.0001 |
| 7.4 | 2.0015 | 2.0028 | 2.0042 | 2.0055 | 2.0069 | 2.0082 | 2.0096 | 2.0109 | 2.0122 | 2.0136 |
| 7.5 | 2.0149 | 2.0162 | 2.0176 | 2.0189 | 2.0202 | 2.0215 | 2.0229 | 2.0242 | 2.0255 | 2.0268 |
| 7.6 | 2.0281 | 2.0295 | 2.0308 | 2.0321 | 2.0334 | 2.0347 | 2.0360 | 2.0373 | 2.0386 | 2.0399 |
| 7.7 | 2.0412 | 2.0425 | 2.0438 | 2.0451 | 2.0464 | 2.0477 | 2.0490 | 2.0503 | 2.0516 | 2.0528 |
| 7.8 | 2.0541 | 2.0554 | 2.0567 | 2.0580 | 2.0592 | 2.0605 | 2.0618 | 2.0631 | 2.0643 | 2.0656 |
| 7.9 | 2.0669 | 2.0681 | 2.0694 | 2.0707 | 2.0719 | 2.0732 | 2.0744 | 2.0757 | 2.0769 | 2.0782 |
| **8.0** | **2.0794** | **2.0807** | **2.0819** | **2.0832** | **2.0844** | **2.0857** | **2.0869** | **2.0882** | **2.0894** | **2.0906** |
| 8.1 | 2.0919 | 2.0931 | 2.0943 | 2.0956 | 2.0968 | 2.0980 | 2.0992 | 2.1005 | 2.1017 | 2.1029 |
| 8.2 | 2.1041 | 2.1054 | 2.1066 | 2.1078 | 2.1090 | 2.1102 | 2.1114 | 2.1126 | 2.1138 | 2.1150 |
| 8.3 | 2.1163 | 2.1175 | 2.1187 | 2.1199 | 2.1211 | 2.1223 | 2.1235 | 2.1247 | 2.1258 | 2.1270 |
| 8.4 | 2.1282 | 2.1294 | 2.1306 | 2.1318 | 2.1330 | 2.1342 | 2.1353 | 2.1365 | 2.1377 | 2.1389 |
| 8.5 | 2.1401 | 2.1412 | 2.1424 | 2.1436 | 2.1448 | 2.1459 | 2.1471 | 2.1483 | 2.1494 | 2.1506 |
| 8.6 | 2.1518 | 2.1529 | 2.1541 | 2.1552 | 2.1564 | 2.1576 | 2.1587 | 2.1599 | 2.1610 | 2.1622 |
| 8.7 | 2.1633 | 2.1645 | 2.1656 | 2.1668 | 2.1679 | 2.1691 | 2.1702 | 2.1713 | 2.1725 | 2.1736 |
| 8.8 | 2.1748 | 2.1759 | 2.1770 | 2.1782 | 2.1793 | 2.1804 | 2.1815 | 2.1827 | 2.1838 | 2.1849 |
| 8.9 | 2.1861 | 2.1872 | 2.1883 | 2.1894 | 2.1905 | 2.1917 | 2.1928 | 2.1939 | 2.1950 | 2.1961 |
| **9.0** | **2.1972** | **2.1983** | **2.1994** | **2.2006** | **2.2017** | **2.2028** | **2.2039** | **2.2050** | **2.2061** | **2.2072** |
| 9.1 | 2.2083 | 2.2094 | 2.2105 | 2.2116 | 2.2127 | 2.2138 | 2.2148 | 2.2159 | 2.2170 | 2.2181 |
| 9.2 | 2.2192 | 2.2203 | 2.2214 | 2.2225 | 2.2235 | 2.2246 | 2.2257 | 2.2268 | 2.2279 | 2.2289 |
| 9.3 | 2.2300 | 2.2311 | 2.2322 | 2.2332 | 2.2343 | 2.2354 | 2.2364 | 2.2375 | 2.2386 | 2.2396 |
| 9.4 | 2.2407 | 2.2418 | 2.2428 | 2.2439 | 2.2450 | 2.2460 | 2.2471 | 2.2481 | 2.2492 | 2.2502 |
| 9.5 | 2.2513 | 2.2523 | 2.2534 | 2.2544 | 2.2555 | 2.2565 | 2.2576 | 2.2586 | 2.2597 | 2.2607 |
| 9.6 | 2.2618 | 2.2628 | 2.2638 | 2.2649 | 2.2659 | 2.2670 | 2.2680 | 2.2690 | 2.2701 | 2.2711 |
| 9.7 | 2.2721 | 2.2732 | 2.2742 | 2.2752 | 2.2762 | 2.2773 | 2.2783 | 2.2793 | 2.2803 | 2.2814 |
| 9.8 | 2.2824 | 2.2834 | 2.2844 | 2.2854 | 2.2865 | 2.2875 | 2.2885 | 2.2895 | 2.2905 | 2.2915 |
| 9.9 | 2.2925 | 2.2935 | 2.2946 | 2.2956 | 2.2966 | 2.2976 | 2.2986 | 2.2996 | 2.3006 | 2.3016 |

### Table 4.—Properties of Numbers

**Decimal Equivalents, Squares, Cubes, Three-halves Powers, Square Roots, Cube Roots, Fifth Roots, Reciprocals, Circumference and Area of Circles**

| Number, $N$ | | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$=Diam.) | |
|---|---|---|---|---|---|---|---|---|---|---|
| Fraction | Decimal | | | | | | | | Circum. | Area |
| 1/64 | .015625 | 0.000244 | $.381\times10^{-5}$ | .1250 | .2500 | .00195 | .4353 | 64.0 | 0.04909 | .00019 |
| 1/32 | .03125 | .000977 | $.305\times10^{-4}$ | .1768 | .3150 | .00552 | .5000 | 32.0 | .09818 | .00077 |
| 3/64 | .046875 | .002197 | $.103\times10^{-3}$ | .2165 | .3606 | .01015 | .5422 | 18.8235 | .14726 | .00173 |
| **1/16** | **.0625** | **.003906** | $\mathbf{.244\times10^{-3}}$ | **.2500** | **.3969** | **.01563** | **.5744** | **16.0** | **.19635** | **.00307** |
| 5/64 | .078125 | .006104 | $.477\times10^{-3}$ | .2795 | .4275 | .02184 | .6006 | 12.80 | .24544 | .00479 |
| 3/32 | .09375 | .008789 | $.824\times10^{-3}$ | .3062 | .4543 | .02871 | .6229 | 10.6667 | .29452 | .00690 |
| | .10 | .010 | .00100 | .3162 | .4642 | .03162 | .6310 | 10.0 | .31416 | .00785 |
| 7/64 | .109375 | .01196 | .001308 | .3307 | .4782 | .03617 | .6424 | 9.1429 | .34361 | .00939 |
| **1/8** | **.125** | **.01563** | **.001953** | **.3536** | **.5000** | **.04419** | **.6598** | **8.0** | **.39270** | **.01227** |
| 9/64 | .140625 | .01978 | .002782 | .3750 | .5200 | .05273 | .6755 | 7.1111 | .44179 | .01554 |
| 5/32 | .15625 | .02441 | .003814 | .3953 | .5386 | .06176 | .6899 | 6.40 | .49087 | .01917 |
| 11/64 | .171875 | .02954 | .005077 | .4146 | .5560 | .07126 | .7031 | 5.8182 | .53996 | .02320 |
| **3/16** | **.1875** | **.03516** | **.006592** | **.4330** | **.5724** | **.08119** | **.7155** | **5.3333** | **.58905** | **.02761** |
| | .20 | .040 | .0080 | .4472 | .5848 | .08944 | .7248 | 5.0 | .62832 | .03142 |
| 13/64 | .203125 | .04126 | .008381 | .4507 | .5878 | .09155 | .7270 | 4.9231 | .63814 | .03241 |
| 7/32 | .21875 | .04785 | .01047 | .4677 | .6025 | .10231 | .7379 | 4.5714 | .68722 | .03758 |
| 15/64 | .234375 | .05493 | .01287 | .4841 | .6166 | .11347 | .7481 | 4.2667 | .73631 | .04314 |
| **1/4** | **.250** | **.0625** | **.01563** | **.5000** | **.6300** | **.12500** | **.7579** | **4.0** | **.78540** | **.04909** |
| 17/64 | .265625 | .07056 | .01874 | .5154 | .6428 | .13690 | .7671 | 3.7647 | .83448 | .05542 |
| 9/32 | .28125 | .07910 | .02225 | .5303 | .6552 | .14916 | .7759 | 3.5556 | .88357 | .06213 |
| 19/64 | .296875 | .08813 | .02616 | .5449 | .6671 | .16176 | .7844 | 3.3684 | .93266 | .06922 |
| | .30 | .090 | .0270 | .5477 | .6694 | .16432 | .7860 | 3.3333 | .94248 | .07069 |
| **5/16** | **.3125** | **.09766** | **.03052** | **.5590** | **.6786** | **.17469** | **.7925** | **3.2000** | **.98175** | **.07670** |
| 21/64 | .328125 | .10767 | .03533 | .5728 | .6897 | .18796 | .8002 | 3.0476 | 1.0308 | .08456 |
| 11/32 | .34375 | .11816 | .04062 | .5863 | .7005 | .20154 | .8077 | 2.9091 | 1.0799 | .09281 |
| 23/64 | .359375 | .12915 | .04641 | .5995 | .7110 | .21544 | .8149 | 2.7826 | 1.1290 | .10143 |
| **3/8** | **.375** | **.14063** | **.05273** | **.6124** | **.7211** | **.22964** | **.8219** | **2.6667** | **1.1781** | **.11045** |
| 25/64 | .390625 | .15259 | .05961 | .6250 | .7310 | .24414 | .8286 | 2.5600 | 1.2272 | .11984 |
| | .40 | .16 | .0640 | .6325 | .7368 | .25298 | .8326 | 2.50 | 1.2566 | .12566 |
| 13/32 | .40625 | .16504 | .06705 | .6374 | .7406 | .25894 | .8351 | 2.4615 | 1.2763 | .12962 |
| 27/64 | .421875 | .17798 | .07508 | .6495 | .7500 | .27402 | .8415 | 2.3704 | 1.3254 | .13979 |
| **7/16** | **.4375** | **.19141** | **.08374** | **.6614** | **.7592** | **.28938** | **.8476** | **2.2857** | **1.3744** | **.15033** |
| 29/64 | .453125 | .20532 | .09304 | .6732 | .7681 | .30502 | .8536 | 2.2069 | 1.4235 | .16126 |
| 15/32 | .46875 | .21973 | .10300 | .6847 | .7768 | .32093 | .8594 | 2.1333 | 1.4726 | .17257 |
| 31/64 | .484375 | .23462 | .11364 | .6960 | .7854 | .33711 | .8650 | 2.0645 | 1.5217 | .18427 |
| **1/2** | **.50** | **.2500** | **.12500** | **.7071** | **.7937** | **.35355** | **.8706** | **2.0** | **1.5708** | **.19635** |
| 33/64 | .515625 | .26587 | .13709 | .7181 | .8019 | .37025 | .8759 | 1.9394 | 1.6199 | .20881 |
| 17/32 | .53125 | .28223 | .14993 | .7289 | .8099 | .38721 | .8812 | 1.8824 | 1.6690 | .22166 |
| 35/64 | .546875 | .29907 | .16355 | .7395 | .8178 | .40442 | .8863 | 1.8286 | 1.7181 | .23489 |
| **9/16** | **.5625** | **.31641** | **.17798** | **.7500** | **.8255** | **.42188** | **.8913** | **1.7778** | **1.7671** | **.24850** |
| 37/64 | .578125 | .33423 | .19323 | .7604 | .8331 | .43957 | .8962 | 1.7297 | 1.8162 | .26250 |
| 19/32 | .59375 | .35254 | .20932 | .7706 | .8405 | .45751 | .9010 | 1.6842 | 1.8653 | .27688 |
| | .60 | .3600 | .21600 | .7746 | .8434 | .46476 | .9029 | 1.6667 | 1.8850 | .28274 |
| 39/64 | .609375 | .37134 | .22628 | .7806 | .8478 | .47569 | .9057 | 1.6410 | 1.9144 | .29165 |
| **5/8** | **.625** | **.39063** | **.24414** | **.7906** | **.8550** | **.49410** | **.9103** | **1.6000** | **1.9635** | **.30680** |
| 41/64 | .640625 | .41040 | .26291 | .8004 | .8621 | .51275 | .9148 | 1.5610 | 2.0126 | .32233 |
| 21/32 | .65625 | .43066 | .28262 | .8101 | .8690 | .53162 | .9192 | 1.5238 | 2.0617 | .33824 |
| 43/64 | .671875 | .45142 | .30330 | .8197 | .8759 | .55072 | .9235 | 1.4884 | 2.1108 | .35454 |
| **11/16** | **.6875** | **.47266** | **.32495** | **.8297** | **.8826** | **.57005** | **.9278** | **1.4545** | **2.1598** | **.37122** |
| | .70 | .4900 | .34300 | .8367 | .8879 | .58566 | .9312 | 1.4286 | 2.1991 | .38485 |
| 45/64 | .703125 | .49438 | .34761 | .8385 | .8892 | .58959 | .9320 | 1.4222 | 2.2089 | .38829 |
| 23/32 | .71875 | .51660 | .37131 | .8478 | .8958 | .60935 | .9361 | 1.3913 | 2.2580 | .40574 |
| 47/64 | .734375 | .53931 | .39605 | .8570 | .9022 | .62933 | .9401 | 1.3617 | 2.3071 | .42357 |
| **3/4** | **.750** | **.56250** | **.42188** | **.8660** | **.9086** | **.64952** | **.9441** | **1.3333** | **2.3562** | **.44179** |
| 49/64 | .765625 | .58618 | .44879 | .8750 | .9148 | .66992 | .9480 | 1.3061 | 2.4053 | .46038 |
| 25/32 | .78125 | .61035 | .47684 | .8839 | .9210 | .69053 | .9518 | 1.2800 | 2.4544 | .47937 |
| 51/64 | .796875 | .63501 | .50602 | .8927 | .9271 | .71135 | .9556 | 1.2549 | 2.5035 | .49874 |
| | .80 | .6400 | .51200 | .8944 | .9283 | .71554 | .9564 | 1.2500 | 2.5133 | .50265 |
| **13/16** | **.8125** | **.66016** | **.53638** | **.9014** | **.9331** | **.73238** | **.9593** | **1.2308** | **2.5525** | **.51849** |
| 53/64 | .828125 | .68579 | .56792 | .9100 | .9391 | .75361 | .9630 | 1.2075 | 2.6016 | .53862 |
| 27/32 | .84375 | .71191 | .60067 | .9186 | .9449 | .77503 | .9666 | 1.1852 | 2.6507 | .55914 |
| 55/64 | .859375 | .73853 | .63467 | .9270 | .9507 | .79666 | .9702 | 1.1636 | 2.6998 | .58004 |
| **7/8** | **.875** | **.76563** | **.66992** | **.9354** | **.9565** | **.81849** | **.9737** | **1.1429** | **2.7489** | **.60132** |
| 57/64 | .890625 | .79321 | .70645 | .9437 | .9621 | .84051 | .9771 | 1.1228 | 2.7980 | .62299 |
| | .90 | .81000 | .72900 | .9487 | .9655 | .85435 | .9792 | 1.1111 | 2.8274 | .63617 |
| 29/32 | .90625 | .82129 | .74429 | .9520 | .9677 | .86272 | .9805 | 1.1034 | 2.8471 | .64504 |
| 59/64 | .921875 | .84985 | .78346 | .9601 | .9733 | .88513 | .9839 | 1.0847 | 2.8962 | .66747 |
| **15/16** | **.9375** | **.87891** | **.82398** | **.9683** | **.9787** | **.90773** | **.9872** | **1.0667** | **2.9452** | **.69029** |
| 61/64 | .953125 | .90845 | .86587 | .9763 | .9841 | .93053 | .9905 | 1.0492 | 2.9943 | .71349 |
| 31/32 | .96875 | .93848 | .90915 | .9843 | .9895 | .95349 | .9937 | 1.0323 | 3.0434 | .73708 |
| 63/64 | .984375 | .96899 | .95385 | .9922 | .9948 | .97666 | .9969 | 1.0159 | 3.0925 | .76104 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **1.** | **1.0000** | **1.0000** | **1.0000** | **1.0000** | **1.0000** | **1.0000** | **1.0000000** | **3.1416** | **0.7854** |
| 1.125 | 1.2656 | 1.4238 | 1.0606 | 1.0400 | 1.1932 | 1.0238 | .8888888 | 3.5343 | .9940 |
| 1.25 | 1.5625 | 1.9531 | 1.1180 | 1.0772 | 1.3975 | 1.0456 | .80000000 | 3.9270 | 1.2272 |
| 1.375 | 1.8906 | 2.5996 | 1.1726 | 1.1120 | 1.6123 | 1.0658 | .72727272 | 4.3197 | 1.4849 |
| 1.5 | 2.25 | 3.3750 | 1.2247 | 1.1447 | 1.8371 | 1.0845 | .66666666 | 4.7124 | 1.7671 |
| 1.625 | 2.6406 | 4.2910 | 1.2748 | 1.1757 | 2.0715 | 1.1020 | .61538462 | 5.1051 | 2.0739 |
| 1.75 | 3.0625 | 5.3594 | 1.3229 | 1.2051 | 2.3150 | 1.1186 | .57142857 | 5.4978 | 2.4053 |
| 1.875 | 3.5156 | 6.5918 | 1.3693 | 1.2331 | 2.5675 | 1.1340 | .53333333 | 5.8905 | 2.7612 |
| **2.** | **4.0000** | **8.0000** | **1.4142** | **1.2599** | **2.8284** | **1.1487** | **.50000000** | **6.2832** | **3.1416** |
| 2.125 | 4.5156 | 9.5957 | 1.4577 | 1.2856 | 3.0977 | 1.1627 | .47058823 | 6.6759 | 3.5466 |
| 2.25 | 5.0625 | 11.3906 | 1.5000 | 1.3104 | 3.3750 | 1.1761 | .44444444 | 7.0686 | 3.9761 |
| 2.375 | 5.6406 | 13.3965 | 1.5411 | 1.3342 | 3.6601 | 1.1889 | .42105263 | 7.4613 | 4.4301 |
| 2.5 | 6.2500 | 15.6250 | 1.5811 | 1.3572 | 3.9529 | 1.2011 | .40000000 | 7.8540 | 4.9087 |
| 2.625 | 6.8906 | 18.0879 | 1.6202 | 1.3795 | 4.2530 | 1.2129 | .38095231 | 8.2467 | 5.4119 |
| 2.75 | 7.5625 | 20.7969 | 1.6583 | 1.4011 | 4.5604 | 1.2242 | .36363636 | 8.6394 | 5.9396 |
| 2.875 | 8.2656 | 23.7637 | 1.6956 | 1.4219 | 4.8748 | 1.2352 | .34782609 | 9.0321 | 6.4918 |
| **3.** | **9.0000** | **27.0000** | **1.7321** | **1.4422** | **5.1962** | **1.2457** | **.33333333** | **9.4248** | **7.0686** |
| 3.125 | 9.7656 | 30.5176 | 1.7678 | 1.4620 | 5.5243 | 1.2559 | .32000000 | 9.8175 | 7.6699 |
| 3.25 | 10.5625 | 34.3281 | 1.8028 | 1.4813 | 5.8590 | 1.2658 | .30769231 | 10.2102 | 8.2958 |
| 3.375 | 11.3906 | 38.4434 | 1.8371 | 1.5000 | 6.2003 | 1.2754 | .29629629 | 10.6029 | 8.9462 |
| 3.5 | 12.2500 | 42.8750 | 1.8708 | 1.5183 | 6.5479 | 1.2847 | .28571429 | 10.9956 | 9.6211 |
| 3.625 | 13.1406 | 47.6348 | 1.9039 | 1.5362 | 6.9018 | 1.2938 | .27586207 | 11.3883 | 10.3206 |
| 3.75 | 14.0625 | 52.7344 | 1.9365 | 1.5536 | 7.2619 | 1.3026 | .26666666 | 11.7810 | 11.0447 |
| 3.875 | 15.0156 | 58.1856 | 1.9685 | 1.5707 | 7.6279 | 1.3112 | .25806452 | 12.1737 | 11.7932 |
| **4.** | **16.0000** | **64.0000** | **2.0000** | **1.5874** | **8.0000** | **1.3195** | **.25000000** | **12.5664** | **12.5664** |
| 4.125 | 17.0156 | 70.1895 | 2.0310 | 1.6038 | 8.3779 | 1.3277 | .24242424 | 12.9591 | 13.3640 |
| 4.25 | 18.0625 | 76.7656 | 2.0616 | 1.6198 | 8.7616 | 1.3356 | .23529412 | 13.3518 | 14.1863 |
| 4.375 | 19.1406 | 83.7402 | 2.0916 | 1.6355 | 9.1510 | 1.3434 | .22857143 | 13.7445 | 15.0330 |
| 4.5 | 20.2500 | 91.1250 | 2.1213 | 1.6510 | 9.5460 | 1.3510 | .22222222 | 14.1372 | 15.9043 |
| 4.625 | 21.3906 | 98.9317 | 2.1506 | 1.6661 | 9.9465 | 1.3584 | .21621622 | 14.5299 | 16.8001 |
| 4.75 | 22.5625 | 107.1719 | 2.1795 | 1.6810 | 10.3524 | 1.3656 | .21052632 | 14.9226 | 17.7205 |
| 4.875 | 23.7656 | 115.8574 | 2.2079 | 1.6956 | 10.7637 | 1.3728 | .20512821 | 15.3153 | 18.6655 |
| **5.** | **25.0000** | **125.0000** | **2.2361** | **1.7100** | **11.1803** | **1.3799** | **.20000000** | **15.7080** | **19.6350** |
| 5.125 | 26.2656 | 134.6113 | 2.2638 | 1.7241 | 11.6022 | 1.3866 | .19512195 | 16.1006 | 20.6289 |
| 5.25 | 27.5625 | 144.7031 | 2.2913 | 1.7380 | 12.0293 | 1.3933 | .19047619 | 16.4933 | 21.6475 |
| 5.375 | 28.8906 | 155.2871 | 2.3184 | 1.7517 | 12.4614 | 1.3998 | .18604651 | 16.8860 | 22.6906 |
| 5.5 | 30.2500 | 166.3750 | 2.3452 | 1.7652 | 12.8987 | 1.4063 | .18181818 | 17.2787 | 23.7583 |
| 5.625 | 31.6406 | 177.9785 | 2.3727 | 1.7784 | 13.3409 | 1.4126 | .17777777 | 17.6714 | 24.8505 |
| 5.75 | 33.0625 | 190.1094 | 2.3979 | 1.7915 | 13.7880 | 1.4188 | .17391304 | 18.0641 | 25.9672 |
| 5.875 | 34.5156 | 202.7793 | 2.4238 | 1.8044 | 14.2400 | 1.4250 | .17021277 | 18.4568 | 27.1085 |
| **6.** | **36.0000** | **216.0000** | **2.4495** | **1.8171** | **14.6969** | **1.4310** | **.16666666** | **18.8495** | **28.2743** |
| 6.125 | 37.5156 | 229.7832 | 2.4749 | 1.8297 | 15.1586 | 1.4369 | .16326531 | 19.2422 | 29.4647 |
| 6.25 | 39.0625 | 244.1406 | 2.5000 | 1.8420 | 15.6250 | 1.4427 | .16000000 | 19.6349 | 30.6796 |
| 6.375 | 40.6406 | 259.0840 | 2.5249 | 1.8542 | 16.0961 | 1.4484 | .15686275 | 20.0276 | 31.9190 |
| 6.5 | 42.2500 | 274.6250 | 2.5495 | 1.8663 | 16.5718 | 1.4542 | .15384615 | 20.4203 | 33.1831 |
| 6.625 | 43.8906 | 290.7754 | 2.5739 | 1.8781 | 17.0522 | 1.4596 | .15094339 | 20.8130 | 34.4716 |
| 6.75 | 45.5625 | 307.5469 | 2.5981 | 1.8899 | 17.5370 | 1.4651 | .14814815 | 21.2057 | 35.7847 |
| 6.875 | 47.2656 | 324.9512 | 2.6220 | 1.9015 | 18.0264 | 1.4705 | .14545454 | 21.5984 | 37.1223 |
| **7.** | **49.0000** | **343.0000** | **2.6458** | **1.9129** | **18.5203** | **1.4758** | **.14285714** | **21.9911** | **38.4845** |
| 7.125 | 50.7656 | 361.7051 | 2.6693 | 1.9243 | 19.0186 | 1.4810 | .14035088 | 22.3838 | 39.8712 |
| 7.25 | 52.5625 | 381.0781 | 2.6926 | 1.9354 | 19.5212 | 1.4862 | .13793103 | 22.7765 | 41.2825 |
| 7.375 | 54.3906 | 401.1309 | 2.7157 | 1.9465 | 20.0283 | 1.4913 | .13559322 | 23.1692 | 42.7183 |
| 7.5 | 56.2500 | 421.8750 | 2.7386 | 1.9574 | 20.5396 | 1.4963 | .13333333 | 23.5619 | 44.1786 |
| 7.625 | 58.1406 | 443.3223 | 2.7613 | 1.9683 | 21.0552 | 1.5012 | .13114754 | 23.9546 | 45.6635 |
| 7.75 | 60.0625 | 465.4844 | 2.7839 | 1.9789 | 21.5751 | 1.5061 | .12903226 | 24.3473 | 47.1730 |
| 7.875 | 62.0156 | 488.3731 | 2.8063 | 1.9895 | 22.0992 | 1.5110 | .12698413 | 24.7400 | 48.7069 |
| **8.** | **64.0000** | **512.0000** | **2.8284** | **2.0000** | **22.6274** | **1.5157** | **.12500000** | **25.1327** | **50.2655** |
| 8.125 | 66.0156 | 536.3770 | 2.8504 | 2.0104 | 23.1598 | 1.5204 | .12307692 | 25.5254 | 51.8485 |
| 8.25 | 68.0625 | 561.5156 | 2.8723 | 2.0206 | 23.6963 | 1.5251 | .12121212 | 25.9181 | 53.4562 |
| 8.375 | 70.1406 | 587.4278 | 2.8940 | 2.0308 | 24.2369 | 1.5297 | .11940298 | 26.3108 | 55.0883 |
| 8.5 | 72.2500 | 614.1250 | 2.9155 | 2.0408 | 24.7816 | 1.5342 | .11764706 | 26.7035 | 56.7450 |
| 8.625 | 74.3906 | 641.6192 | 2.9368 | 2.0508 | 25.3301 | 1.5387 | .11594203 | 27.0962 | 58.4262 |
| 8.75 | 76.5625 | 669.9219 | 2.9580 | 2.0606 | 25.8828 | 1.5431 | .11428571 | 27.4889 | 60.1320 |
| 8.875 | 78.7656 | 699.0450 | 2.9791 | 2.0704 | 26.4394 | 1.5475 | .11267605 | 27.8816 | 61.8623 |
| **9.** | **81.0000** | **729.0000** | **3.0000** | **2.0801** | **27.0000** | **1.5518** | **.11111111** | **28.2743** | **63.6172** |
| 9.125 | 83.2656 | 759.7989 | 3.0207 | 2.0897 | 27.5645 | 1.5561 | .10958904 | 28.6670 | 65.3966 |
| 9.25 | 85.5625 | 791.4531 | 3.0414 | 2.0992 | 28.1328 | 1.5604 | .10810811 | 29.0597 | 67.2006 |
| 9.375 | 87.8906 | 823.9746 | 3.0619 | 2.1086 | 28.7050 | 1.5646 | .10666666 | 29.4524 | 69.0291 |
| 9.5 | 90.2500 | 857.3750 | 3.0822 | 2.1179 | 29.2810 | 1.5687 | .10526316 | 29.8451 | 70.8822 |
| 9.625 | 92.6406 | 891.6660 | 3.1024 | 2.1272 | 29.8608 | 1.5728 | .10389610 | 30.2378 | 72.7597 |
| 9.75 | 95.0625 | 926.8594 | 3.1225 | 2.1363 | 30.4444 | 1.5769 | .10256410 | 30.6305 | 74.6619 |
| 9.875 | 97.5156 | 962.9668 | 3.1425 | 2.1454 | 31.0317 | 1.5809 | .10126582 | 31.0232 | 76.5886 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **10** | **100** | **1000** | **3.1623** | **2.1544** | **31.623** | **1.5849** | **.10000000** | **31.4159** | **78.5398** |
| 11 | 121 | 1331 | 3.3166 | 2.2240 | 36.483 | 1.6154 | .09090909 | 34.5575 | 95.0332 |
| 12 | 144 | 1728 | 3.4641 | 2.2894 | 41.569 | 1.6438 | .08333333 | 37.6991 | 113.0973 |
| 13 | 169 | 2197 | 3.6056 | 2.3513 | 46.873 | 1.6703 | .07692308 | 40.8407 | 132.7323 |
| 14 | 196 | 2744 | 3.7417 | 2.4101 | 52.384 | 1.6953 | .07142857 | 43.9823 | 153.9380 |
| 15 | 225 | 3375 | 3.8730 | 2.4662 | 58.095 | 1.7188 | .06666667 | 47.1239 | 176.7146 |
| 16 | 256 | 4096 | 4.0000 | 2.5198 | 64.000 | 1.7411 | .06250000 | 50.2654 | 201.0619 |
| 17 | 289 | 4913 | 4.1231 | 2.5713 | 70.093 | 1.7623 | .05882353 | 53.4070 | 226.9801 |
| 18 | 324 | 5832 | 4.2426 | 2.6207 | 76.367 | 1.7826 | .05555556 | 56.5486 | 254.4690 |
| 19 | 361 | 6859 | 4.3589 | 2.6684 | 82.819 | 1.8020 | .05263158 | 59.6902 | 283.5287 |
| **20** | **400** | **8000** | **4.4721** | **2.7144** | **89.442** | **1.8206** | **.05000000** | **62.8318** | **314.1593** |
| 21 | 441 | 9261 | 4.5826 | 2.7589 | 96.235 | 1.8384 | .04761905 | 65.9734 | 346.3606 |
| 22 | 484 | 10648 | 4.6904 | 2.8020 | 103.19 | 1.8556 | .04545455 | 69.1150 | 380.1327 |
| 23 | 529 | 12167 | 4.7958 | 2.8439 | 110.30 | 1.8722 | .04347826 | 72.2566 | 415.4756 |
| 24 | 576 | 13824 | 4.8990 | 2.8845 | 117.58 | 1.8882 | .04166667 | 75.3982 | 452.3893 |
| 25 | 625 | 15625 | 5.0000 | 2.9240 | 125.00 | 1.9037 | .04000000 | 78.5398 | 490.8739 |
| 26 | 676 | 17576 | 5.0990 | 2.9625 | 132.57 | 1.9186 | .03846154 | 81.6813 | 530.9292 |
| 27 | 729 | 19683 | 5.1962 | 3.0000 | 140.30 | 1.9332 | .03703704 | 84.8229 | 572.5553 |
| 28 | 784 | 21952 | 5.2915 | 3.0366 | 148.16 | 1.9473 | .03571429 | 87.9645 | 615.7522 |
| 29 | 841 | 24389 | 5.3852 | 3.0723 | 156.17 | 1.9610 | .03448276 | 91.1061 | 660.5198 |
| **30** | **900** | **27000** | **5.4772** | **3.1072** | **164.32** | **1.9744** | **.03333333** | **94.2477** | **706.8583** |
| 31 | 961 | 29791 | 5.5678 | 3.1414 | 172.60 | 1.9873 | .03225806 | 97.3893 | 754.7676 |
| 32 | 1024 | 32768 | 5.6569 | 3.1748 | 181.02 | 2.0000 | .03125000 | 100.5309 | 804.2477 |
| 33 | 1089 | 35937 | 5.7446 | 3.2075 | 189.57 | 2.0123 | .03030303 | 103.6725 | 855.2986 |
| 34 | 1156 | 39304 | 5.8310 | 3.2396 | 198.25 | 2.0244 | .02941176 | 106.8141 | 907.9203 |
| 35 | 1225 | 42875 | 5.9161 | 3.2711 | 207.06 | 2.0362 | .02857143 | 109.9557 | 962.1127 |
| 36 | 1296 | 46656 | 6.0000 | 3.3019 | 216.00 | 2.0477 | .02777778 | 113.0972 | 1017.8760 |
| 37 | 1369 | 50653 | 6.0828 | 3.3322 | 225.06 | 2.0589 | .02702703 | 116.2388 | 1075.2101 |
| 38 | 1444 | 54872 | 6.1644 | 3.3620 | 234.25 | 2.0699 | .02631579 | 119.3804 | 1134.1149 |
| 39 | 1521 | 59319 | 6.2450 | 3.3912 | 243.56 | 2.0807 | .02564103 | 122.5220 | 1194.5906 |
| **40** | **1600** | **64000** | **6.3246** | **3.4200** | **252.98** | **2.0913** | **.02500000** | **125.6636** | **1256.6371** |
| 41 | 1681 | 68921 | 6.4031 | 3.4482 | 262.53 | 2.1016 | .02439024 | 128.8052 | 1320.2543 |
| 42 | 1764 | 74088 | 6.4807 | 3.4760 | 272.19 | 2.1118 | .02380952 | 131.9468 | 1385.4424 |
| 43 | 1849 | 79507 | 6.5574 | 3.5034 | 281.97 | 2.1218 | .02325581 | 135.0884 | 1452.2012 |
| 44 | 1936 | 85184 | 6.6332 | 3.5303 | 291.86 | 2.1315 | .02272727 | 138.2300 | 1520.5308 |
| 45 | 2025 | 91125 | 6.7082 | 3.5569 | 301.87 | 2.1411 | .02222222 | 141.3716 | 1590.4313 |
| 46 | 2116 | 97336 | 6.7823 | 3.5830 | 311.99 | 2.1506 | .02173913 | 144.5131 | 1661.9025 |
| 47 | 2209 | 103823 | 6.8557 | 3.6088 | 322.22 | 2.1598 | .02127660 | 147.6547 | 1734.9445 |
| 48 | 2304 | 110592 | 6.9282 | 3.6342 | 332.55 | 2.1689 | .02083333 | 150.7963 | 1809.5574 |
| 49 | 2401 | 117649 | 7.0000 | 3.6593 | 343.00 | 2.1779 | .02040816 | 153.9379 | 1885.7410 |
| **50** | **2500** | **125000** | **7.0711** | **3.6840** | **353.55** | **2.1867** | **.02000000** | **157.0795** | **1963.500** |
| 51 | 2601 | 132651 | 7.1414 | 3.7084 | 364.21 | 2.1954 | .01960784 | 160.2211 | 2042.820 |
| 52 | 2704 | 140608 | 7.2111 | 3.7325 | 374.98 | 2.2039 | .01923077 | 163.3627 | 2123.716 |
| 53 | 2809 | 148877 | 7.2801 | 3.7563 | 385.85 | 2.2124 | .01886792 | 166.5043 | 2206.183 |
| 54 | 2916 | 157464 | 7.3485 | 3.7798 | 396.82 | 2.2206 | .01851852 | 169.6459 | 2290.221 |
| 55 | 3025 | 166375 | 7.4162 | 3.8030 | 407.89 | 2.2288 | .01818182 | 172.7875 | 2375.829 |
| 56 | 3136 | 175616 | 7.4833 | 3.8259 | 419.07 | 2.2369 | .01785714 | 175.9290 | 2463.008 |
| 57 | 3249 | 185193 | 7.5498 | 3.8485 | 430.35 | 2.2448 | .01754386 | 179.0706 | 2551.758 |
| 58 | 3364 | 195112 | 7.6158 | 3.8709 | 441.72 | 2.2526 | .01724138 | 182.2122 | 2642.079 |
| 59 | 3481 | 205379 | 7.6811 | 3.8930 | 453.19 | 2.2603 | .01694915 | 185.3538 | 2733.970 |
| **60** | **3600** | **216000** | **7.7460** | **3.9149** | **464.76** | **2.2679** | **.01666667** | **188.4954** | **2827.433** |
| 61 | 3721 | 226981 | 7.8102 | 3.9365 | 476.43 | 2.2755 | .01639344 | 191.6370 | 2922.466 |
| 62 | 3844 | 238328 | 7.8740 | 3.9579 | 488.19 | 2.2829 | .01612903 | 194.7786 | 3019.070 |
| 63 | 3969 | 250047 | 7.9373 | 3.9791 | 500.05 | 2.2902 | .01587302 | 197.9202 | 3117.245 |
| 64 | 4096 | 262144 | 8.0000 | 4.0000 | 512.00 | 2.2974 | .01562500 | 201.0618 | 3216.990 |
| 65 | 4225 | 274625 | 8.0623 | 4.0207 | 524.05 | 2.3045 | .01538462 | 204.2034 | 3318.307 |
| 66 | 4356 | 287496 | 8.1240 | 4.0412 | 536.19 | 2.3116 | .01515152 | 207.3449 | 3421.194 |
| 67 | 4489 | 300763 | 8.1854 | 4.0615 | 548.42 | 2.3186 | .01492537 | 210.4865 | 3525.652 |
| 68 | 4624 | 314432 | 8.2462 | 4.0817 | 560.74 | 2.3254 | .01470588 | 213.6281 | 3631.680 |
| 69 | 4761 | 328509 | 8.3066 | 4.1016 | 573.16 | 2.3322 | .01449275 | 216.7697 | 3739.280 |
| **70** | **4900** | **343000** | **8.3666** | **4.1213** | **585.66** | **2.3389** | **.01428571** | **219.9113** | **3848.450** |
| 71 | 5041 | 357911 | 8.4261 | 4.1408 | 598.26 | 2.3456 | .01408451 | 223.0529 | 3959.191 |
| 72 | 5184 | 373248 | 8.4853 | 4.1602 | 610.94 | 2.3522 | .01388889 | 226.1945 | 4071.503 |
| 73 | 5329 | 389017 | 8.5440 | 4.1793 | 623.71 | 2.3587 | .01369863 | 229.3361 | 4185.386 |
| 74 | 5476 | 405224 | 8.6023 | 4.1983 | 636.57 | 2.3651 | .01351351 | 232.4777 | 4300.839 |
| 75 | 5625 | 421875 | 8.6603 | 4.2172 | 649.52 | 2.3714 | .01333333 | 235.6193 | 4417.864 |
| 76 | 5776 | 438976 | 8.7178 | 4.2358 | 662.55 | 2.3777 | .01315789 | 238.7608 | 4536.459 |
| 77 | 5929 | 456533 | 8.7750 | 4.2543 | 675.68 | 2.3840 | .01298701 | 241.9024 | 4656.625 |
| 78 | 6084 | 474552 | 8.8318 | 4.2727 | 688.88 | 2.3901 | .01282051 | 245.0440 | 4778.361 |
| 79 | 6241 | 493039 | 8.8882 | 4.2908 | 702.17 | 2.3962 | .01265823 | 248.1856 | 4901.669 |

**Table 4.—Properties of Numbers**—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **80** | **6400** | **512000** | **8.9443** | **4.3089** | **715.54** | **2.4022** | **.01250000** | **251.327** | **5026.547** |
| 81 | 6561 | 531441 | 9.0000 | 4.3267 | 729.00 | 2.4082 | .01234568 | 254.469 | 5152.998 |
| 82 | 6724 | 551368 | 9.0554 | 4.3445 | 742.54 | 2.4141 | .01219512 | 257.610 | 5281.016 |
| 83 | 6889 | 571787 | 9.1104 | 4.3621 | 756.17 | 2.4200 | .01204819 | 260.752 | 5410.607 |
| 84 | 7056 | 592704 | 9.1652 | 4.3795 | 769.88 | 2.4258 | .01190476 | 263.894 | 5541.770 |
| 85 | 7225 | 614125 | 9.2195 | 4.3968 | 783.66 | 2.4315 | .01176471 | 267.035 | 5674.501 |
| 86 | 7396 | 636056 | 9.2736 | 4.4140 | 797.53 | 2.4372 | .01162791 | 270.177 | 5808.805 |
| 87 | 7569 | 658503 | 9.3274 | 4.4310 | 811.49 | 2.4429 | .01149425 | 273.318 | 5944.679 |
| 88 | 7744 | 681472 | 9.3808 | 4.4480 | 825.52 | 2.4485 | .01136364 | 276.460 | 6082.124 |
| 89 | 7921 | 704969 | 9.4340 | 4.4647 | 839.63 | 2.4540 | .01123596 | 279.602 | 6221.138 |
| **90** | **8100** | **729000** | **9.4868** | **4.4814** | **853.82** | **2.4595** | **.01111111** | **282.743** | **6361.725** |
| 91 | 8281 | 753571 | 9.5394 | 4.4979 | 868.09 | 2.4650 | .01098901 | 285.885 | 6503.882 |
| 92 | 8464 | 778688 | 9.5917 | 4.5144 | 882.44 | 2.4705 | .01086957 | 289.026 | 6647.610 |
| 93 | 8649 | 804357 | 9.6437 | 4.5307 | 896.86 | 2.4758 | .01075269 | 292.168 | 6792.909 |
| 94 | 8836 | 830584 | 9.6954 | 4.5468 | 911.36 | 2.4810 | .01063830 | 295.309 | 6939.778 |
| 95 | 9025 | 857375 | 9.7468 | 4.5629 | 925.95 | 2.4863 | .01052632 | 298.451 | 7088.219 |
| 96 | 9216 | 884736 | 9.7980 | 4.5789 | 940.61 | 2.4915 | .01041667 | 301.593 | 7238.230 |
| 97 | 9409 | 912673 | 9.8489 | 4.5947 | 955.34 | 2.4966 | .01030928 | 304.734 | 7389.812 |
| 98 | 9604 | 941192 | 9.8995 | 4.6104 | 970.15 | 2.5018 | .01020408 | 307.876 | 7542.962 |
| 99 | 9801 | 970299 | 9.9499 | 4.6261 | 985.04 | 2.5069 | .01010101 | 311.017 | 7697.688 |
| **100** | **10000** | **1000000** | **10.0000** | **4.6416** | **1000.0** | **2.5119** | **.01000000** | **314.159** | **7853.982** |
| 101 | 10201 | 1030301 | 10.0499 | 4.6570 | 1015.0 | 2.5169 | .00990099 | 317.301 | 8011.85 |
| 102 | 10404 | 1061208 | 10.0995 | 4.6723 | 1030.1 | 2.5219 | .00980392 | 320.442 | 8171.28 |
| 103 | 10609 | 1092727 | 10.1489 | 4.6875 | 1045.3 | 2.5268 | .00970874 | 323.584 | 8332.29 |
| 104 | 10816 | 1124864 | 10.1980 | 4.7027 | 1060.6 | 2.5317 | .00961538 | 326.725 | 8494.87 |
| 105 | 11025 | 1157625 | 10.2470 | 4.7177 | 1075.9 | 2.5365 | .00952381 | 329.867 | 8659.01 |
| 106 | 11236 | 1191016 | 10.2956 | 4.7326 | 1091.3 | 2.5413 | .00943396 | 333.009 | 8824.73 |
| 107 | 11449 | 1225043 | 10.3441 | 4.7475 | 1106.8 | 2.5461 | .00934579 | 336.150 | 8992.02 |
| 108 | 11664 | 1259712 | 10.3923 | 4.7622 | 1122.4 | 2.5509 | .00925926 | 339.292 | 9160.88 |
| 109 | 11881 | 1295029 | 10.4403 | 4.7769 | 1138.0 | 2.5556 | .00917431 | 342.433 | 9331.32 |
| **110** | **12100** | **1331000** | **10.4881** | **4.7914** | **1153.7** | **2.5603** | **.00909091** | **345.575** | **9503.32** |
| 111 | 12321 | 1367631 | 10.5357 | 4.8059 | 1169.5 | 2.5649 | .00900901 | 348.716 | 9676.89 |
| 112 | 12544 | 1404928 | 10.5830 | 4.8203 | 1185.3 | 2.5695 | .00892857 | 351.858 | 9852.03 |
| 113 | 12769 | 1442897 | 10.6301 | 4.8346 | 1201.2 | 2.5740 | .00884956 | 355.000 | 10028.75 |
| 114 | 12996 | 1481544 | 10.6771 | 4.8488 | 1217.2 | 2.5786 | .00877193 | 358.141 | 10207.03 |
| 115 | 13225 | 1520875 | 10.7238 | 4.8629 | 1233.2 | 2.5831 | .00869565 | 361.283 | 10386.89 |
| 116 | 13456 | 1560896 | 10.7703 | 4.8770 | 1249.4 | 2.5876 | .00862069 | 364.424 | 10568.32 |
| 117 | 13689 | 1601613 | 10.8167 | 4.8910 | 1265.5 | 2.5920 | .00854701 | 367.566 | 10751.31 |
| 118 | 13924 | 1643032 | 10.8628 | 4.9049 | 1281.8 | 2.5964 | .00847458 | 370.708 | 10935.88 |
| 119 | 14161 | 1685159 | 10.9087 | 4.9187 | 1298.1 | 2.6008 | .00840336 | 373.849 | 11122.02 |
| **120** | **14400** | **1728000** | **10.9545** | **4.9324** | **1314.5** | **2.6052** | **.00833333** | **376.991** | **11309.73** |
| 121 | 14641 | 1771561 | 11.0000 | 4.9461 | 1331.0 | 2.6095 | .00826446 | 380.132 | 11499.01 |
| 122 | 14884 | 1815848 | 11.0454 | 4.9597 | 1347.5 | 2.6138 | .00819672 | 383.274 | 11689.86 |
| 123 | 15129 | 1860867 | 11.0905 | 4.9732 | 1364.1 | 2.6181 | .00813008 | 386.416 | 11882.29 |
| 124 | 15376 | 1906624 | 11.1355 | 4.9866 | 1380.8 | 2.6223 | .00806452 | 389.557 | 12076.28 |
| 125 | 15625 | 1953125 | 11.1803 | 5.0000 | 1397.5 | 2.6265 | .00800000 | 392.699 | 12271.84 |
| 126 | 15876 | 2000376 | 11.2250 | 5.0133 | 1414.4 | 2.6307 | .00793651 | 395.840 | 12468.98 |
| 127 | 16129 | 2048383 | 11.2694 | 5.0265 | 1431.2 | 2.6349 | .00787402 | 398.982 | 12667.68 |
| 128 | 16384 | 2097152 | 11.3137 | 5.0397 | 1448.2 | 2.6390 | .00781250 | 402.124 | 12867.96 |
| 129 | 16641 | 2146689 | 11.3578 | 5.0528 | 1465.2 | 2.6431 | .00775194 | 405.265 | 13069.81 |
| **130** | **16900** | **2197000** | **11.4018** | **5.0658** | **1482.2** | **2.6472** | **.00769231** | **408.407** | **13273.23** |
| 131 | 17161 | 2248091 | 11.4455 | 5.0788 | 1499.4 | 2.6513 | .00763359 | 411.548 | 13478.22 |
| 132 | 17424 | 2299968 | 11.4891 | 5.0916 | 1516.6 | 2.6553 | .00757576 | 414.690 | 13684.77 |
| 133 | 17689 | 2352637 | 11.5326 | 5.1045 | 1533.8 | 2.6593 | .00751880 | 417.831 | 13892.91 |
| 134 | 17956 | 2406104 | 11.5758 | 5.1172 | 1551.2 | 2.6633 | .00746269 | 420.973 | 14102.61 |
| 135 | 18225 | 2460375 | 11.6190 | 5.1299 | 1568.6 | 2.6673 | .00740741 | 424.115 | 14313.88 |
| 136 | 18496 | 2515456 | 11.6619 | 5.1426 | 1586.0 | 2.6712 | .00735294 | 427.256 | 14526.72 |
| 137 | 18769 | 2571353 | 11.7047 | 5.1551 | 1603.6 | 2.6751 | .00729927 | 430.398 | 14741.14 |
| 138 | 19044 | 2628072 | 11.7473 | 5.1676 | 1621.1 | 2.6790 | .00724638 | 433.539 | 14957.12 |
| 139 | 19321 | 2685619 | 11.7898 | 5.1801 | 1638.8 | 2.6829 | .00719424 | 436.681 | 15174.67 |
| **140** | **19600** | **2744000** | **11.8322** | **5.1925** | **1656.5** | **2.6867** | **.00714286** | **439.823** | **15393.80** |
| 141 | 19881 | 2803221 | 11.8743 | 5.2048 | 1674.3 | 2.6906 | .00709220 | 442.964 | 15614.50 |
| 142 | 20164 | 2863288 | 11.9164 | 5.2171 | 1692.1 | 2.6944 | .00704225 | 446.106 | 15836.77 |
| 143 | 20449 | 2924207 | 11.9583 | 5.2293 | 1710.0 | 2.6981 | .00699301 | 449.247 | 16060.60 |
| 144 | 20736 | 2985984 | 12.0000 | 5.2415 | 1728.0 | 2.7019 | .00694444 | 452.389 | 16286.01 |
| 145 | 21025 | 3048625 | 12.0416 | 5.2536 | 1746.0 | 2.7057 | .00689655 | 455.531 | 16512.99 |
| 146 | 21316 | 3112136 | 12.0830 | 5.2656 | 1764.1 | 2.7094 | .00684932 | 458.672 | 16741.54 |
| 147 | 21609 | 3176523 | 12.1244 | 5.2776 | 1782.2 | 2.7131 | .00680272 | 461.814 | 16971.67 |
| 148 | 21904 | 3241792 | 12.1655 | 5.2896 | 1800.5 | 2.7168 | .00675676 | 464.955 | 17203.36 |
| 149 | 22201 | 3307949 | 12.2066 | 5.3015 | 1818.8 | 2.7204 | .00671141 | 468.097 | 17436.62 |

**Table 4.—Properties of Numbers—*Continued***

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **150** | **22500** | **3375000** | **12.2474** | **5.3133** | **1837.1** | **2.7241** | **.00666667** | **471.239** | **17671.46** |
| 151 | 22801 | 3442951 | 12.2882 | 5.3251 | 1855.5 | 2.7277 | .00662252 | 474.380 | 17907.86 |
| 152 | 23104 | 3511808 | 12.3288 | 5.3368 | 1874.0 | 2.7314 | .00657895 | 477.522 | 18145.84 |
| 153 | 23409 | 3581577 | 12.3693 | 5.3485 | 1892.5 | 2.7349 | .00653595 | 480.663 | 18385.38 |
| 154 | 23716 | 3652264 | 12.4097 | 5.3601 | 1911.1 | 2.7385 | .00649351 | 483.805 | 18626.50 |
| 155 | 24025 | 3723875 | 12.4499 | 5.3717 | 1929.7 | 2.7420 | .00645161 | 486.946 | 18869.19 |
| 156 | 24336 | 3796416 | 12.4900 | 5.3832 | 1948.4 | 2.7455 | .00641026 | 490.088 | 19113.45 |
| 157 | 24649 | 3869893 | 12.5300 | 5.3947 | 1967.2 | 2.7490 | .00636943 | 493.230 | 19359.28 |
| 158 | 24964 | 3944312 | 12.5698 | 5.4061 | 1986.0 | 2.7525 | .00632911 | 496.371 | 19606.68 |
| 159 | 25281 | 4019679 | 12.6095 | 5.4175 | 2004.9 | 2.7560 | .00628931 | 499.513 | 19855.65 |
| **160** | **25600** | **4096000** | **12.6491** | **5.4288** | **2023.9** | **2.7595** | **.00625000** | **502.654** | **20106.19** |
| 161 | 25921 | 4173281 | 12.6886 | 5.4401 | 2042.9 | 2.7629 | .00621118 | 505.796 | 20358.30 |
| 162 | 26244 | 4251528 | 12.7279 | 5.4514 | 2061.9 | 2.7663 | .00617284 | 508.938 | 20611.99 |
| 163 | 26569 | 4330747 | 12.7671 | 5.4626 | 2081.0 | 2.7697 | .00613497 | 512.079 | 20867.24 |
| 164 | 26896 | 4410944 | 12.8062 | 5.4737 | 2100.2 | 2.7731 | .00609756 | 515.221 | 21124.06 |
| 165 | 27225 | 4492125 | 12.8452 | 5.4848 | 2119.5 | 2.7765 | .00606061 | 518.362 | 21382.46 |
| 166 | 27556 | 4574296 | 12.8841 | 5.4959 | 2138.8 | 2.7799 | .00602410 | 521.504 | 21642.43 |
| 167 | 27889 | 4657463 | 12.9228 | 5.5069 | 2158.1 | 2.7832 | .00598802 | 524.646 | 21903.96 |
| 168 | 28224 | 4741632 | 12.9615 | 5.5178 | 2177.5 | 2.7865 | .00595238 | 527.787 | 22167.07 |
| 169 | 28561 | 4826809 | 13.0000 | 5.5288 | 2197.0 | 2.7898 | .00591716 | 530.929 | 22431.75 |
| **170** | **28900** | **4913000** | **13.0384** | **5.5397** | **2216.5** | **2.7931** | **.00588235** | **534.070** | **22698.00** |
| 171 | 29241 | 5000211 | 13.0767 | 5.5505 | 2236.1 | 2.7964 | .00584795 | 537.212 | 22965.82 |
| 172 | 29584 | 5088448 | 13.1149 | 5.5613 | 2255.8 | 2.7997 | .00581395 | 540.353 | 23235.21 |
| 173 | 29929 | 5177717 | 13.1529 | 5.5721 | 2275.5 | 2.8029 | .00578035 | 543.495 | 23506.18 |
| 174 | 30276 | 5268024 | 13.1909 | 5.5828 | 2295.2 | 2.8061 | .00574713 | 546.637 | 23778.71 |
| 175 | 30625 | 5359375 | 13.2288 | 5.5934 | 2315.0 | 2.8094 | .00571429 | 549.778 | 24052.81 |
| 176 | 30976 | 5451776 | 13.2665 | 5.6041 | 2334.9 | 2.8126 | .00568182 | 552.920 | 24328.49 |
| 177 | 31329 | 5545233 | 13.3041 | 5.6147 | 2354.8 | 2.8158 | .00564972 | 556.061 | 24605.73 |
| 178 | 31684 | 5639752 | 13.3417 | 5.6252 | 2374.8 | 2.8189 | .00561798 | 559.203 | 24884.55 |
| 179 | 32041 | 5735339 | 13.3791 | 5.6357 | 2394.9 | 2.8221 | .00558659 | 562.345 | 25164.94 |
| **180** | **32400** | **5832000** | **13.4164** | **5.6462** | **2415.0** | **2.8252** | **.00555556** | **565.486** | **25446.90** |
| 181 | 32761 | 5929741 | 13.4536 | 5.6567 | 2435.1 | 2.8284 | .00552486 | 568.628 | 25730.42 |
| 182 | 33124 | 6028568 | 13.4907 | 5.6671 | 2455.3 | 2.8315 | .00549451 | 571.769 | 26015.52 |
| 183 | 33489 | 6128487 | 13.5277 | 5.6774 | 2475.6 | 2.8346 | .00546448 | 574.911 | 26302.19 |
| 184 | 33856 | 6229504 | 13.5647 | 5.6877 | 2495.9 | 2.8377 | .00543478 | 578.053 | 26590.43 |
| 185 | 34225 | 6331625 | 13.6015 | 5.6980 | 2516.3 | 2.8408 | .00540541 | 581.194 | 26880.25 |
| 186 | 34596 | 6434856 | 13.6382 | 5.7083 | 2536.7 | 2.8438 | .00537634 | 584.336 | 27171.63 |
| 187 | 34969 | 6539203 | 13.6748 | 5.7185 | 2557.2 | 2.8469 | .00534759 | 587.477 | 27464.58 |
| 188 | 35344 | 6644672 | 13.7113 | 5.7287 | 2577.7 | 2.8499 | .00531915 | 590.619 | 27759.11 |
| 189 | 35721 | 6751269 | 13.7477 | 5.7388 | 2598.3 | 2.8529 | .00529101 | 593.761 | 28055.20 |
| **190** | **36100** | **6859000** | **13.7840** | **5.7489** | **2619.0** | **2.8560** | **.00526316** | **596.902** | **28352.87** |
| 191 | 36481 | 6967871 | 13.8203 | 5.7590 | 2639.7 | 2.8590 | .00523560 | 600.044 | 28652.10 |
| 192 | 36864 | 7077888 | 13.8564 | 5.7690 | 2660.4 | 2.8619 | .00520833 | 603.185 | 28952.91 |
| 193 | 37249 | 7189057 | 13.8924 | 5.7790 | 2681.2 | 2.8649 | .00518135 | 606.327 | 29255.29 |
| 194 | 37636 | 7301384 | 13.9284 | 5.7890 | 2702.1 | 2.8679 | .00515464 | 609.468 | 29559.24 |
| 195 | 38025 | 7414875 | 13.9642 | 5.7989 | 2723.0 | 2.8708 | .00512821 | 612.610 | 29864.76 |
| 196 | 38416 | 7529536 | 14.0000 | 5.8088 | 2744.0 | 2.8738 | .00510204 | 615.752 | 30171.85 |
| 197 | 38809 | 7645373 | 14.0357 | 5.8186 | 2765.0 | 2.8767 | .00507614 | 618.893 | 30480.51 |
| 198 | 39204 | 7762392 | 14.0712 | 5.8285 | 2786.1 | 2.8796 | .00505051 | 622.035 | 30790.74 |
| 199 | 39601 | 7880599 | 14.1067 | 5.8383 | 2807.2 | 2.8825 | .00502513 | 625.176 | 31102.55 |
| **200** | **40000** | **8000000** | **14.1421** | **5.8480** | **2828.4** | **2.8854** | **.00500000** | **628.318** | **31415.93** |
| 201 | 40401 | 8120601 | 14.1774 | 5.8578 | 2849.7 | 2.8883 | .00497512 | 631.460 | 31730.87 |
| 202 | 40804 | 8242408 | 14.2127 | 5.8675 | 2871.0 | 2.8911 | .00495050 | 634.601 | 32047.39 |
| 203 | 41209 | 8365427 | 14.2478 | 5.8771 | 2892.3 | 2.8940 | .00492611 | 637.743 | 32365.47 |
| 204 | 41616 | 8489664 | 14.2829 | 5.8868 | 2913.7 | 2.8968 | .00490196 | 640.884 | 32685.13 |
| 205 | 42025 | 8615125 | 14.3178 | 5.8964 | 2935.2 | 2.8997 | .00487805 | 644.026 | 33006.36 |
| 206 | 42436 | 8741816 | 14.3527 | 5.9059 | 2956.7 | 2.9025 | .00485437 | 647.168 | 33329.16 |
| 207 | 42849 | 8869743 | 14.3875 | 5.9155 | 2978.2 | 2.9053 | .00483092 | 650.309 | 33653.53 |
| 208 | 43264 | 8998912 | 14.4222 | 5.9250 | 2999.8 | 2.9081 | .00480769 | 653.451 | 33979.47 |
| 209 | 43681 | 9129329 | 14.4568 | 5.9345 | 3021.5 | 2.9109 | .00478469 | 656.592 | 34306.98 |
| **210** | **44100** | **9261000** | **14.4914** | **5.9439** | **3043.2** | **2.9137** | **.00476190** | **659.734** | **34636.06** |
| 211 | 44521 | 9393931 | 14.5258 | 5.9533 | 3065.0 | 2.9165 | .00473934 | 662.875 | 34966.71 |
| 212 | 44944 | 9528128 | 14.5602 | 5.9627 | 3086.8 | 2.9192 | .00471698 | 666.017 | 35298.94 |
| 213 | 45369 | 9663597 | 14.5945 | 5.9721 | 3108.7 | 2.9220 | .00469484 | 669.159 | 35632.73 |
| 214 | 45796 | 9800344 | 14.6287 | 5.9814 | 3130.6 | 2.9247 | .00467290 | 672.300 | 35968.09 |
| 215 | 46225 | 9938375 | 14.6629 | 5.9907 | 3152.5 | 2.9274 | .00465116 | 675.442 | 36305.03 |
| 216 | 46656 | 10077696 | 14.6969 | 6.0000 | 3174.5 | 2.9302 | .00462963 | 678.583 | 36643.54 |
| 217 | 47089 | 10218313 | 14.7309 | 6.0092 | 3196.6 | 2.9329 | .00460829 | 681.725 | 36983.61 |
| 218 | 47524 | 10360232 | 14.7648 | 6.0185 | 3218.7 | 2.9356 | .00458716 | 684.867 | 37325.26 |
| 219 | 47961 | 10503459 | 14.7986 | 6.0277 | 3240.9 | 2.9383 | .00456621 | 688.008 | 37668.48 |

**Table 4.—Properties of Numbers—*Continued***

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **220** | **48400** | **10648000** | **14.8324** | **6.0368** | **3263.1** | **2.9409** | **.00454545** | **691.150** | **38013.27** |
| 221 | 48841 | 10793861 | 14.8661 | 6.0459 | 3285.4 | 2.9436 | .00452489 | 694.291 | 38359.63 |
| 222 | 49284 | 10941048 | 14.8997 | 6.0550 | 3307.7 | 2.9463 | .00450450 | 697.433 | 38707.56 |
| 223 | 49729 | 11089567 | 14.9332 | 6.0641 | 3330.1 | 2.9489 | .00448430 | 700.575 | 39057.07 |
| 224 | 50176 | 11239424 | 14.9666 | 6.0732 | 3352.5 | 2.9516 | .00446429 | 703.716 | 39408.14 |
| 225 | 50625 | 11390625 | 15.0000 | 6.0822 | 3375.0 | 2.9542 | .00444444 | 706.858 | 39760.78 |
| 226 | 51076 | 11543176 | 15.0333 | 6.0912 | 3397.5 | 2.9568 | .00442478 | 709.999 | 40115.00 |
| 227 | 51529 | 11697083 | 15.0665 | 6.1002 | 3420.1 | 2.9594 | .00440529 | 713.141 | 40470.78 |
| 228 | 51984 | 11852352 | 15.0997 | 6.1091 | 3442.7 | 2.9620 | .00438596 | 716.283 | 40828.14 |
| 229 | 52441 | 12008989 | 15.1327 | 6.1180 | 3465.4 | 2.9646 | .00436681 | 719.424 | 41187.07 |
| **230** | **52900** | **12167000** | **15.1658** | **6.1269** | **3488.1** | **2.9672** | **.00434783** | **722.566** | **41547.56** |
| 231 | 53361 | 12326391 | 15.1987 | 6.1358 | 3510.9 | 2.9698 | .00432900 | 725.707 | 41909.63 |
| 232 | 53824 | 12487168 | 15.2315 | 6.1446 | 3533.7 | 2.9723 | .00431034 | 728.849 | 42273.27 |
| 233 | 54289 | 12649337 | 15.2643 | 6.1534 | 3556.6 | 2.9749 | .00429185 | 731.990 | 42638.48 |
| 234 | 54756 | 12812904 | 15.2971 | 6.1622 | 3579.5 | 2.9774 | .00427350 | 735.132 | 43005.26 |
| 235 | 55225 | 12977875 | 15.3297 | 6.1710 | 3602.5 | 2.9800 | .00425532 | 738.274 | 43373.61 |
| 236 | 55696 | 13144256 | 15.3623 | 6.1797 | 3625.5 | 2.9825 | .00423729 | 741.415 | 43743.54 |
| 237 | 56169 | 13312053 | 15.3948 | 6.1885 | 3648.6 | 2.9850 | .00421941 | 744.557 | 44115.03 |
| 238 | 56644 | 13481272 | 15.4272 | 6.1972 | 3671.7 | 2.9875 | .00420168 | 747.698 | 44488.09 |
| 239 | 57121 | 13651919 | 15.4596 | 6.2058 | 3694.8 | 2.9900 | .00418410 | 750.840 | 44862.73 |
| **240** | **57600** | **13824000** | **15.4919** | **6.2145** | **3718.0** | **2.9925** | **.00416667** | **753.982** | **45238.93** |
| 241 | 58081 | 13997521 | 15.5242 | 6.2231 | 3741.3 | 2.9950 | .00414938 | 757.123 | 45616.71 |
| 242 | 58564 | 14172488 | 15.5563 | 6.2317 | 3764.6 | 2.9975 | .00413223 | 760.265 | 45996.06 |
| 243 | 59049 | 14348907 | 15.5885 | 6.2403 | 3788.0 | 3.0000 | .00411523 | 763.406 | 46376.98 |
| 244 | 59536 | 14526784 | 15.6205 | 6.2488 | 3811.4 | 3.0025 | .00409836 | 766.548 | 46759.47 |
| 245 | 60025 | 14706125 | 15.6525 | 6.2573 | 3834.9 | 3.0049 | .00408163 | 769.690 | 47143.52 |
| 246 | 60516 | 14886936 | 15.6844 | 6.2658 | 3858.4 | 3.0074 | .00406504 | 772.831 | 47529.16 |
| 247 | 61009 | 15069223 | 15.7162 | 6.2743 | 3881.9 | 3.0098 | .00404858 | 775.973 | 47916.36 |
| 248 | 61504 | 15252992 | 15.7480 | 6.2828 | 3905.5 | 3.0122 | .00403226 | 779.114 | 48305.13 |
| 249 | 62001 | 15438249 | 15.7797 | 6.2912 | 3929.2 | 3.0147 | .00401606 | 782.256 | 48695.47 |
| **250** | **62500** | **15625000** | **15.8114** | **6.2996** | **3952.9** | **3.0171** | **.00400000** | **785.398** | **49087.39** |
| 251 | 63001 | 15813251 | 15.8430 | 6.3080 | 3976.6 | 3.0195 | .00398406 | 788.539 | 49480.87 |
| 252 | 63504 | 16003008 | 15.8745 | 6.3164 | 4000.4 | 3.0219 | .00396825 | 791.681 | 49875.92 |
| 253 | 64009 | 16194277 | 15.9060 | 6.3247 | 4024.2 | 3.0243 | .00395257 | 794.822 | 50272.55 |
| 254 | 64516 | 16387064 | 15.9374 | 6.3330 | 4048.1 | 3.0267 | .00393701 | 797.964 | 50670.75 |
| 255 | 65025 | 16581375 | 15.9687 | 6.3413 | 4072.0 | 3.0291 | .00392157 | 801.105 | 51070.52 |
| 256 | 65536 | 16777216 | 16.0000 | 6.3496 | 4096.0 | 3.0314 | .00390625 | 804.247 | 51471.85 |
| 257 | 66049 | 16974593 | 16.0312 | 6.3579 | 4120.0 | 3.0338 | .00389105 | 807.389 | 51874.76 |
| 258 | 66564 | 17173512 | 16.0624 | 6.3661 | 4144.1 | 3.0362 | .00387597 | 810.530 | 52279.24 |
| 259 | 67081 | 17373979 | 16.0935 | 6.3743 | 4168.2 | 3.0385 | .00386100 | 813.672 | 52685.29 |
| **260** | **67600** | **17576000** | **16.1245** | **6.3825** | **4192.4** | **3.0418** | **.00384615** | **816.813** | **53092.92** |
| 261 | 68121 | 17779581 | 16.1555 | 6.3907 | 4216.6 | 3.0432 | .00383142 | 819.955 | 53502.11 |
| 262 | 68644 | 17984728 | 16.1864 | 6.3988 | 4240.8 | 3.0455 | .00381679 | 823.097 | 53912.87 |
| 263 | 69169 | 18191447 | 16.2173 | 6.4070 | 4265.1 | 3.0478 | .00380228 | 826.238 | 54325.21 |
| 264 | 69696 | 18399744 | 16.2481 | 6.4151 | 4289.5 | 3.0501 | .00378788 | 829.380 | 54739.11 |
| 265 | 70225 | 18609625 | 16.2788 | 6.4232 | 4313.9 | 3.0524 | .00377358 | 832.521 | 55154.59 |
| 266 | 70756 | 18821096 | 16.3095 | 6.4312 | 4338.3 | 3.0547 | .00375940 | 835.663 | 55571.63 |
| 267 | 71289 | 19034163 | 16.3401 | 6.4393 | 4362.8 | 3.0570 | .00374532 | 838.805 | 55990.25 |
| 268 | 71824 | 19248832 | 16.3707 | 6.4473 | 4387.3 | 3.0593 | .00373134 | 841.946 | 56410.44 |
| 269 | 72361 | 19465109 | 16.4012 | 6.4553 | 4411.9 | 3.0616 | .00371747 | 845.088 | 56832.20 |
| **270** | **72900** | **19683000** | **16.4317** | **6.4633** | **4436.5** | **3.0639** | **.00370370** | **848.229** | **57255.53** |
| 271 | 73441 | 19902511 | 16.4621 | 6.4713 | 4461.2 | 3.0662 | .00369004 | 851.371 | 57680.43 |
| 272 | 73984 | 20123648 | 16.4924 | 6.4792 | 4485.9 | 3.0684 | .00367647 | 854.512 | 58106.90 |
| 273 | 74529 | 20346417 | 16.5227 | 6.4872 | 4510.7 | 3.0707 | .00366300 | 857.654 | 58534.94 |
| 274 | 75076 | 20570824 | 16.5529 | 6.4951 | 4535.5 | 3.0729 | .00364964 | 860.796 | 58964.55 |
| 275 | 75625 | 20796875 | 16.5831 | 6.5030 | 4560.4 | 3.0752 | .00363636 | 863.937 | 59395.74 |
| 276 | 76176 | 21024576 | 16.6132 | 6.5108 | 4585.3 | 3.0774 | .00362319 | 867.079 | 59828.49 |
| 277 | 76729 | 21253933 | 16.6433 | 6.5187 | 4610.2 | 3.0796 | .00361011 | 870.220 | 60262.82 |
| 278 | 77284 | 21484952 | 16.6733 | 6.5265 | 4635.2 | 3.0818 | .00359712 | 873.362 | 60698.71 |
| 279 | 77841 | 21717639 | 16.7033 | 6.5343 | 4660.2 | 3.0840 | .00358423 | 876.504 | 61136.18 |
| **280** | **78400** | **21952000** | **16.7332** | **6.5421** | **4685.3** | **3.0863** | **.00357143** | **879.645** | **61575.22** |
| 281 | 78961 | 22188041 | 16.7631 | 6.5499 | 4710.4 | 3.0885 | .00355872 | 882.787 | 62015.82 |
| 282 | 79524 | 22425768 | 16.7929 | 6.5577 | 4735.6 | 3.0907 | .00354610 | 885.928 | 62458.00 |
| 283 | 80089 | 22665187 | 16.8226 | 6.5654 | 4760.8 | 3.0928 | .00353357 | 889.070 | 62901.75 |
| 284 | 80656 | 22906304 | 16.8523 | 6.5731 | 4786.0 | 3.0950 | .00352113 | 892.212 | 63347.07 |
| 285 | 81225 | 23149125 | 16.8819 | 6.5808 | 4811.3 | 3.0972 | .00350877 | 895.353 | 63793.97 |
| 286 | 81796 | 23393656 | 16.9115 | 6.5885 | 4836.7 | 3.0994 | .00349650 | 898.495 | 64242.43 |
| 287 | 82369 | 23639903 | 16.9411 | 6.5962 | 4862.1 | 3.1015 | .00348432 | 901.636 | 64692.46 |
| 288 | 82944 | 23887872 | 16.9706 | 6.6039 | 4887.5 | 3.1037 | .00347222 | 904.778 | 65144.07 |
| 289 | 83521 | 24137569 | 17.0000 | 6.6115 | 4913.0 | 3.1058 | .00346021 | 907.920 | 65597.24 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **290** | **84100** | **24389000** | **17.0294** | **6.6191** | **4938.5** | **3.1080** | **.00344828** | **911.061** | **66051.99** |
| 291 | 84681 | 24642171 | 17.0587 | 6.6267 | 4964.1 | 3.1101 | .00343643 | 914.203 | 66508.30 |
| 292 | 85264 | 24897088 | 17.0880 | 6.6343 | 4989.7 | 3.1123 | .00342466 | 917.344 | 66966.19 |
| 293 | 85849 | 25153757 | 17.1172 | 6.6419 | 5015.4 | 3.1144 | .00341297 | 920.486 | 67425.65 |
| 294 | 86436 | 25412184 | 17.1464 | 6.6494 | 5041.1 | 3.1165 | .00340136 | 923.627 | 67886.68 |
| 295 | 87025 | 25672375 | 17.1756 | 6.6569 | 5066.8 | 3.1186 | .00338983 | 926.769 | 68349.28 |
| 296 | 87616 | 25934336 | 17.2047 | 6.6644 | 5092.6 | 3.1207 | .00337838 | 929.911 | 68813.45 |
| 297 | 88209 | 26198073 | 17.2337 | 6.6719 | 5118.4 | 3.1228 | .00336700 | 933.052 | 69279.19 |
| 298 | 88804 | 26463592 | 17.2627 | 6.6794 | 5144.3 | 3.1249 | .00335570 | 936.194 | 69746.50 |
| 299 | 89401 | 26730899 | 17.2916 | 6.6869 | 5170.2 | 3.1270 | .00334448 | 939.335 | 70215.38 |
| **300** | **90000** | **27000000** | **17.3205** | **6.6943** | **5196.2** | **3.1291** | **.00333333** | **942.477** | **70685.83** |
| 301 | 90601 | 27270901 | 17.3494 | 6.7018 | 5222.2 | 3.1312 | .00332226 | 945.619 | 71157.86 |
| 302 | 91204 | 27543608 | 17.3781 | 6.7092 | 5248.2 | 3.1333 | .00331126 | 948.760 | 71631.45 |
| 303 | 91809 | 27818127 | 17.4069 | 6.7166 | 5274.3 | 3.1354 | .00330033 | 951.902 | 72106.62 |
| 304 | 92416 | 28094464 | 17.4356 | 6.7240 | 5300.4 | 3.1374 | .00328947 | 955.043 | 72583.36 |
| 305 | 93025 | 28372625 | 17.4642 | 6.7313 | 5326.6 | 3.1395 | .00327869 | 958.185 | 73061.66 |
| 306 | 93636 | 28652616 | 17.4929 | 6.7387 | 5352.8 | 3.1416 | .00326797 | 961.327 | 73541.54 |
| 307 | 94249 | 28934443 | 17.5214 | 6.7460 | 5379.1 | 3.1436 | .00325733 | 964.468 | 74022.99 |
| 308 | 94864 | 29218112 | 17.5499 | 6.7533 | 5405.4 | 3.1456 | .00324675 | 967.610 | 74506.01 |
| 309 | 95481 | 29503629 | 17.5784 | 6.7606 | 5431.7 | 3.1477 | .00323625 | 970.751 | 74990.60 |
| **310** | **96100** | **29791000** | **17.6068** | **6.7679** | **5458.1** | **3.1497** | **.00322581** | **973.893** | **75476.76** |
| 311 | 96721 | 30080231 | 17.6352 | 6.7752 | 5484.5 | 3.1518 | .00321543 | 977.034 | 75964.50 |
| 312 | 97344 | 30371328 | 17.6635 | 6.7824 | 5511.0 | 3.1538 | .00320513 | 980.176 | 76453.80 |
| 313 | 97969 | 30664297 | 17.6918 | 6.7897 | 5537.5 | 3.1558 | .00319489 | 983.318 | 76944.67 |
| 314 | 98596 | 30959144 | 17.7200 | 6.7969 | 5564.1 | 3.1578 | .00318471 | 986.459 | 77437.12 |
| 315 | 99225 | 31255875 | 17.7482 | 6.8041 | 5590.7 | 3.1598 | .00317460 | 989.601 | 77931.13 |
| 316 | 99856 | 31554496 | 17.7764 | 6.8113 | 5617.3 | 3.1618 | .00316456 | 992.742 | 78426.72 |
| 317 | 100489 | 31855013 | 17.8045 | 6.8185 | 5644.0 | 3.1638 | .00315457 | 995.884 | 78923.88 |
| 318 | 101124 | 32157432 | 17.8326 | 6.8256 | 5670.7 | 3.1658 | .00314465 | 999.026 | 79422.60 |
| 319 | 101761 | 32461759 | 17.8606 | 6.8328 | 5697.5 | 3.1678 | .00313480 | 1002.167 | 79922.90 |
| **320** | **102400** | **32768000** | **17.8885** | **6.8399** | **5724.3** | **3.1698** | **.00312500** | **1005.309** | **80424.77** |
| 321 | 103041 | 33076161 | 17.9165 | 6.8470 | 5751.2 | 3.1718 | .00311526 | 1008.450 | 80928.21 |
| 322 | 103684 | 33386248 | 17.9444 | 6.8541 | 5778.1 | 3.1737 | .00310559 | 1011.592 | 81433.22 |
| 323 | 104329 | 33698267 | 17.9722 | 6.8612 | 5805.0 | 3.1757 | .00309598 | 1014.734 | 81939.80 |
| 324 | 104976 | 34012224 | 18.0000 | 6.8683 | 5832.0 | 3.1777 | .00308642 | 1017.875 | 82447.96 |
| 325 | 105625 | 34328125 | 18.0278 | 6.8753 | 5859.0 | 3.1796 | .00307692 | 1021.017 | 82957.68 |
| 326 | 106276 | 34645976 | 18.0555 | 6.8824 | 5886.1 | 3.1816 | .00306748 | 1024.158 | 83468.97 |
| 327 | 106929 | 34965783 | 18.0831 | 6.8894 | 5913.2 | 3.1835 | .00305810 | 1027.300 | 83981.84 |
| 328 | 107584 | 35287552 | 18.1108 | 6.8964 | 5940.3 | 3.1855 | .00304878 | 1030.442 | 84496.28 |
| 329 | 108241 | 35611289 | 18.1384 | 6.9034 | 5967.5 | 3.1874 | .00303951 | 1033.583 | 85012.28 |
| **330** | **108900** | **35937000** | **18.1659** | **6.9104** | **5994.7** | **3.1894** | **.00303030** | **1036.725** | **85529.86** |
| 331 | 109561 | 36264691 | 18.1934 | 6.9174 | 6022.0 | 3.1913 | .00302115 | 1039.866 | 86049.01 |
| 332 | 110224 | 36594368 | 18.2209 | 6.9244 | 6049.3 | 3.1932 | .00301205 | 1043.008 | 86569.73 |
| 333 | 110889 | 36926037 | 18.2483 | 6.9313 | 6076.7 | 3.1951 | .00300300 | 1046.149 | 87092.02 |
| 334 | 111556 | 37259704 | 18.2757 | 6.9382 | 6104.1 | 3.1970 | .00299401 | 1049.291 | 87615.88 |
| 335 | 112225 | 37595375 | 18.3030 | 6.9451 | 6131.5 | 3.1989 | .00298507 | 1052.433 | 88141.31 |
| 336 | 112896 | 37933056 | 18.3303 | 6.9521 | 6159.0 | 3.2009 | .00297619 | 1055.574 | 88668.31 |
| 337 | 113569 | 38272753 | 18.3576 | 6.9589 | 6186.5 | 3.2028 | .00296736 | 1058.716 | 89196.88 |
| 338 | 114244 | 38614472 | 18.3848 | 6.9658 | 6214.1 | 3.2047 | .00295858 | 1061.857 | 89727.03 |
| 339 | 114921 | 38958219 | 18.4120 | 6.9727 | 6241.7 | 3.2066 | .00294985 | 1064.999 | 90258.74 |
| **340** | **115600** | **39304000** | **18.4391** | **6.9795** | **6269.3** | **3.2085** | **.00294118** | **1068.141** | **90792.03** |
| 341 | 116281 | 39651821 | 18.4662 | 6.9864 | 6297.0 | 3.2103 | .00293255 | 1071.282 | 91326.88 |
| 342 | 116964 | 40001688 | 18.4932 | 6.9932 | 6324.7 | 3.2122 | .00292398 | 1074.424 | 91863.31 |
| 343 | 117649 | 40353607 | 18.5203 | 7.0000 | 6352.4 | 3.2141 | .00291545 | 1077.565 | 92401.31 |
| 344 | 118336 | 40707584 | 18.5472 | 7.0068 | 6380.2 | 3.2160 | .00290698 | 1080.707 | 92940.88 |
| 345 | 119025 | 41063625 | 18.5742 | 7.0136 | 6408.1 | 3.2178 | .00289855 | 1083.849 | 93482.02 |
| 346 | 119716 | 41421736 | 18.6011 | 7.0203 | 6436.0 | 3.2197 | .00289017 | 1086.990 | 94024.73 |
| 347 | 120409 | 41781923 | 18.6279 | 7.0271 | 6463.9 | 3.2216 | .00288184 | 1090.132 | 94569.01 |
| 348 | 121104 | 42144192 | 18.6548 | 7.0338 | 6491.9 | 3.2234 | .00287356 | 1093.273 | 95114.86 |
| 349 | 121801 | 42508549 | 18.6815 | 7.0406 | 6519.9 | 3.2253 | .00286533 | 1096.415 | 95662.28 |
| **350** | **122500** | **42875000** | **18.7083** | **7.0473** | **6547.9** | **3.2271** | **.00285714** | **1099.557** | **96211.28** |
| 351 | 123201 | 43243551 | 18.7350 | 7.0540 | 6576.0 | 3.2289 | .00284900 | 1102.698 | 96761.84 |
| 352 | 123904 | 43614208 | 18.7617 | 7.0607 | 6604.1 | 3.2308 | .00284091 | 1105.840 | 97313.97 |
| 353 | 124609 | 43986977 | 18.7883 | 7.0674 | 6632.3 | 3.2326 | .00283286 | 1108.981 | 97867.68 |
| 354 | 125316 | 44361864 | 18.8149 | 7.0740 | 6660.5 | 3.2345 | .00282486 | 1112.123 | 98422.96 |
| 355 | 126025 | 44738875 | 18.8414 | 7.0807 | 6688.7 | 3.2363 | .00281690 | 1115.264 | 98979.80 |
| 356 | 126736 | 45118016 | 18.8680 | 7.0873 | 6717.0 | 3.2381 | .00280899 | 1118.406 | 99538.22 |
| 357 | 127449 | 45499293 | 18.8944 | 7.0940 | 6745.3 | 3.2399 | .00280112 | 1121.548 | 100098.21 |
| 358 | 128164 | 45882712 | 18.9209 | 7.1006 | 6773.7 | 3.2417 | .00279330 | 1124.689 | 100659.77 |
| 359 | 128881 | 46268279 | 18.9473 | 7.1072 | 6802.1 | 3.2435 | .00278552 | 1127.831 | 101222.90 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | (Circle $N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **360** | **129600** | **46656000** | **18.9737** | **7.1138** | **6830.5** | **3.2453** | **.00277778** | **1130.972** | **101787.60** |
| 361 | 130321 | 47045881 | 19.0000 | 7.1204 | 6859.0 | 3.2471 | .00277008 | 1134.114 | 102353.87 |
| 362 | 131044 | 47437928 | 19.0263 | 7.1269 | 6887.5 | 3.2489 | .00276243 | 1137.256 | 102921.72 |
| 363 | 131769 | 47832147 | 19.0526 | 7.1335 | 6916.1 | 3.2507 | .00275482 | 1140.397 | 103491.13 |
| 364 | 132496 | 48228544 | 19.0788 | 7.1400 | 6944.7 | 3.2525 | .00274725 | 1143.539 | 104062.12 |
| 365 | 133225 | 48627125 | 19.1050 | 7.1466 | 6973.3 | 3.2543 | .00273973 | 1146.680 | 104634.67 |
| 366 | 133956 | 49027896 | 19.1311 | 7.1531 | 7002.0 | 3.2561 | .00273224 | 1149.822 | 105208.80 |
| 367 | 134689 | 49430863 | 19.1572 | 7.1596 | 7030.7 | 3.2579 | .00272480 | 1152.964 | 105784.49 |
| 368 | 135424 | 49836032 | 19.1833 | 7.1661 | 7059.5 | 3.2597 | .00271739 | 1156.105 | 106361.76 |
| 369 | 136161 | 50243409 | 19.2094 | 7.1726 | 7088.3 | 3.2614 | .00271003 | 1159.247 | 106940.60 |
| **370** | **136900** | **50653000** | **19.2354** | **7.1791** | **7117.1** | **3.2632** | **.00270270** | **1162.388** | **107521.01** |
| 371 | 137641 | 51064811 | 19.2614 | 7.1855 | 7146.0 | 3.2650 | .00269542 | 1165.530 | 108102.99 |
| 372 | 138384 | 51478848 | 19.2873 | 7.1920 | 7174.9 | 3.2668 | .00268817 | 1168.671 | 108686.54 |
| 373 | 139129 | 51895117 | 19.3132 | 7.1984 | 7203.9 | 3.2685 | .00268097 | 1171.813 | 109271.66 |
| 374 | 139876 | 52313624 | 19.3391 | 7.2048 | 7232.8 | 3.2702 | .00267380 | 1174.955 | 109858.35 |
| 375 | 140625 | 52734375 | 19.3649 | 7.2112 | 7261.8 | 3.2719 | .00266667 | 1178.096 | 110446.62 |
| 376 | 141376 | 53157376 | 19.3907 | 7.2177 | 7290.9 | 3.2737 | .00265957 | 1181.238 | 111036.45 |
| 377 | 142129 | 53582633 | 19.4165 | 7.2240 | 7320.0 | 3.2754 | .00265252 | 1184.379 | 111627.86 |
| 378 | 142884 | 54010152 | 19.4422 | 7.2304 | 7349.2 | 3.2772 | .00264550 | 1187.521 | 112220.83 |
| 379 | 143641 | 54439939 | 19.4679 | 7.2368 | 7378.4 | 3.2789 | .00263852 | 1190.663 | 112815.38 |
| **380** | **144400** | **54872000** | **19.4936** | **7.2432** | **7407.6** | **3.2807** | **.00263158** | **1193.804** | **113411.49** |
| 381 | 145161 | 55306341 | 19.5192 | 7.2495 | 7436.8 | 3.2824 | .00262467 | 1196.946 | 114009.18 |
| 382 | 145924 | 55742968 | 19.5448 | 7.2558 | 7466.1 | 3.2841 | .00261780 | 1200.087 | 114608.44 |
| 383 | 146689 | 56181887 | 19.5704 | 7.2622 | 7495.4 | 3.2858 | .00261097 | 1203.229 | 115209.27 |
| 384 | 147456 | 56623104 | 19.5959 | 7.2685 | 7524.8 | 3.2875 | .00260417 | 1206.371 | 115811.67 |
| 385 | 148225 | 57066625 | 19.6214 | 7.2748 | 7554.2 | 3.2892 | .00259740 | 1209.512 | 116415.64 |
| 386 | 148996 | 57512456 | 19.6469 | 7.2811 | 7583.7 | 3.2909 | .00259067 | 1212.654 | 117021.18 |
| 387 | 149769 | 57960603 | 19.6723 | 7.2874 | 7613.2 | 3.2926 | .00258398 | 1215.795 | 117628.30 |
| 388 | 150544 | 58411072 | 19.6977 | 7.2936 | 7642.7 | 3.2943 | .00257732 | 1218.937 | 118236.98 |
| 389 | 151321 | 58863869 | 19.7231 | 7.2999 | 7672.3 | 3.2960 | .00257069 | 1222.079 | 118847.24 |
| **390** | **152100** | **59319000** | **19.7484** | **7.3061** | **7701.9** | **3.2977** | **.00256410** | **1225.220** | **119459.06** |
| 391 | 152881 | 59776471 | 19.7737 | 7.3124 | 7731.5 | 3.2994 | .00255754 | 1228.362 | 120072.46 |
| 392 | 153664 | 60236288 | 19.7990 | 7.3186 | 7761.2 | 3.3011 | .00255102 | 1231.503 | 120687.42 |
| 393 | 154449 | 60698457 | 19.8242 | 7.3248 | 7790.9 | 3.3028 | .00254453 | 1234.645 | 121303.96 |
| 394 | 155236 | 61162984 | 19.8494 | 7.3310 | 7820.7 | 3.3045 | .00253807 | 1237.786 | 121922.07 |
| 395 | 156025 | 61629875 | 19.8746 | 7.3372 | 7850.5 | 3.3061 | .00253165 | 1240.928 | 122541.75 |
| 396 | 156816 | 62099136 | 19.8997 | 7.3434 | 7880.3 | 3.3078 | .00252525 | 1244.070 | 123163.00 |
| 397 | 157609 | 62570773 | 19.9249 | 7.3496 | 7910.2 | 3.3095 | .00251889 | 1247.211 | 123785.82 |
| 398 | 158404 | 63044792 | 19.9499 | 7.3558 | 7940.1 | 3.3111 | .00251256 | 1250.353 | 124410.21 |
| 399 | 159201 | 63521199 | 19.9750 | 7.3619 | 7970.0 | 3.3128 | .00250627 | 1253.494 | 125036.17 |
| **400** | **160000** | **64000000** | **20.0000** | **7.3681** | **8000.0** | **3.3145** | **.00250000** | **1256.636** | **125663.71** |
| 401 | 160801 | 64481201 | 20.0250 | 7.3742 | 8030.0 | 3.3161 | .00249377 | 1259.778 | 126292.81 |
| 402 | 161604 | 64964808 | 20.0499 | 7.3803 | 8061.1 | 3.3178 | .00248756 | 1262.919 | 126923.48 |
| 403 | 162409 | 65450827 | 20.0749 | 7.3864 | 8090.2 | 3.3194 | .00248139 | 1266.061 | 127555.73 |
| 404 | 163216 | 65939264 | 20.0998 | 7.3925 | 8120.3 | 3.3211 | .00247525 | 1269.202 | 128189.55 |
| 405 | 164025 | 66430125 | 20.1246 | 7.3986 | 8150.5 | 3.3227 | .00246914 | 1272.344 | 128824.93 |
| 406 | 164836 | 66923416 | 20.1494 | 7.4047 | 8180.7 | 3.3243 | .00246305 | 1275.486 | 129461.89 |
| 407 | 165649 | 67419143 | 20.1742 | 7.4108 | 8210.9 | 3.3260 | .00245700 | 1278.627 | 130100.42 |
| 408 | 166464 | 67917312 | 20.1990 | 7.4169 | 8241.2 | 3.3276 | .00245098 | 1281.769 | 130740.52 |
| 409 | 167281 | 68417929 | 20.2237 | 7.4229 | 8271.5 | 3.3292 | .00244499 | 1284.910 | 131382.19 |
| **410** | **168100** | **68921000** | **20.2485** | **7.4290** | **8301.9** | **3.3308** | **.00243902** | **1288.052** | **132025.43** |
| 411 | 168921 | 69426531 | 20.2731 | 7.4350 | 8332.3 | 3.3325 | .00243309 | 1291.193 | 132670.24 |
| 412 | 169744 | 69934528 | 20.2978 | 7.4410 | 8362.7 | 3.3341 | .00242718 | 1294.335 | 133316.63 |
| 413 | 170569 | 70444997 | 20.3224 | 7.4470 | 8393.2 | 3.3357 | .00242131 | 1297.477 | 133964.58 |
| 414 | 171396 | 70957944 | 20.3470 | 7.4530 | 8423.7 | 3.3373 | .00241546 | 1300.618 | 134614.10 |
| 415 | 172225 | 71473375 | 20.3715 | 7.4590 | 8454.2 | 3.3390 | .00240964 | 1303.760 | 135265.20 |
| 416 | 173056 | 71991296 | 20.3961 | 7.4650 | 8484.8 | 3.3406 | .00240385 | 1306.901 | 135917.86 |
| 417 | 173889 | 72511713 | 20.4206 | 7.4710 | 8515.4 | 3.3422 | .00239808 | 1310.043 | 136572.10 |
| 418 | 174724 | 73034632 | 20.4450 | 7.4770 | 8546.0 | 3.3438 | .00239234 | 1313.185 | 137227.91 |
| 419 | 175561 | 73560059 | 20.4695 | 7.4829 | 8576.7 | 3.3454 | .00238663 | 1316.326 | 137885.29 |
| **420** | **176400** | **74088000** | **20.4939** | **7.4889** | **8607.4** | **3.3470** | **.00238095** | **1319.468** | **138544.24** |
| 421 | 177241 | 74618461 | 20.5183 | 7.4948 | 8638.2 | 3.3485 | .00237530 | 1322.609 | 139204.76 |
| 422 | 178084 | 75151448 | 20.5426 | 7.5007 | 8669.0 | 3.3501 | .00236967 | 1325.751 | 139866.85 |
| 423 | 178929 | 75686967 | 20.5670 | 7.5067 | 8699.8 | 3.3517 | .00236407 | 1328.893 | 140530.51 |
| 424 | 179776 | 76225024 | 20.5913 | 7.5126 | 8730.7 | 3.3533 | .00235849 | 1332.034 | 141195.74 |
| 425 | 180625 | 76765625 | 20.6155 | 7.5185 | 8761.6 | 3.3559 | .00235294 | 1335.176 | 141862.54 |
| 426 | 181476 | 77308776 | 20.6398 | 7.5244 | 8792.5 | 3.3564 | .00234742 | 1338.317 | 142530.92 |
| 427 | 182329 | 77854483 | 20.6640 | 7.5302 | 8823.5 | 3.3580 | .00234192 | 1341.459 | 143200.86 |
| 428 | 183184 | 78402752 | 20.6882 | 7.5361 | 8854.5 | 3.3596 | .00233645 | 1344.601 | 143872.38 |
| 429 | 184041 | 78953589 | 20.7123 | 7.5420 | 8885.6 | 3.3612 | .00233100 | 1347.742 | 144545.46 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **430** | **184900** | **79507000** | **20.7364** | **7.5478** | **8916.7** | **3.3627** | **.00232558** | **1350.884** | **145220.12** |
| 431 | 185761 | 80062991 | 20.7605 | 7.5537 | 8947.8 | 3.3643 | .00232019 | 1354.025 | 145896.35 |
| 432 | 186624 | 80621568 | 20.7846 | 7.5595 | 8979.0 | 3.3659 | .00231481 | 1357.167 | 146574.15 |
| 433 | 187489 | 81182737 | 20.8087 | 7.5654 | 9010.1 | 3.3674 | .00230947 | 1360.308 | 147253.52 |
| 434 | 188356 | 81746504 | 20.8327 | 7.5712 | 9041.4 | 3.3690 | .00230415 | 1363.450 | 147934.46 |
| 435 | 189225 | 82312875 | 20.8567 | 7.5770 | 9072.7 | 3.3705 | .00229885 | 1366.592 | 148616.97 |
| 436 | 190096 | 82881856 | 20.8806 | 7.5828 | 9104.0 | 3.3720 | .00229358 | 1369.733 | 149301.05 |
| 437 | 190969 | 83453453 | 20.9045 | 7.5886 | 9135.3 | 3.3736 | .00228833 | 1372.875 | 149986.70 |
| 438 | 191844 | 84027672 | 20.9284 | 7.5944 | 9166.7 | 3.3752 | .00228311 | 1376.016 | 150673.92 |
| 439 | 192721 | 84604519 | 20.9523 | 7.6001 | 9198.1 | 3.3767 | .00227790 | 1379.158 | 151362.72 |
| **440** | **193600** | **85184000** | **20.9762** | **7.6059** | **9229.5** | **3.3783** | **.00227273** | **1382.300** | **152053.08** |
| 441 | 194481 | 85766121 | 21.0000 | 7.6117 | 9261.0 | 3.3798 | .00226757 | 1385.441 | 152745.02 |
| 442 | 195364 | 86350888 | 21.0238 | 7.6174 | 9292.5 | 3.3813 | .00226244 | 1388.583 | 153438.53 |
| 443 | 196249 | 86938307 | 21.0476 | 7.6232 | 9324.1 | 3.3828 | .00225734 | 1391.724 | 154133.60 |
| 444 | 197136 | 87528384 | 21.0713 | 7.6289 | 9355.7 | 3.3844 | .00225225 | 1394.866 | 154830.25 |
| 445 | 198025 | 88121125 | 21.0950 | 7.6346 | 9387.3 | 3.3859 | .00224719 | 1398.008 | 155528.47 |
| 446 | 198916 | 88716536 | 21.1187 | 7.6403 | 9419.0 | 3.3874 | .00224215 | 1401.149 | 156228.26 |
| 447 | 199809 | 89314623 | 21.1424 | 7.6460 | 9450.7 | 3.3889 | .00223714 | 1404.291 | 156929.62 |
| 448 | 200704 | 89915392 | 21.1660 | 7.6517 | 9482.4 | 3.3904 | .00223214 | 1407.432 | 157632.55 |
| 449 | 201601 | 90518849 | 21.1896 | 7.6574 | 9514.2 | 3.3919 | .00222717 | 1410.574 | 158337.05 |
| **450** | **202500** | **91125000** | **21.2132** | **7.6631** | **9546.0** | **3.3935** | **.00222222** | **1413.716** | **159043.13** |
| 451 | 203401 | 91733851 | 21.2368 | 7.6688 | 9577.8 | 3.3950 | .00221729 | 1416.857 | 159750.77 |
| 452 | 204304 | 92345408 | 21.2603 | 7.6744 | 9609.6 | 3.3965 | .00221239 | 1419.999 | 160459.99 |
| 453 | 205209 | 92959677 | 21.2838 | 7.6801 | 9641.5 | 3.3980 | .00220751 | 1423.140 | 161170.77 |
| 454 | 206116 | 93576664 | 21.3073 | 7.6857 | 9673.5 | 3.3995 | .00220264 | 1426.282 | 161883.13 |
| 455 | 207025 | 94196375 | 21.3307 | 7.6914 | 9705.5 | 3.4010 | .00219780 | 1429.423 | 162597.05 |
| 456 | 207936 | 94818816 | 21.3542 | 7.6970 | 9737.5 | 3.4025 | .00219298 | 1432.565 | 163312.55 |
| 457 | 208849 | 95443993 | 21.3776 | 7.7026 | 9769.5 | 3.4039 | .00218818 | 1435.707 | 164029.62 |
| 458 | 209764 | 96071912 | 21.4009 | 7.7082 | 9801.6 | 3.4054 | .00218341 | 1438.848 | 164748.26 |
| 459 | 210681 | 96702579 | 21.4243 | 7.7138 | 9833.8 | 3.4069 | .00217865 | 1441.990 | 165468.47 |
| **460** | **211600** | **97336000** | **21.4476** | **7.7194** | **9865.9** | **3.4084** | **.00217391** | **1445.131** | **166190.25** |
| 461 | 212521 | 97972181 | 21.4709 | 7.7250 | 9898.1 | 3.4199 | .00216920 | 1448.273 | 166913.60 |
| 462 | 213444 | 98611128 | 21.4942 | 7.7306 | 9930.3 | 3.4113 | .00216450 | 1451.415 | 167638.52 |
| 463 | 214369 | 99252847 | 21.5174 | 7.7362 | 9962.6 | 3.4128 | .00215983 | 1454.556 | 168365.02 |
| 464 | 215296 | 99897344 | 21.5407 | 7.7418 | 9994.8 | 3.4143 | .00215517 | 1457.698 | 169093.08 |
| 465 | 216225 | 100544625 | 21.5639 | 7.7473 | 10027. | 3.4158 | .00215054 | 1460.839 | 169822.72 |
| 466 | 217156 | 101194696 | 21.5870 | 7.7529 | 10060. | 3.4173 | .00214592 | 1463.981 | 170553.92 |
| 467 | 218089 | 101847563 | 21.6102 | 7.7584 | 10092. | 3.4187 | .00214133 | 1467.123 | 171286.70 |
| 468 | 219024 | 102503232 | 21.6333 | 7.7639 | 10124. | 3.4202 | .00213675 | 1470.264 | 172021.05 |
| 469 | 219961 | 103161709 | 21.6564 | 7.7695 | 10157. | 3.4217 | .00213220 | 1473.406 | 172756.96 |
| **470** | **220900** | **103823000** | **21.6795** | **7.7750** | **10189.** | **3.4231** | **.00212766** | **1476.547** | **173494.45** |
| 471 | 221841 | 104487111 | 21.7025 | 7.7805 | 10222. | 3.4246 | .00212314 | 1479.689 | 174233.51 |
| 472 | 222784 | 105154048 | 21.7256 | 7.7860 | 10255. | 3.4260 | .00211864 | 1482.830 | 174974.14 |
| 473 | 223729 | 105823817 | 21.7486 | 7.7915 | 10287. | 3.4275 | .00211416 | 1485.972 | 175716.34 |
| 474 | 224676 | 106496424 | 21.7715 | 7.7970 | 10320. | 3.4289 | .00210970 | 1489.114 | 176460.12 |
| 475 | 225625 | 107171875 | 21.7945 | 7.8025 | 10352. | 3.4304 | .00210526 | 1492.255 | 177205.46 |
| 476 | 226576 | 107850176 | 21.8174 | 7.8079 | 10385. | 3.4318 | .00210084 | 1495.397 | 177952.37 |
| 477 | 227529 | 108531333 | 21.8403 | 7.8134 | 10418. | 3.4332 | .00209644 | 1498.538 | 178700.86 |
| 478 | 228484 | 109215352 | 21.8632 | 7.8188 | 10450. | 3.4347 | .00209205 | 1501.680 | 179450.91 |
| 479 | 229441 | 109902239 | 21.8861 | 7.8243 | 10483. | 3.4361 | .00208768 | 1504.822 | 180202.54 |
| **480** | **230400** | **110592000** | **21.9089** | **7.8297** | **10516.** | **3.4375** | **.00208333** | **1507.963** | **180955.74** |
| 481 | 231361 | 111284641 | 21.9317 | 7.8352 | 10549. | 3.4390 | .00207900 | 1511.105 | 181710.50 |
| 482 | 232324 | 111980168 | 21.9545 | 7.8406 | 10582. | 3.4404 | .00207469 | 1514.246 | 182466.84 |
| 483 | 233289 | 112678587 | 21.9773 | 7.8460 | 10615. | 3.4418 | .00207039 | 1517.388 | 183224.75 |
| 484 | 234256 | 113379904 | 22.0000 | 7.8514 | 10648. | 3.4433 | .00206612 | 1520.530 | 183984.23 |
| 485 | 235225 | 114084125 | 22.0227 | 7.8568 | 10681. | 3.4447 | .00206186 | 1523.671 | 184745.28 |
| 486 | 236196 | 114791256 | 22.0454 | 7.8622 | 10714. | 3.4461 | .00205761 | 1526.813 | 185507.90 |
| 487 | 237169 | 115501303 | 22.0681 | 7.8676 | 10747. | 3.4475 | .00205339 | 1529.954 | 186272.10 |
| 488 | 238144 | 116214272 | 22.0907 | 7.8730 | 10780. | 3.4489 | .00204918 | 1533.096 | 187037.86 |
| 489 | 239121 | 116930169 | 22.1133 | 7.8784 | 10813. | 3.4504 | .00204499 | 1536.238 | 187805.19 |
| **490** | **240100** | **117649000** | **22.1359** | **7.8837** | **10847.** | **3.4518** | **.00204082** | **1539.379** | **188574.10** |
| 491 | 241081 | 118370771 | 22.1585 | 7.8891 | 10880. | 3.4532 | .00203666 | 1542.521 | 189344.57 |
| 492 | 242064 | 119095488 | 22.1811 | 7.8944 | 10913. | 3.4546 | .00203252 | 1545.662 | 190116.62 |
| 493 | 243049 | 119823157 | 22.2036 | 7.8998 | 10946. | 3.4560 | .00202840 | 1548.804 | 190890.24 |
| 494 | 244036 | 120553784 | 22.2261 | 7.9051 | 10980. | 3.4574 | .00202429 | 1551.945 | 191665.43 |
| 495 | 245025 | 121287375 | 22.2486 | 7.9105 | 11013. | 3.4588 | .00202020 | 1555.087 | 192442.18 |
| 496 | 246016 | 122023936 | 22.2711 | 7.9158 | 11046. | 3.4602 | .00201613 | 1558.229 | 193220.51 |
| 497 | 247009 | 122763473 | 22.2935 | 7.9211 | 11080. | 3.4616 | .00201207 | 1561.370 | 194000.41 |
| 498 | 248004 | 123505992 | 22.3159 | 7.9264 | 11113. | 3.4630 | .00200803 | 1564.512 | 194781.89 |
| 499 | 249001 | 124251499 | 22.3383 | 7.9317 | 11147. | 3.4643 | .00200401 | 1567.653 | 195564.93 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **500** | **250000** | **125000000** | **22.3607** | **7.9370** | **11180** | **3.4657** | **.00200000** | **1570.795** | **196349.54** |
| 501 | 251001 | 125751501 | 22.3830 | 7.9423 | 11214 | 3.4671 | .00199601 | 1573.937 | 197135.72 |
| 502 | 252004 | 126506008 | 22.4054 | 7.9476 | 11247 | 3.4685 | .00199203 | 1577.078 | 197923.48 |
| 503 | 253009 | 127263527 | 22.4277 | 7.9528 | 11281 | 3.4699 | .00198807 | 1580.220 | 198712.80 |
| 504 | 254016 | 128024064 | 22.4499 | 7.9581 | 11315 | 3.4713 | .00198413 | 1583.361 | 199503.70 |
| 505 | 255025 | 128787625 | 22.4722 | 7.9634 | 11348 | 3.4726 | .00198020 | 1586.503 | 200296.17 |
| 506 | 256036 | 129554216 | 22.4944 | 7.9686 | 11382 | 3.4740 | .00197628 | 1589.645 | 201090.20 |
| 507 | 257049 | 130323843 | 22.5167 | 7.9739 | 11416 | 3.4754 | .00197239 | 1592.786 | 201885.81 |
| 508 | 258064 | 131096512 | 22.5389 | 7.9791 | 11450 | 3.4768 | .00196850 | 1595.928 | 202682.99 |
| 509 | 259081 | 131872229 | 22.5610 | 7.9843 | 11484 | 3.4781 | .00196464 | 1599.069 | 203481.74 |
| **510** | **260100** | **132651000** | **22.5832** | **7.9896** | **11517** | **3.4795** | **.00196078** | **1602.211** | **204282.06** |
| 511 | 261121 | 133432831 | 22.6053 | 7.9948 | 11551 | 3.4808 | .00195695 | 1605.352 | 205083.95 |
| 512 | 262144 | 134217728 | 22.6274 | 8.0000 | 11585 | 3.4822 | .00195313 | 1608.494 | 205887.42 |
| 513 | 263169 | 135005697 | 22.6495 | 8.0052 | 11619 | 3.4836 | .00194932 | 1611.636 | 206692.45 |
| 514 | 264196 | 135796744 | 22.6716 | 8.0104 | 11653 | 3.4849 | .00194553 | 1614.777 | 207499.05 |
| 515 | 265225 | 136590875 | 22.6936 | 8.0156 | 11687 | 3.4863 | .00194175 | 1617.919 | 208307.23 |
| 516 | 266256 | 137388096 | 22.7156 | 8.0208 | 11721 | 3.4876 | .00193798 | 1621.060 | 209116.97 |
| 517 | 267289 | 138188413 | 22.7376 | 8.0260 | 11755 | 3.4890 | .00193424 | 1624.202 | 209928.29 |
| 518 | 268324 | 138991832 | 22.7596 | 8.0311 | 11789 | 3.4904 | .00193050 | 1627.344 | 210741.18 |
| 519 | 269361 | 139798359 | 22.7816 | 8.0363 | 11824 | 3.4917 | .00192678 | 1630.485 | 211555.63 |
| **520** | **270400** | **140608000** | **22.8035** | **8.0415** | **11858** | **3.4930** | **.00192308** | **1633.627** | **212371.66** |
| 521 | 271441 | 141420761 | 22.8254 | 8.0466 | 11892 | 3.4944 | .00191939 | 1636.768 | 213189.26 |
| 522 | 272484 | 142236648 | 22.8473 | 8.0517 | 11926 | 3.4957 | .00191571 | 1639.910 | 214008.43 |
| 523 | 273529 | 143055667 | 22.8692 | 8.0569 | 11960 | 3.4970 | .00191205 | 1643.052 | 214829.17 |
| 524 | 274576 | 143877824 | 22.8910 | 8.0620 | 11995 | 3.4984 | .00190840 | 1646.193 | 215651.49 |
| 525 | 275625 | 144703125 | 22.9129 | 8.0671 | 12029 | 3.4997 | .00190476 | 1649.335 | 216475.37 |
| 526 | 276676 | 145531576 | 22.9347 | 8.0723 | 12064 | 3.5010 | .00190114 | 1652.476 | 217300.82 |
| 527 | 277729 | 146363183 | 22.9565 | 8.0774 | 12098 | 3.5024 | .00189753 | 1655.618 | 218127.85 |
| 528 | 278784 | 147197952 | 22.9783 | 8.0825 | 12133 | 3.5037 | .00189394 | 1658.760 | 218956.44 |
| 529 | 279841 | 148035889 | 23.0000 | 8.0876 | 12167 | 3.5050 | .00189036 | 1661.901 | 219786.61 |
| **530** | **280900** | **148877000** | **23.0217** | **8.0927** | **12202** | **3.5064** | **.00188679** | **1665.043** | **220618.34** |
| 531 | 281961 | 149721291 | 23.0434 | 8.0978 | 12236 | 3.5077 | .00188324 | 1668.184 | 221451.65 |
| 532 | 283024 | 150568768 | 23.0651 | 8.1028 | 12271 | 3.5090 | .00187970 | 1671.326 | 222286.53 |
| 533 | 284089 | 151419437 | 23.0868 | 8.1079 | 12305 | 3.5103 | .00187617 | 1674.467 | 223122.98 |
| 534 | 285156 | 152273304 | 23.1084 | 8.1130 | 12340 | 3.5116 | .00187266 | 1677.609 | 223961.00 |
| 535 | 286225 | 153130375 | 23.1301 | 8.1180 | 12375 | 3.5130 | .00186916 | 1680.751 | 224800.59 |
| 536 | 287296 | 153990656 | 23.1517 | 8.1231 | 12410 | 3.5143 | .00186567 | 1683.892 | 225641.75 |
| 537 | 288369 | 154854153 | 23.1733 | 8.1281 | 12444 | 3.5156 | .00186220 | 1687.034 | 226484.48 |
| 538 | 289444 | 155720872 | 23.1948 | 8.1332 | 12479 | 3.5169 | .00185874 | 1690.175 | 227328.79 |
| 539 | 290521 | 156590819 | 23.2164 | 8.1382 | 12514 | 3.5182 | .00185529 | 1693.317 | 228174.66 |
| **540** | **291600** | **157464000** | **23.2379** | **8.1433** | **12549** | **3.5195** | **.00185185** | **1696.459** | **229022.10** |
| 541 | 292681 | 158340421 | 23.2594 | 8.1483 | 12583 | 3.5208 | .00184843 | 1699.600 | 229871.12 |
| 542 | 293764 | 159220088 | 23.2809 | 8.1533 | 12618 | 3.5221 | .00184502 | 1702.742 | 230721.71 |
| 543 | 294849 | 160103007 | 23.3024 | 8.1583 | 12653 | 3.5234 | .00184162 | 1705.883 | 231573.86 |
| 544 | 295936 | 160989184 | 23.3238 | 8.1633 | 12688 | 3.5247 | .00183824 | 1709.025 | 232427.59 |
| 545 | 297025 | 161878625 | 23.3452 | 8.1683 | 12723 | 3.5260 | .00183486 | 1712.167 | 233282.89 |
| 546 | 298116 | 162771336 | 23.3666 | 8.1733 | 12758 | 3.5273 | .00183150 | 1715.308 | 234139.76 |
| 547 | 299209 | 163667323 | 23.3880 | 8.1783 | 12793 | 3.5286 | .00182815 | 1718.450 | 234998.20 |
| 548 | 300304 | 164566592 | 23.4094 | 8.1833 | 12828 | 3.5299 | .00182482 | 1721.591 | 235858.21 |
| 549 | 301401 | 165469149 | 23.4307 | 8.1882 | 12863 | 3.5311 | .00182149 | 1724.733 | 236719.79 |
| **550** | **302500** | **166375000** | **23.4521** | **8.1932** | **12899** | **3.5324** | **.00181818** | **1727.875** | **237582.94** |
| 551 | 303601 | 167284151 | 23.4734 | 8.1982 | 12934 | 3.5337 | .00181488 | 1731.016 | 238447.67 |
| 552 | 304704 | 168196608 | 23.4947 | 8.2031 | 12969 | 3.5350 | .00181159 | 1734.158 | 239313.96 |
| 553 | 305809 | 169112377 | 23.5160 | 8.2081 | 13004 | 3.5363 | .00180832 | 1737.299 | 240181.83 |
| 554 | 306916 | 170031464 | 23.5372 | 8.2130 | 13040 | 3.5376 | .00180505 | 1740.441 | 241051.26 |
| 555 | 308025 | 170953875 | 23.5584 | 8.2180 | 13075 | 3.5388 | .00180180 | 1743.582 | 241922.27 |
| 556 | 309136 | 171879616 | 23.5797 | 8.2229 | 13110 | 3.5401 | .00179856 | 1746.724 | 242794.85 |
| 557 | 310249 | 172808693 | 23.6008 | 8.2278 | 13146 | 3.5414 | .00179533 | 1749.866 | 243668.99 |
| 558 | 311364 | 173741112 | 23.6220 | 8.2327 | 13181 | 3.5426 | .00179211 | 1753.007 | 244544.71 |
| 559 | 312481 | 174676879 | 23.6432 | 8.2377 | 13217 | 3.5439 | .00178891 | 1756.149 | 245422.00 |
| **560** | **313600** | **175616000** | **23.6643** | **8.2426** | **13252** | **3.5451** | **.00178571** | **1759.290** | **246300.86** |
| 561 | 314721 | 176558481 | 23.6854 | 8.2475 | 13288 | 3.5464 | .00178253 | 1762.432 | 247181.30 |
| 562 | 315844 | 177504328 | 23.7065 | 8.2524 | 13323 | 3.5477 | .00177936 | 1765.574 | 248063.30 |
| 563 | 316969 | 178453547 | 23.7276 | 8.2573 | 13359 | 3.5490 | .00177620 | 1768.715 | 248946.87 |
| 564 | 318096 | 179406144 | 23.7487 | 8.2621 | 13394 | 3.5502 | .00177305 | 1771.857 | 249832.01 |
| 565 | 319225 | 180362125 | 23.7697 | 8.2670 | 13430 | 3.5515 | .00176991 | 1774.998 | 250718.73 |
| 566 | 320356 | 181321496 | 23.7908 | 8.2719 | 13466 | 3.5527 | .00176678 | 1778.140 | 251607.01 |
| 567 | 321489 | 182284263 | 23.8118 | 8.2768 | 13501 | 3.5540 | .00176367 | 1781.282 | 252496.87 |
| 568 | 322624 | 183250432 | 23.8328 | 8.2816 | 13537 | 3.5553 | .00176056 | 1784.423 | 253388.30 |
| 569 | 323761 | 184220009 | 23.8537 | 8.2865 | 13573 | 3.5565 | .00175747 | 1787.565 | 254281.29 |

### Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **570** | **324900** | **185193000** | **23.8747** | **8.2913** | **13609** | **3.5577** | **.00175439** | **1790.706** | **255175.86** |
| 571 | 326041 | 186169411 | 23.8956 | 8.2962 | 13644 | 3.5590 | .00175131 | 1793.848 | 256072.00 |
| 572 | 327184 | 187149248 | 23.9165 | 8.3010 | 13680 | 3.5602 | .00174825 | 1796.989 | 256969.71 |
| 573 | 328329 | 188132517 | 23.9374 | 8.3059 | 13716 | 3.5615 | .00174520 | 1800.131 | 257868.99 |
| 574 | 329476 | 189119224 | 23.9583 | 8.3107 | 13752 | 3.5627 | .00174216 | 1803.273 | 258769.85 |
| 575 | 330625 | 190109375 | 23.9792 | 8.3155 | 13788 | 3.5640 | .00173913 | 1806.414 | 259672.27 |
| 576 | 331776 | 191102976 | 24.0000 | 8.3203 | 13824 | 3.5652 | .00173611 | 1809.556 | 260576.26 |
| 577 | 332929 | 192100033 | 24.0208 | 8.3251 | 13860 | 3.5664 | .00173310 | 1812.697 | 261481.83 |
| 578 | 334084 | 193100552 | 24.0416 | 8.3300 | 13896 | 3.5677 | .00173010 | 1815.839 | 262388.96 |
| 579 | 335241 | 194104539 | 24.0624 | 8.3348 | 13932 | 3.5689 | .00172712 | 1818.981 | 263297.67 |
| **580** | **336400** | **195112000** | **24.0832** | **8.3396** | **13968** | **3.5702** | **.00172414** | **1822.122** | **264207.94** |
| 581 | 337561 | 196122941 | 24.1039 | 8.3443 | 14004 | 3.5714 | .00172117 | 1825.264 | 265119.79 |
| 582 | 338724 | 197137368 | 24.1247 | 8.3491 | 14040 | 3.5726 | .00171821 | 1828.405 | 266033.21 |
| 583 | 339889 | 198155287 | 24.1454 | 8.3539 | 14077 | 3.5738 | .00171527 | 1831.547 | 266948.20 |
| 584 | 341056 | 199176704 | 24.1661 | 8.3587 | 14113 | 3.5751 | .00171233 | 1834.689 | 267864.76 |
| 585 | 342225 | 200201625 | 24.1868 | 8.3634 | 14149 | 3.5763 | .00170940 | 1837.830 | 268782.89 |
| 586 | 343396 | 201230056 | 24.2074 | 8.3682 | 14186 | 3.5775 | .00170648 | 1840.972 | 269702.59 |
| 587 | 344569 | 202262003 | 24.2281 | 8.3730 | 14222 | 3.5787 | .00170358 | 1844.113 | 270623.86 |
| 588 | 345744 | 203297472 | 24.2487 | 8.3777 | 14258 | 3.5799 | .00170068 | 1847.255 | 271546.70 |
| 589 | 346921 | 204336469 | 24.2693 | 8.3825 | 14295 | 3.5812 | .00169779 | 1850.397 | 272471.12 |
| **590** | **348100** | **205379000** | **24.2899** | **8.3872** | **14331** | **3.5824** | **.00169492** | **1853.538** | **273397.10** |
| 591 | 349281 | 206425071 | 24.3105 | 8.3919 | 14368 | 3.5886 | .00169205 | 1856.680 | 274324.66 |
| 592 | 350464 | 207474688 | 24.3311 | 8.3967 | 14404 | 3.5848 | .00168919 | 1859.821 | 275253.78 |
| 593 | 351649 | 208527857 | 24.3516 | 8.4014 | 14440 | 3.5860 | .00168634 | 1862.963 | 276184.48 |
| 594 | 352836 | 209584584 | 24.3721 | 8.4061 | 14477 | 3.5872 | .00168350 | 1866.104 | 277116.75 |
| 595 | 354025 | 210644875 | 24.3926 | 8.4108 | 14514 | 3.5884 | .00168067 | 1869.246 | 278050.58 |
| 596 | 355216 | 211708736 | 24.4131 | 8.4155 | 14550 | 3.5896 | .00167785 | 1872.388 | 278985.99 |
| 597 | 356409 | 212776173 | 24.4336 | 8.4202 | 14587 | 3.5908 | .00167504 | 1875.529 | 279922.97 |
| 598 | 357604 | 213847192 | 24.4540 | 8.4249 | 14624 | 3.5920 | .00167224 | 1878.671 | 280861.52 |
| 599 | 358801 | 214921799 | 24.4745 | 8.4296 | 14660 | 3.5932 | .00166945 | 1881.812 | 281801.65 |
| **600** | **360000** | **216000000** | **24.4949** | **8.4343** | **14697** | **3.5944** | **.00166667** | **1884.954** | **282743.34** |
| 601 | 361201 | 217081801 | 24.5153 | 8.4390 | 14734 | 3.5956 | .00166389 | 1888.096 | 283686.60 |
| 602 | 362404 | 218167208 | 24.5357 | 8.4437 | 14770 | 3.5958 | .00166113 | 1891.237 | 284631.44 |
| 603 | 363609 | 219256227 | 24.5561 | 8.4484 | 14807 | 3.5980 | .00165837 | 1894.379 | 285577.84 |
| 604 | 364816 | 220348864 | 24.5764 | 8.4530 | 14844 | 3.5992 | .00165563 | 1897.520 | 286525.82 |
| 605 | 366025 | 221445125 | 24.5967 | 8.4577 | 14881 | 3.6004 | .00165289 | 1900.662 | 287475.36 |
| 606 | 367236 | 222545016 | 24.6171 | 8.4623 | 14918 | 3.6016 | .00165017 | 1903.804 | 288426.48 |
| 607 | 368449 | 223648543 | 24.6374 | 8.4670 | 14955 | 3.6028 | .00164745 | 1906.945 | 289379.17 |
| 608 | 369664 | 224755712 | 24.6577 | 8.4716 | 14992 | 3.6040 | .00164474 | 1910.087 | 290333.43 |
| 609 | 370881 | 225866529 | 24.6779 | 8.4763 | 15029 | 3.6052 | .00164204 | 1913.228 | 291289.26 |
| **610** | **372100** | **226981000** | **24.6982** | **8.4809** | **15066** | **3.6063** | **.00163934** | **1916.370** | **292246.66** |
| 611 | 373321 | 228099131 | 24.7184 | 8.4856 | 15103 | 3.6075 | .00163666 | 1919.511 | 293205.63 |
| 612 | 374544 | 229220928 | 24.7386 | 8.4902 | 15140 | 3.6087 | .00163399 | 1922.653 | 294166.17 |
| 613 | 375769 | 230346397 | 24.7588 | 8.4948 | 15177 | 3.6099 | .00163132 | 1925.795 | 295128.28 |
| 614 | 376996 | 231475544 | 24.7790 | 8.4994 | 15214 | 3.6111 | .00162866 | 1928.936 | 296091.97 |
| 615 | 378225 | 232608375 | 24.7992 | 8.5040 | 15252 | 3.6122 | .00162602 | 1932.078 | 297057.22 |
| 616 | 379456 | 233744896 | 24.8193 | 8.5086 | 15289 | 3.6134 | .00162338 | 1935.219 | 298024.05 |
| 617 | 380689 | 234885113 | 24.8395 | 8.5132 | 15326 | 3.6146 | .00162075 | 1938.361 | 298992.44 |
| 618 | 381924 | 236029032 | 24.8596 | 8.5178 | 15363 | 3.6158 | .00161812 | 1941.503 | 299962.41 |
| 619 | 383161 | 237176659 | 24.8797 | 8.5224 | 15400 | 3.6169 | .00161551 | 1944.644 | 300933.95 |
| **620** | **384400** | **238328000** | **24.8998** | **8.5270** | **15437** | **3.6181** | **.00161290** | **1947.786** | **301907.05** |
| 621 | 385641 | 239483061 | 24.9199 | 8.5316 | 15475 | 3.6192 | .00161031 | 1950.927 | 302881.73 |
| 622 | 386884 | 240641848 | 24.9399 | 8.5362 | 15513 | 3.6204 | .00160772 | 1954.069 | 303857.98 |
| 623 | 388129 | 241804367 | 24.9600 | 8.5408 | 15550 | 3.6216 | .00160514 | 1957.211 | 304835.80 |
| 624 | 389376 | 242970624 | 24.9800 | 8.5453 | 15588 | 3.6227 | .00160256 | 1960.352 | 305815.20 |
| 625 | 390625 | 244140625 | 25.0000 | 8.5499 | 15625 | 3.6239 | .00160000 | 1963.494 | 306796.16 |
| 626 | 391876 | 245314376 | 25.0200 | 8.5544 | 15663 | 3.6250 | .00159744 | 1966.635 | 307778.69 |
| 627 | 393129 | 246491883 | 25.0400 | 8.5590 | 15700 | 3.6262 | .00159490 | 1969.777 | 308762.79 |
| 628 | 394384 | 247673152 | 25.0599 | 8.5635 | 15738 | 3.6274 | .00159236 | 1972.919 | 309748.47 |
| 629 | 395641 | 248858189 | 25.0799 | 8.5681 | 15775 | 3.6285 | .00158983 | 1976.060 | 310735.71 |
| **630** | **396900** | **250047000** | **25.0998** | **8.5726** | **15813** | **3.6297** | **.00158730** | **1979.202** | **311724.53** |
| 631 | 398161 | 251239591 | 25.1197 | 8.5772 | 15850 | 3.6309 | .00158479 | 1982.343 | 312714.92 |
| 632 | 399424 | 252435968 | 25.1396 | 8.5817 | 15888 | 3.6320 | .00158228 | 1985.485 | 313706.88 |
| 633 | 400689 | 253636137 | 25.1595 | 8.5862 | 15926 | 3.6331 | .00157978 | 1988.626 | 314700.40 |
| 634 | 401956 | 254840104 | 25.1794 | 8.5907 | 15964 | 3.6343 | .00157729 | 1991.768 | 315695.50 |
| 635 | 403225 | 256047875 | 25.1992 | 8.5952 | 16002 | 3.6354 | .00157480 | 1994.910 | 316692.17 |
| 636 | 404496 | 257259456 | 25.2190 | 8.5997 | 16040 | 3.6366 | .00157233 | 1998.051 | 317690.42 |
| 637 | 405769 | 258474853 | 25.2389 | 8.6043 | 16077 | 3.6377 | .00156986 | 2001.193 | 318690.23 |
| 638 | 407044 | 259694072 | 25.2587 | 8.6088 | 16115 | 3.6389 | .00156740 | 2004.334 | 319691.61 |
| 639 | 408321 | 260917119 | 25.2784 | 8.6132 | 16153 | 3.6400 | .00156495 | 2007.476 | 320694.56 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **640** | **409600** | **262144000** | **25.2982** | **8.6177** | **16191** | **3.6411** | **.00156250** | **2010.618** | **321699.09** |
| 641 | 410881 | 263374721 | 25.3180 | 8.6222 | 16229 | 3.6423 | .00156006 | 2013.759 | 322705.18 |
| 642 | 412164 | 264609288 | 25.3377 | 8.6267 | 16267 | 3.6435 | .00155763 | 2016.901 | 323712.85 |
| 643 | 413449 | 265847707 | 25.3574 | 8.6312 | 16305 | 3.6446 | .00155521 | 2020.042 | 324722.09 |
| 644 | 414736 | 267089984 | 25.3772 | 8.6357 | 16343 | 3.6457 | .00155280 | 2023.184 | 325732.89 |
| 645 | 416025 | 268336125 | 25.3969 | 8.6401 | 16381 | 3.6468 | .00155039 | 2026.326 | 326745.27 |
| 646 | 417316 | 269586136 | 25.4165 | 8.6446 | 16419 | 3.6479 | .00154799 | 2029.467 | 327759.22 |
| 647 | 418609 | 270840023 | 25.4362 | 8.6490 | 16457 | 3.6499 | .00154560 | 2032.609 | 328774.74 |
| 648 | 419904 | 272097792 | 25.4558 | 8.6535 | 16495 | 3.6502 | .00154321 | 2035.750 | 329791.83 |
| 649 | 421201 | 273359449 | 25.4755 | 8.6579 | 16534 | 3.6513 | .00154083 | 2038.892 | 330810.49 |
| **650** | **422500** | **274625000** | **25.4951** | **8.6624** | **16572** | **3.6524** | **.00153846** | **2042.034** | **331830.72** |
| 651 | 423801 | 275894451 | 25.5147 | 8.6668 | 16610 | 3.6536 | .00153610 | 2045.175 | 332852.53 |
| 652 | 425104 | 277167808 | 25.5343 | 8.6713 | 16648 | 3.6547 | .00153374 | 2048.317 | 333875.90 |
| 653 | 426409 | 278445077 | 25.5539 | 8.6757 | 16687 | 3.6558 | .00153139 | 2051.458 | 334900.85 |
| 654 | 427716 | 279726264 | 25.5734 | 8.6801 | 16725 | 3.6569 | .00152905 | 2054.600 | 335927.36 |
| 655 | 429025 | 281011375 | 25.5930 | 8.6845 | 16764 | 3.6580 | .00152672 | 2057.741 | 336955.45 |
| 656 | 430336 | 282300416 | 25.6125 | 8.6890 | 16802 | 3.6592 | .00152439 | 2060.883 | 337985.10 |
| 657 | 431649 | 283593393 | 25.6320 | 8.6934 | 16840 | 3.6603 | .00152207 | 2064.025 | 339016.33 |
| 658 | 432964 | 284890312 | 25.6515 | 8.6978 | 16879 | 3.6614 | .00151976 | 2067.166 | 340049.13 |
| 659 | 434281 | 286191179 | 25.6710 | 8.7022 | 16917 | 3.6625 | .00151745 | 2070.308 | 341083.50 |
| **660** | **435600** | **287496000** | **25.6905** | **8.7066** | **16956** | **3.6636** | **.00151515** | **2073.449** | **342119.44** |
| 661 | 436921 | 288804781 | 25.7099 | 8.7110 | 16994 | 3.6647 | .00151286 | 2076.591 | 343156.95 |
| 662 | 438244 | 290117528 | 25.7294 | 8.7154 | 17033 | 3.6658 | .00151057 | 2079.733 | 344196.03 |
| 663 | 439569 | 291434247 | 25.7488 | 8.7198 | 17071 | 3.6669 | .00150830 | 2082.874 | 345236.69 |
| 664 | 440896 | 292754944 | 25.7682 | 8.7241 | 17110 | 3.6680 | .00150602 | 2086.016 | 346278.91 |
| 665 | 442225 | 294079625 | 25.7876 | 8.7285 | 17149 | 3.6691 | .00150376 | 2089.157 | 347322.70 |
| 666 | 443556 | 295408296 | 25.8070 | 8.7329 | 17187 | 3.6702 | .00150150 | 2092.299 | 348368.07 |
| 667 | 444889 | 296740963 | 25.8263 | 8.7373 | 17226 | 3.6713 | .00149925 | 2095.441 | 349415.00 |
| 668 | 446224 | 298077632 | 25.8457 | 8.7416 | 17265 | 3.6724 | .00149701 | 2098.582 | 350463.51 |
| 669 | 447561 | 299418309 | 25.8650 | 8.7460 | 17304 | 3.6735 | .00149477 | 2101.724 | 351513.59 |
| **670** | **448900** | **300763000** | **25.8844** | **8.7503** | **17343** | **3.6746** | **.00149254** | **2104.865** | **352565.24** |
| 671 | 450241 | 302111711 | 25.9037 | 8.7547 | 17381 | 3.6757 | .00149031 | 2108.007 | 353618.45 |
| 672 | 451584 | 303464448 | 25.9230 | 8.7590 | 17420 | 3.6768 | .00148810 | 2111.148 | 354673.24 |
| 673 | 452929 | 304821217 | 25.9422 | 8.7634 | 17459 | 3.6779 | .00148588 | 2114.290 | 355729.60 |
| 674 | 454276 | 306182024 | 25.9615 | 8.7677 | 17498 | 3.6790 | .00148368 | 2117.432 | 356787.54 |
| 675 | 455625 | 307546875 | 25.9808 | 8.7721 | 17537 | 3.6801 | .00148148 | 2120.573 | 357847.04 |
| 676 | 456976 | 308915776 | 26.0000 | 8.7764 | 17576 | 3.6812 | .00147929 | 2123.715 | 358908.11 |
| 677 | 458329 | 310288733 | 26.0192 | 8.7807 | 17615 | 3.6823 | .00147710 | 2126.856 | 359970.75 |
| 678 | 459684 | 311665752 | 26.0384 | 8.7850 | 17654 | 3.6834 | .00147493 | 2129.998 | 361034.97 |
| 679 | 461041 | 313046839 | 26.0576 | 8.7893 | 17693 | 3.6845 | .00147275 | 2133.140 | 362100.75 |
| **680** | **462400** | **314432000** | **26.0768** | **8.7937** | **17732** | **3.6856** | **.00147059** | **2136.281** | **363168.11** |
| 681 | 463761 | 315821241 | 26.0960 | 8.7980 | 17771 | 3.6866 | .00146843 | 2139.423 | 364237.04 |
| 682 | 465124 | 317214568 | 26.1151 | 8.8023 | 17810 | 3.6877 | .00146628 | 2142.564 | 365307.54 |
| 683 | 466489 | 318611987 | 26.1343 | 8.8066 | 17850 | 3.6888 | .00146413 | 2145.706 | 366379.60 |
| 684 | 467856 | 320013504 | 26.1534 | 8.8109 | 17889 | 3.6899 | .00146199 | 2148.848 | 367453.24 |
| 685 | 469225 | 321419125 | 26.1725 | 8.8152 | 17928 | 3.6909 | .00145985 | 2151.989 | 368528.45 |
| 686 | 470596 | 322828856 | 26.1916 | 8.8194 | 17967 | 3.6920 | .00145773 | 2155.131 | 369605.23 |
| 687 | 471969 | 324242703 | 26.2107 | 8.8237 | 18007 | 3.6931 | .00145560 | 2158.272 | 370683.59 |
| 688 | 473344 | 325660672 | 26.2298 | 8.8280 | 18046 | 3.6942 | .00145349 | 2161.414 | 371763.51 |
| 689 | 474721 | 327082769 | 26.2488 | 8.8323 | 18085 | 3.6953 | .00145138 | 2164.556 | 372845.00 |
| **690** | **476100** | **328509000** | **26.2679** | **8.8366** | **18125** | **3.6963** | **.00144928** | **2167.697** | **373928.07** |
| 691 | 477481 | 329939371 | 26.2869 | 8.8408 | 18164 | 3.6974 | .00144718 | 2170.839 | 375012.70 |
| 692 | 478864 | 331373888 | 26.3059 | 8.8451 | 18204 | 3.6985 | .00144509 | 2173.980 | 376098.91 |
| 693 | 480249 | 332812557 | 26.3249 | 8.8493 | 18243 | 3.6995 | .00144300 | 2177.122 | 377186.68 |
| 694 | 481636 | 334255384 | 26.3439 | 8.8536 | 18283 | 3.7006 | .00144092 | 2180.263 | 378276.03 |
| 695 | 483025 | 335702375 | 26.3629 | 8.8578 | 18322 | 3.7016 | .00143885 | 2183.405 | 379366.95 |
| 696 | 484416 | 337153536 | 26.3818 | 8.8621 | 18362 | 3.7027 | .00143678 | 2186.547 | 380459.44 |
| 697 | 485809 | 338608873 | 26.4008 | 8.8663 | 18401 | 3.7038 | .00143472 | 2189.688 | 381553.50 |
| 698 | 487204 | 340068392 | 26.4197 | 8.8706 | 18441 | 3.7049 | .00143266 | 2192.830 | 382649.13 |
| 699 | 488601 | 341532099 | 26.4386 | 8.8748 | 18480 | 3.7059 | .00143062 | 2195.971 | 383746.33 |
| **700** | **490000** | **343000000** | **26.4575** | **8.8790** | **18520** | **3.7070** | **.00142857** | **2199.113** | **384845.10** |
| 701 | 491401 | 344472101 | 26.4764 | 8.8833 | 18560 | 3.7080 | .00142653 | 2202.255 | 385945.44 |
| 702 | 492804 | 345948408 | 26.4953 | 8.8875 | 18600 | 3.7091 | .00142450 | 2205.396 | 387047.36 |
| 703 | 494209 | 347428927 | 26.5141 | 8.8917 | 18640 | 3.7101 | .00142248 | 2208.538 | 388150.84 |
| 704 | 495616 | 348913664 | 26.5330 | 8.8959 | 18679 | 3.7112 | .00142045 | 2211.679 | 389255.90 |
| 705 | 497025 | 350402625 | 26.5518 | 8.9001 | 18719 | 3.7123 | .00141844 | 2214.821 | 390362.52 |
| 706 | 498436 | 351895816 | 26.5707 | 8.9043 | 18759 | 3.7133 | .00141643 | 2217.963 | 391470.72 |
| 707 | 499849 | 353393243 | 26.5895 | 8.9085 | 18799 | 3.7144 | .00141443 | 2221.104 | 392580.49 |
| 708 | 501264 | 354894912 | 26.6083 | 8.9127 | 18839 | 3.7154 | .00141243 | 2224.246 | 393691.82 |
| 709 | 502681 | 356400829 | 26.6271 | 8.9169 | 18879 | 3.7165 | .00141044 | 2227.387 | 394804.73 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **710** | **504100** | **357911000** | **26.6458** | **8.9211** | **18919** | **3.7175** | **.00140845** | **2230.529** | **395919.21** |
| 711 | 505521 | 359425431 | 26.6646 | 8.9253 | 18959 | 3.7185 | .00140647 | 2233.670 | 397035.26 |
| 712 | 506944 | 360944128 | 26.6833 | 8.9295 | 18999 | 3.7196 | .00140449 | 2236.812 | 398152.89 |
| 713 | 508369 | 362467097 | 26.7021 | 8.9337 | 19039 | 3.7206 | .00140252 | 2239.954 | 399272.08 |
| 714 | 509796 | 363994344 | 26.7208 | 8.9378 | 19079 | 3.7217 | .00140056 | 2243.095 | 400392.84 |
| 715 | 511225 | 365525875 | 26.7395 | 8.9420 | 19119 | 3.7227 | .00139860 | 2246.237 | 401515.18 |
| 716 | 512656 | 367061696 | 26.7582 | 8.9462 | 19159 | 3.7238 | .00139665 | 2249.378 | 402639.08 |
| 717 | 514089 | 368601813 | 26.7769 | 8.9503 | 19199 | 3.7248 | .00139470 | 2252.520 | 403764.56 |
| 718 | 515524 | 370146232 | 26.7955 | 8.9545 | 19239 | 3.7258 | .00139276 | 2255.662 | 404891.60 |
| 719 | 516961 | 371694959 | 26.8142 | 8.9587 | 19280 | 3.7269 | .00139082 | 2258.803 | 406020.22 |
| **720** | **518400** | **373248000** | **26.8328** | **8.9628** | **19320** | **3.7279** | **.00138889** | **2261.945** | **407150.41** |
| 721 | 519841 | 374805361 | 26.8514 | 8.9670 | 19360 | 3.7290 | .00138696 | 2265.086 | 408282.17 |
| 722 | 521284 | 376367048 | 26.8701 | 8.9711 | 19400 | 3.7300 | .00138504 | 2268.228 | 409415.50 |
| 723 | 522729 | 377933067 | 26.8887 | 8.9752 | 19440 | 3.7310 | .00138313 | 2271.370 | 410550.40 |
| 724 | 524176 | 379503424 | 26.9072 | 8.9794 | 19481 | 3.7321 | .00138122 | 2274.511 | 411686.87 |
| 725 | 525625 | 381078125 | 26.9258 | 8.9835 | 19521 | 3.7331 | .00137931 | 2277.653 | 412824.91 |
| 726 | 527076 | 382657176 | 26.9444 | 8.9876 | 19562 | 3.7341 | .00137741 | 2280.794 | 413964.52 |
| 727 | 528529 | 384240583 | 26.9629 | 8.9918 | 19602 | 3.7351 | .00137552 | 2283.936 | 415105.71 |
| 728 | 529984 | 385828352 | 26.9815 | 8.9959 | 19643 | 3.7362 | .00137363 | 2287.078 | 416248.46 |
| 729 | 531441 | 387420489 | 27.0000 | 9.0000 | 19683 | 3.7372 | .00137174 | 2290.219 | 417392.79 |
| **730** | **532900** | **389017000** | **27.0185** | **9.0041** | **19724** | **3.7382** | **.00136986** | **2293.361** | **418538.68** |
| 731 | 534361 | 390617891 | 27.0370 | 9.0082 | 19764 | 3.7392 | .00136799 | 2296.502 | 419686.15 |
| 732 | 535824 | 392223168 | 27.0555 | 9.0123 | 19805 | 3.7403 | .00136612 | 2299.644 | 420835.19 |
| 733 | 537289 | 393832837 | 27.0740 | 9.0164 | 19845 | 3.7413 | .00136426 | 2302.785 | 421985.79 |
| 734 | 538756 | 395446904 | 27.0924 | 9.0205 | 19886 | 3.7423 | .00136240 | 2305.927 | 423137.97 |
| 735 | 540225 | 397065375 | 27.1109 | 9.0246 | 19927 | 3.7433 | .00136054 | 2309.069 | 424291.72 |
| 736 | 541696 | 398688256 | 27.1293 | 9.0287 | 19967 | 3.7443 | .00135870 | 2312.210 | 425447.04 |
| 737 | 543169 | 400315553 | 27.1477 | 9.0328 | 20008 | 3.7454 | .00135685 | 2315.352 | 426603.94 |
| 738 | 544644 | 401947272 | 27.1662 | 9.0369 | 20049 | 3.7464 | .00135501 | 2318.493 | 427762.40 |
| 739 | 546121 | 403583419 | 27.1846 | 9.0410 | 20090 | 3.7474 | .00135318 | 2321.635 | 428922.43 |
| **740** | **547600** | **405224000** | **27.2029** | **9.0450** | **20130** | **3.7484** | **.00135135** | **2324.777** | **430084.03** |
| 741 | 549081 | 406869021 | 27.2213 | 9.0491 | 20171 | 3.7494 | .00134953 | 2327.918 | 431247.21 |
| 742 | 550564 | 408518488 | 27.2397 | 9.0532 | 20212 | 3.7504 | .00134771 | 2331.060 | 432411.95 |
| 743 | 552049 | 410172407 | 27.2580 | 9.0572 | 20253 | 3.7514 | .00134590 | 2334.201 | 433578.27 |
| 744 | 553536 | 411830784 | 27.2764 | 9.0613 | 20294 | 3.7524 | .00134409 | 2337.343 | 434746.16 |
| 745 | 555025 | 413493625 | 27.2947 | 9.0654 | 20335 | 3.7534 | .00134228 | 2340.485 | 435915.62 |
| 746 | 556516 | 415160936 | 27.3130 | 9.0694 | 20376 | 3.7545 | .00134048 | 2343.626 | 437086.64 |
| 747 | 558009 | 416832723 | 27.3313 | 9.0735 | 20417 | 3.7555 | .00133869 | 2346.768 | 438259.24 |
| 748 | 559504 | 418508992 | 27.3496 | 9.0775 | 20458 | 3.7565 | .00133690 | 2349.909 | 439433.41 |
| 749 | 561001 | 420189749 | 27.3679 | 9.0816 | 20499 | 3.7575 | .00133511 | 2353.051 | 440609.16 |
| **750** | **562500** | **421875000** | **27.3861** | **9.0856** | **20540** | **3.7585** | **.00133333** | **2356.193** | **441786.47** |
| 751 | 564001 | 423564751 | 27.4044 | 9.0896 | 20581 | 3.7595 | .00133156 | 2359.334 | 442965.35 |
| 752 | 565504 | 425259008 | 27.4226 | 9.0937 | 20622 | 3.7605 | .00132979 | 2362.476 | 444145.80 |
| 753 | 567009 | 426957777 | 27.4408 | 9.0977 | 20663 | 3.7615 | .00132802 | 2365.617 | 445327.83 |
| 754 | 568516 | 428661064 | 27.4591 | 9.1017 | 20704 | 3.7625 | .00132626 | 2368.759 | 446511.42 |
| 755 | 570025 | 430368875 | 27.4773 | 9.1057 | 20745 | 3.7635 | .00132450 | 2371.900 | 447696.59 |
| 756 | 571536 | 432081216 | 27.4955 | 9.1098 | 20787 | 3.7645 | .00132275 | 2375.042 | 448883.32 |
| 757 | 573049 | 433798093 | 27.5136 | 9.1138 | 20828 | 3.7655 | .00132100 | 2378.184 | 450071.63 |
| 758 | 574564 | 435519512 | 27.5318 | 9.1178 | 20869 | 3.7665 | .00131926 | 2381.325 | 451261.51 |
| 759 | 576081 | 437245479 | 27.5500 | 9.1218 | 20910 | 3.7675 | .00131752 | 2384.467 | 452452.96 |
| **760** | **577600** | **438976000** | **27.5681** | **9.1258** | **20952** | **3.7685** | **.00131579** | **2387.608** | **453645.93** |
| 761 | 579121 | 440711081 | 27.5862 | 9.1298 | 20993 | 3.7694 | .00131406 | 2390.750 | 454840.57 |
| 762 | 580644 | 442450728 | 27.6043 | 9.1338 | 21035 | 3.7704 | .00131234 | 2393.892 | 456036.73 |
| 763 | 582169 | 444194947 | 27.6225 | 9.1378 | 21076 | 3.7714 | .00131062 | 2397.033 | 457234.46 |
| 764 | 583696 | 445943744 | 27.6405 | 9.1418 | 21117 | 3.7724 | .00130890 | 2400.175 | 458433.77 |
| 765 | 585225 | 447697125 | 27.6586 | 9.1458 | 21159 | 3.7734 | .00130719 | 2403.316 | 459634.64 |
| 766 | 586756 | 449455096 | 27.6767 | 9.1498 | 21200 | 3.7744 | .00130548 | 2406.458 | 460837.08 |
| 767 | 588289 | 451217663 | 27.6948 | 9.1537 | 21242 | 3.7754 | .00130378 | 2409.600 | 462041.10 |
| 768 | 589824 | 452984832 | 27.7128 | 9.1577 | 21283 | 3.7764 | .00130208 | 2412.741 | 463246.69 |
| 769 | 591361 | 454756609 | 27.7308 | 9.1617 | 21325 | 3.7774 | .00130039 | 2415.883 | 464453.84 |
| **770** | **592900** | **456533000** | **27.7489** | **9.1657** | **21367** | **3.7784** | **.00129870** | **2419.024** | **465662.57** |
| 771 | 594441 | 458314011 | 27.7669 | 9.1696 | 21408 | 3.7793 | .00129702 | 2422.166 | 466872.87 |
| 772 | 595984 | 460099648 | 27.7849 | 9.1736 | 21450 | 3.7803 | .00129534 | 2425.307 | 468084.74 |
| 773 | 597529 | 461889917 | 27.8029 | 9.1775 | 21492 | 3.7813 | .00129366 | 2428.449 | 469298.18 |
| 774 | 599076 | 463684824 | 27.8209 | 9.1815 | 21533 | 3.7822 | .00129199 | 2431.591 | 470513.19 |
| 775 | 600625 | 465484375 | 27.8388 | 9.1855 | 21575 | 3.7832 | .00129032 | 2434.732 | 471729.77 |
| 776 | 602176 | 467288576 | 27.8568 | 9.1894 | 21617 | 3.7842 | .00128866 | 2437.874 | 472947.92 |
| 777 | 603729 | 469097433 | 27.8747 | 9.1933 | 21658 | 3.7852 | .00128700 | 2441.015 | 474167.65 |
| 778 | 605284 | 470910952 | 27.8927 | 9.1973 | 21700 | 3.7861 | .00128535 | 2444.157 | 475388.94 |
| 779 | 606841 | 472729139 | 27.9106 | 9.2012 | 21742 | 3.7871 | .00128370 | 2447.299 | 476611.81 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **780** | **608400** | **474552000** | **27.9285** | **9.2052** | **21784** | **3.7881** | **.00128205** | **2450.440** | **477836.24** |
| 781 | 609961 | 476379541 | 27.9464 | 9.2091 | 21826 | 3.7890 | .00128041 | 2453.582 | 479062.25 |
| 782 | 611524 | 478211768 | 27.9643 | 9.2130 | 21868 | 3.7900 | .00127877 | 2456.723 | 480289.83 |
| 783 | 613089 | 480048687 | 27.9821 | 9.2170 | 21910 | 3.7910 | .00127714 | 2459.865 | 481518.97 |
| 784 | 614656 | 481890304 | 28.0000 | 9.2209 | 21952 | 3.7920 | .00127551 | 2463.007 | 482749.69 |
| 785 | 616225 | 483736625 | 28.0179 | 9.2248 | 21994 | 3.7929 | .00127389 | 2466.148 | 483981.98 |
| 786 | 617796 | 485587656 | 28.0357 | 9.2287 | 22036 | 3.7939 | .00127226 | 2469.290 | 485215.84 |
| 787 | 619369 | 487443403 | 28.0535 | 9.2326 | 22078 | 3.7949 | .00127065 | 2472.431 | 486451.28 |
| 788 | 620944 | 489303872 | 28.0713 | 9.2365 | 22120 | 3.7959 | .00126904 | 2475.573 | 487688.28 |
| 789 | 622521 | 491169069 | 28.0891 | 9.2404 | 22162 | 3.7969 | .00126743 | 2478.715 | 488926.85 |
| **790** | **624100** | **493039000** | **28.1069** | **9.2443** | **22205** | **3.7978** | **.00126582** | **2481.856** | **490166.99** |
| 791 | 625681 | 494913671 | 28.1247 | 9.2482 | 22247 | 3.7987 | .00126422 | 2484.998 | 491408.71 |
| 792 | 627264 | 496793088 | 28.1425 | 9.2521 | 22289 | 3.7997 | .00126263 | 2488.139 | 492651.99 |
| 793 | 628849 | 498677257 | 28.1603 | 9.2560 | 22331 | 3.8006 | .00126103 | 2491.281 | 493896.85 |
| 794 | 630436 | 500566184 | 28.1780 | 9.2599 | 22373 | 3.8016 | .00125945 | 2494.422 | 495143.28 |
| 795 | 632025 | 502459875 | 28.1957 | 9.2638 | 22416 | 3.8025 | .00125786 | 2497.564 | 496391.27 |
| 796 | 633616 | 504358336 | 28.2135 | 9.2677 | 22458 | 3.8035 | .00125628 | 2500.706 | 497640.84 |
| 797 | 635209 | 506261573 | 28.2312 | 9.2716 | 22500 | 3.8044 | .00125471 | 2503.847 | 498891.98 |
| 798 | 636804 | 508169592 | 28.2489 | 9.2754 | 22543 | 3.8054 | .00125313 | 2506.989 | 500144.69 |
| 799 | 638401 | 510082399 | 28.2666 | 9.2793 | 22585 | 3.8064 | .00125156 | 2510.130 | 501398.97 |
| **800** | **640000** | **512000000** | **28.2843** | **9.2832** | **22627** | **3.8073** | **.00125000** | **2513.272** | **502654.82** |
| 801 | 641601 | 513922401 | 28.3019 | 9.2870 | 22670 | 3.8083 | .00124844 | 2516.414 | 503912.25 |
| 802 | 643204 | 515849608 | 28.3196 | 9.2909 | 22712 | 3.8092 | .00124688 | 2519.555 | 505171.24 |
| 803 | 644809 | 517781627 | 28.3373 | 9.2948 | 22755 | 3.8102 | .00124533 | 2522.697 | 506431.80 |
| 804 | 646416 | 519718464 | 28.3549 | 9.2986 | 22797 | 3.8111 | .00124378 | 2525.838 | 507693.94 |
| 805 | 648025 | 521660125 | 28.3725 | 9.3025 | 22840 | 3.8121 | .00124224 | 2528.980 | 508957.64 |
| 806 | 649636 | 523606616 | 28.3901 | 9.3063 | 22883 | 3.8130 | .00124069 | 2532.122 | 510222.92 |
| 807 | 651249 | 525557943 | 28.4077 | 9.3102 | 22925 | 3.8139 | .00123916 | 2535.263 | 511489.77 |
| 808 | 652864 | 527514112 | 28.4253 | 9.3140 | 22968 | 3.8149 | .00123762 | 2538.405 | 512758.19 |
| 809 | 654481 | 529475129 | 28.4429 | 9.3179 | 23010 | 3.8158 | .00123609 | 2541.546 | 514028.18 |
| **810** | **656100** | **531441000** | **28.4605** | **9.3217** | **23053** | **3.8168** | **.00123457** | **2544.688** | **515299.74** |
| 811 | 657721 | 533411731 | 28.4781 | 9.3255 | 23096 | 3.8177 | .00123305 | 2547.829 | 516572.87 |
| 812 | 659344 | 535387328 | 28.4956 | 9.3294 | 23138 | 3.8186 | .00123153 | 2550.971 | 517847.57 |
| 813 | 660969 | 537367797 | 28.5132 | 9.3332 | 23181 | 3.8196 | .00123001 | 2554.113 | 519123.84 |
| 814 | 662596 | 539353144 | 28.5307 | 9.3370 | 23224 | 3.8205 | .00122850 | 2557.254 | 520401.68 |
| 815 | 664225 | 541343375 | 28.5482 | 9.3408 | 23267 | 3.8215 | .00122699 | 2560.396 | 521681.10 |
| 816 | 665856 | 543338496 | 28.5657 | 9.3447 | 23310 | 3.8224 | .00122549 | 2563.537 | 522962.08 |
| 817 | 667489 | 545338513 | 28.5832 | 9.3485 | 23352 | 3.8234 | .00122399 | 2566.679 | 524244.63 |
| 818 | 669124 | 547343432 | 28.6007 | 9.3523 | 23395 | 3.8243 | .00122249 | 2569.821 | 525528.76 |
| 819 | 670761 | 549353259 | 28.6182 | 9.3561 | 23438 | 3.8252 | .00122100 | 2572.962 | 526814.46 |
| **820** | **672400** | **551368000** | **28.6356** | **9.3599** | **23481** | **3.8262** | **.00121951** | **2576.104** | **528101.73** |
| 821 | 674041 | 553387661 | 28.6531 | 9.3637 | 23524 | 3.8271 | .00121803 | 2579.245 | 529390.56 |
| 822 | 675684 | 555412248 | 28.6705 | 9.3675 | 23567 | 3.8280 | .00121655 | 2582.387 | 530680.97 |
| 823 | 677329 | 557441767 | 28.6880 | 9.3713 | 23610 | 3.8290 | .00121507 | 2585.529 | 531972.95 |
| 824 | 678976 | 559476224 | 28.7054 | 9.3751 | 23653 | 3.8299 | .00121359 | 2588.670 | 533266.50 |
| 825 | 680625 | 561515625 | 28.7228 | 9.3789 | 23696 | 3.8308 | .00121212 | 2591.812 | 534561.62 |
| 826 | 682276 | 563559976 | 28.7402 | 9.3827 | 23740 | 3.8317 | .00121065 | 2594.953 | 535858.32 |
| 827 | 683929 | 565609283 | 28.7576 | 9.3865 | 23783 | 3.8327 | .00120919 | 2598.095 | 537156.58 |
| 828 | 685584 | 567663552 | 28.7750 | 9.3902 | 23826 | 3.8336 | .00120773 | 2601.237 | 538456.41 |
| 829 | 687241 | 569722789 | 28.7924 | 9.3940 | 23869 | 3.8345 | .00120627 | 2604.378 | 539757.82 |
| **830** | **688900** | **571787000** | **28.8097** | **9.3978** | **23912** | **3.8355** | **.00120482** | **2607.520** | **541060.79** |
| 831 | 690561 | 573856191 | 28.8271 | 9.4016 | 23955 | 3.8364 | .00120337 | 2610.661 | 542365.34 |
| 832 | 692224 | 575930368 | 28.8444 | 9.4053 | 23999 | 3.8373 | .00120192 | 2613.803 | 543671.46 |
| 833 | 693889 | 578009537 | 28.8617 | 9.4091 | 24042 | 3.8382 | .00120048 | 2616.944 | 544979.15 |
| 834 | 695556 | 580093704 | 28.8791 | 9.4129 | 24085 | 3.8391 | .00119904 | 2620.086 | 546288.40 |
| 835 | 697225 | 582182875 | 28.8964 | 9.4166 | 24128 | 3.8401 | .00119760 | 2623.228 | 547599.23 |
| 836 | 698896 | 584277056 | 28.9137 | 9.4204 | 24172 | 3.8410 | .00119617 | 2626.369 | 548911.63 |
| 837 | 700569 | 586376253 | 28.9310 | 9.4241 | 24215 | 3.8419 | .00119474 | 2629.511 | 550225.61 |
| 838 | 702244 | 588480472 | 28.9482 | 9.4279 | 24259 | 3.8428 | .00119332 | 2632.652 | 551541.15 |
| 839 | 703921 | 590589719 | 28.9655 | 9.4316 | 24302 | 3.8437 | .00119190 | 2635.794 | 552858.26 |
| **840** | **705600** | **592704000** | **28.9828** | **9.4354** | **24346** | **3.8446** | **.00119048** | **2638.936** | **554176.94** |
| 841 | 707281 | 594823321 | 29.0000 | 9.4391 | 24389 | 3.8456 | .00118906 | 2642.077 | 555497.20 |
| 842 | 708964 | 596947688 | 29.0172 | 9.4429 | 24432 | 3.8465 | .00118765 | 2645.219 | 556819.02 |
| 843 | 710649 | 599077107 | 29.0345 | 9.4466 | 24476 | 3.8474 | .00118624 | 2648.360 | 558142.42 |
| 844 | 712336 | 601211584 | 29.0517 | 9.4503 | 24520 | 3.8483 | .00118483 | 2651.502 | 559467.39 |
| 845 | 714025 | 603351125 | 29.0689 | 9.4541 | 24563 | 3.8492 | .00118343 | 2654.644 | 560793.92 |
| 846 | 715716 | 605495736 | 29.0861 | 9.4578 | 24607 | 3.8501 | .00118203 | 2657.785 | 562122.03 |
| 847 | 717409 | 607645423 | 29.1033 | 9.4615 | 24650 | 3.8510 | .00118064 | 2660.927 | 563451.71 |
| 848 | 719104 | 609800192 | 29.1204 | 9.4652 | 24694 | 3.8519 | .00117925 | 2664.068 | 564782.96 |
| 849 | 720801 | 611960049 | 29.1376 | 9.4690 | 24738 | 3.8528 | .00117786 | 2667.210 | 566115.78 |

**Table 4.—Properties of Numbers**—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **850** | **722500** | **614125000** | **29.1548** | **9.4727** | **24782** | **3.8538** | **.00117647** | **2670.352** | **567450.17** |
| 851 | 724201 | 616295051 | 29.1719 | 9.4764 | 24825 | 3.8547 | .00117509 | 2673.493 | 568786.14 |
| 852 | 725904 | 618470208 | 29.1890 | 9.4801 | 24869 | 3.8556 | .00117371 | 2676.635 | 570123.67 |
| 853 | 727609 | 620650477 | 29.2062 | 9.4838 | 24913 | 3.8565 | .00117233 | 2679.776 | 571462.77 |
| 854 | 729316 | 622835864 | 29.2233 | 9.4875 | 24957 | 3.8574 | .00117096 | 2682.918 | 572803.45 |
| 855 | 731025 | 625026375 | 29.2404 | 9.4912 | 25000 | 3.8582 | .00116959 | 2686.059 | 574145.69 |
| 856 | 732736 | 627222016 | 29.2575 | 9.4949 | 25044 | 3.8592 | .00116822 | 2689.201 | 575489.51 |
| 857 | 734449 | 629422793 | 29.2746 | 9.4986 | 25088 | 3.8601 | .00116686 | 2692.343 | 576834.90 |
| 858 | 736164 | 631628712 | 29.2916 | 9.5023 | 25132 | 3.8610 | .00116550 | 2695.484 | 578181.85 |
| 859 | 737881 | 633839779 | 29.3087 | 9.5060 | 25176 | 3.8619 | .00116414 | 2698.626 | 579530.38 |
| **860** | **739600** | **636056000** | **29.3258** | **9.5097** | **25220** | **3.8628** | **.00116279** | **2701.767** | **580880.48** |
| 861 | 741321 | 638277381 | 29.3428 | 9.5134 | 25264 | 3.8637 | .00116144 | 2704.909 | 582232.15 |
| 862 | 743044 | 640503928 | 29.3598 | 9.5171 | 25308 | 3.8646 | .00116009 | 2708.051 | 583585.39 |
| 863 | 744769 | 642735647 | 29.3769 | 9.5207 | 25352 | 3.8655 | .00115875 | 2711.192 | 584940.20 |
| 864 | 746496 | 644972544 | 29.3939 | 9.5244 | 25396 | 3.8664 | .00115741 | 2714.334 | 586296.59 |
| 865 | 748225 | 647214625 | 29.4109 | 9.5281 | 25440 | 3.8673 | .00115607 | 2717.475 | 587654.54 |
| 866 | 749956 | 649461896 | 29.4279 | 9.5317 | 25485 | 3.8682 | .00115473 | 2720.617 | 589014.07 |
| 867 | 751689 | 651714363 | 29.4449 | 9.5354 | 25529 | 3.8691 | .00115340 | 2723.759 | 590375.16 |
| 868 | 753424 | 653972032 | 29.4618 | 9.5391 | 25573 | 3.8700 | .00115207 | 2726.900 | 591737.83 |
| 869 | 755161 | 656234909 | 29.4788 | 9.5427 | 25617 | 3.8708 | .00115075 | 2730.042 | 593102.06 |
| **870** | **756900** | **658503000** | **29.4958** | **9.5464** | **25661** | **3.8717** | **.00114943** | **2733.183** | **594467.87** |
| 871 | 758641 | 660776311 | 29.5127 | 9.5501 | 25706 | 3.8726 | .00114811 | 2736.325 | 595835.25 |
| 872 | 760384 | 663054848 | 29.5296 | 9.5537 | 25750 | 3.8735 | .00114679 | 2739.466 | 597204.20 |
| 873 | 762129 | 665338617 | 29.5466 | 9.5574 | 25794 | 3.8744 | .00114548 | 2742.608 | 598574.72 |
| 874 | 763876 | 667627624 | 29.5635 | 9.5610 | 25839 | 3.8753 | .00114416 | 2745.750 | 599946.81 |
| 875 | 765625 | 669921875 | 29.5804 | 9.5647 | 25883 | 3.8762 | .00114286 | 2748.891 | 601320.47 |
| 876 | 767376 | 672221376 | 29.5973 | 9.5683 | 25927 | 3.8771 | .00114155 | 2752.033 | 602695.70 |
| 877 | 769129 | 674526133 | 29.6142 | 9.5719 | 25972 | 3.8780 | .00114025 | 2755.174 | 604072.50 |
| 878 | 770884 | 676836152 | 29.6311 | 9.5756 | 26016 | 3.8789 | .00113895 | 2758.316 | 605450.88 |
| 879 | 772641 | 679151439 | 29.6479 | 9.5792 | 26061 | 3.8797 | .00113766 | 2761.458 | 606830.82 |
| **880** | **774400** | **681472000** | **29.6648** | **9.5828** | **26105** | **3.8806** | **.00113636** | **2764.599** | **608212.34** |
| 881 | 776161 | 683797841 | 29.6816 | 9.5865 | 26150 | 3.8815 | .00113507 | 2767.741 | 609595.42 |
| 882 | 777924 | 686128968 | 29.6985 | 9.5901 | 26194 | 3.8823 | .00113379 | 2770.882 | 610980.08 |
| 883 | 779689 | 688465387 | 29.7153 | 9.5937 | 26239 | 3.8832 | .00113250 | 2774.024 | 612366.31 |
| 884 | 781456 | 690807104 | 29.7321 | 9.5973 | 26283 | 3.8841 | .00113122 | 2777.166 | 613754.11 |
| 885 | 783225 | 693154125 | 29.7489 | 9.6010 | 26328 | 3.8850 | .00112994 | 2780.307 | 615143.48 |
| 886 | 784996 | 695506456 | 29.7658 | 9.6046 | 26373 | 3.8859 | .00112867 | 2783.449 | 616534.42 |
| 887 | 786769 | 697864103 | 29.7825 | 9.6082 | 26417 | 3.8868 | .00112740 | 2786.590 | 617926.93 |
| 888 | 788544 | 700227072 | 29.7993 | 9.6118 | 26462 | 3.8877 | .00112613 | 2789.732 | 619321.01 |
| 889 | 790321 | 702595369 | 29.8161 | 9.6154 | 26507 | 3.8885 | .00112486 | 2792.874 | 620716.66 |
| **890** | **792100** | **704969000** | **29.8329** | **9.6190** | **26551** | **3.8894** | **.00112360** | **2796.015** | **622113.89** |
| 891 | 793881 | 707347971 | 29.8496 | 9.6226 | 26596 | 3.8902 | .00112233 | 2799.157 | 623512.68 |
| 892 | 795664 | 709732288 | 29.8664 | 9.6262 | 26641 | 3.8911 | .00112108 | 2802.298 | 624913.04 |
| 893 | 797449 | 712121957 | 29.8831 | 9.6298 | 26686 | 3.8920 | .00111982 | 2805.440 | 626314.98 |
| 894 | 799236 | 714516984 | 29.8998 | 9.6334 | 26730 | 3.8929 | .00111857 | 2808.581 | 627718.49 |
| 895 | 801025 | 716917375 | 29.9166 | 9.6370 | 26775 | 3.8937 | .00111732 | 2811.723 | 629123.56 |
| 896 | 802816 | 719323136 | 29.9333 | 9.6406 | 26820 | 3.8946 | .00111607 | 2814.865 | 630530.21 |
| 897 | 804609 | 721734273 | 29.9500 | 9.6442 | 26865 | 3.8955 | .00111483 | 2818.006 | 631938.43 |
| 898 | 806404 | 724150792 | 29.9666 | 9.6477 | 26910 | 3.8963 | .00111359 | 2821.148 | 633348.22 |
| 899 | 808201 | 726572699 | 29.9833 | 9.6513 | 26955 | 3.8972 | .00111235 | 2824.289 | 634759.58 |
| **900** | **810000** | **729000000** | **30.0000** | **9.6549** | **27000** | **3.8981** | **.00111111** | **2827.431** | **636172.51** |
| 901 | 811801 | 731432701 | 30.0167 | 9.6585 | 27045 | 3.8989 | .00110988 | 2830.573 | 637587.01 |
| 902 | 813604 | 733870808 | 30.0333 | 9.6620 | 27090 | 3.8998 | .00110865 | 2833.714 | 639003.09 |
| 903 | 815409 | 736314327 | 30.0500 | 9.6656 | 27135 | 3.9007 | .00110742 | 2836.856 | 640420.73 |
| 904 | 817216 | 738763264 | 30.0666 | 9.6692 | 27180 | 3.9015 | .00110619 | 2839.997 | 641839.95 |
| 905 | 819025 | 741217625 | 30.0832 | 9.6727 | 27225 | 3.9024 | .00110497 | 2843.139 | 643260.73 |
| 906 | 820836 | 743677416 | 30.0998 | 9.6763 | 27270 | 3.9032 | .00110375 | 2846.281 | 644683.09 |
| 907 | 822649 | 746142643 | 30.1164 | 9.6799 | 27316 | 3.9041 | .00110254 | 2849.422 | 646107.01 |
| 908 | 824464 | 748613312 | 30.1330 | 9.6834 | 27361 | 3.9050 | .00110132 | 2852.564 | 647532.51 |
| 909 | 826281 | 751089429 | 30.1496 | 9.6870 | 27406 | 3.9059 | .00110011 | 2855.705 | 648959.58 |
| **910** | **828100** | **753571000** | **30.1662** | **9.6905** | **27451** | **3.9067** | **.00109890** | **2858.847** | **650388.22** |
| 911 | 829921 | 756058031 | 30.1828 | 9.6941 | 27497 | 3.9076 | .00109769 | 2861.988 | 651818.43 |
| 912 | 831744 | 758550528 | 30.1993 | 9.6976 | 27542 | 3.9084 | .00109649 | 2865.130 | 653250.21 |
| 913 | 833569 | 761048497 | 30.2159 | 9.7012 | 27587 | 3.9093 | .00109529 | 2868.272 | 654683.56 |
| 914 | 835396 | 763551944 | 30.2324 | 9.7047 | 27632 | 3.9101 | .00109409 | 2871.413 | 656118.48 |
| 915 | 837225 | 766060875 | 30.2490 | 9.7082 | 27678 | 3.9110 | .00109290 | 2874.555 | 657554.98 |
| 916 | 839056 | 768575296 | 30.2655 | 9.7118 | 27723 | 3.9118 | .00109170 | 2877.696 | 658993.04 |
| 917 | 840889 | 771095213 | 30.2820 | 9.7153 | 27769 | 3.9127 | .00109051 | 2880.838 | 660432.68 |
| 918 | 842724 | 773620632 | 30.2985 | 9.7188 | 27814 | 3.9135 | .00108932 | 2883.980 | 661873.88 |
| 919 | 844561 | 776151559 | 30.3150 | 9.7224 | 27859 | 3.9144 | .00108814 | 2887.121 | 663316.66 |

Table 4.—Properties of Numbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circle ($N$ = Diam.) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **920** | **846400** | **778688000** | **30.3315** | **9.7259** | **27905** | **3.9153** | **.00108696** | **2890.263** | **664761.01** |
| 921 | 848241 | 781229961 | 30.3480 | 9.7294 | 27950 | 3.9161 | .00108578 | 2893.404 | 666206.92 |
| 922 | 850084 | 783777448 | 30.3645 | 9.7329 | 27996 | 3.9169 | .00108460 | 2896.546 | 667654.41 |
| 923 | 851929 | 786330467 | 30.3809 | 9.7364 | 28042 | 3.9178 | .00108342 | 2899.688 | 669103.47 |
| 924 | 853776 | 788889024 | 30.3974 | 9.7400 | 28087 | 3.9186 | .00108225 | 2902.829 | 670554.10 |
| 925 | 855625 | 791453125 | 30.4138 | 9.7435 | 28133 | 3.9194 | .00108108 | 2905.971 | 672006.30 |
| 926 | 857476 | 794022776 | 30.4302 | 9.7470 | 28179 | 3.9203 | .00107991 | 2909.112 | 673460.08 |
| 927 | 859329 | 796597983 | 30.4467 | 9.7505 | 28224 | 3.9212 | .00107875 | 2912.254 | 674915.42 |
| 928 | 861184 | 799178752 | 30.4631 | 9.7540 | 28270 | 3.9220 | .00107759 | 2915.396 | 676372.33 |
| 929 | 863041 | 801765089 | 30.4795 | 9.7575 | 28315 | 3.9229 | .00107643 | 2918.537 | 677830.82 |
| **930** | **864900** | **804357000** | **30.4959** | **9.7610** | **28361** | **3.9237** | **.00107527** | **2921.679** | **679290.87** |
| 931 | 866761 | 806954491 | 30.5123 | 9.7645 | 28407 | 3.9246 | .00107411 | 2924.820 | 680752.50 |
| 932 | 868624 | 809557568 | 30.5287 | 9.7680 | 28453 | 3.9254 | .00107296 | 2927.962 | 682215.69 |
| 933 | 870489 | 812166237 | 30.5450 | 9.7715 | 28499 | 3.9262 | .00107181 | 2931.103 | 683680.46 |
| 934 | 872356 | 814780504 | 30.5614 | 9.7750 | 28544 | 3.9271 | .00107066 | 2934.245 | 685146.80 |
| 935 | 874225 | 817400375 | 30.5778 | 9.7785 | 28590 | 3.9279 | .00106952 | 2937.387 | 686614.71 |
| 936 | 876096 | 820025856 | 30.5941 | 9.7819 | 28636 | 3.9288 | .00106838 | 2940.528 | 688084.19 |
| 937 | 877969 | 822656953 | 30.6105 | 9.7854 | 28682 | 3.9296 | .00106724 | 2943.670 | 689555.24 |
| 938 | 879844 | 825293672 | 30.6268 | 9.7889 | 28728 | 3.9304 | .00106610 | 2946.811 | 691027.86 |
| 939 | 881721 | 827936019 | 30.6431 | 9.7924 | 28774 | 3.9313 | .00106496 | 2949.953 | 692502.05 |
| **940** | **883600** | **830584000** | **30.6594** | **9.7959** | **28820** | **3.9321** | **.00106383** | **2953.095** | **693977.82** |
| 941 | 885481 | 833237621 | 30.6757 | 9.7993 | 28866 | 3.9329 | .00106270 | 2956.236 | 695455.15 |
| 942 | 887364 | 835896888 | 30.6920 | 9.8028 | 28912 | 3.9338 | .00106157 | 2959.378 | 696934.06 |
| 943 | 889249 | 838561807 | 30.7083 | 9.8063 | 28958 | 3.9346 | .00106045 | 2962.519 | 698414.53 |
| 944 | 891136 | 841232384 | 30.7246 | 9.8097 | 29004 | 3.9354 | .00105932 | 2965.661 | 699896.58 |
| 945 | 893025 | 843908625 | 30.7409 | 9.8132 | 29050 | 3.9363 | .00105820 | 2968.803 | 701380.19 |
| 946 | 894916 | 846590536 | 30.7571 | 9.8167 | 29096 | 3.9371 | .00105708 | 2971.944 | 702865.38 |
| 947 | 896809 | 849278123 | 30.7734 | 9.8201 | 29142 | 3.9379 | .00105597 | 2975.086 | 704352.14 |
| 948 | 898704 | 851971392 | 30.7896 | 9.8236 | 29189 | 3.9388 | .00105485 | 2978.227 | 705840.47 |
| 949 | 900601 | 854670349 | 30.8058 | 9.8270 | 29235 | 3.9396 | .00105374 | 2981.369 | 707330.37 |
| **950** | **902500** | **857375000** | **30.8221** | **9.8305** | **29281** | **3.9404** | **.00105263** | **2984.511** | **708821.84** |
| 951 | 904401 | 860085351 | 30.8383 | 9.8339 | 29327 | 3.9413 | .00105152 | 2987.652 | 710314.88 |
| 952 | 906304 | 862801408 | 30.8545 | 9.8374 | 29374 | 3.9421 | .00105042 | 2990.794 | 711809.50 |
| 953 | 908209 | 865523177 | 30.8707 | 9.8408 | 29420 | 3.9429 | .00104932 | 2993.935 | 713305.68 |
| 954 | 910116 | 868250664 | 30.8869 | 9.8443 | 29466 | 3.9438 | .00104822 | 2997.077 | 714803.43 |
| 955 | 912025 | 870983875 | 30.9031 | 9.8477 | 29513 | 3.9446 | .00104712 | 3000.218 | 716302.76 |
| 956 | 913936 | 873722816 | 30.9192 | 9.8511 | 29559 | 3.9454 | .00104603 | 3003.360 | 717803.66 |
| 957 | 915849 | 876467493 | 30.9354 | 9.8546 | 29605 | 3.9462 | .00104493 | 3006.502 | 719306.12 |
| 958 | 917764 | 879217912 | 30.9516 | 9.8580 | 29652 | 3.9471 | .00104384 | 3009.643 | 720810.16 |
| 959 | 919681 | 881974079 | 30.9677 | 9.8614 | 29698 | 3.9479 | .00104275 | 3012.785 | 722315.77 |
| **960** | **921600** | **884736000** | **30.9839** | **9.8648** | **29745** | **3.9487** | **.00104167** | **3015.926** | **723822.95** |
| 961 | 923521 | 887503681 | 31.0000 | 9.8683 | 29791 | 3.9495 | .00104058 | 3019.068 | 725331.70 |
| 962 | 925444 | 890277128 | 31.0161 | 9.8717 | 29838 | 3.9503 | .00103950 | 3022.210 | 726842.02 |
| 963 | 927369 | 893056347 | 31.0322 | 9.8751 | 29884 | 3.9512 | .00103842 | 3025.351 | 728353.91 |
| 964 | 929296 | 895841344 | 31.0483 | 9.8785 | 29931 | 3.9520 | .00103734 | 3028.493 | 729867.37 |
| 965 | 931225 | 898632125 | 31.0644 | 9.8819 | 29977 | 3.9528 | .00103627 | 3031.634 | 731382.40 |
| 966 | 933156 | 901428696 | 31.0805 | 9.8854 | 30024 | 3.9536 | .00103520 | 3034.776 | 732899.01 |
| 967 | 935089 | 904231063 | 31.0966 | 9.8888 | 30070 | 3.9544 | .00103413 | 3037.918 | 734417.18 |
| 968 | 937024 | 907039232 | 31.1127 | 9.8922 | 30117 | 3.9553 | .00103306 | 3041.059 | 735936.93 |
| 969 | 938961 | 909853209 | 31.1288 | 9.8956 | 30164 | 3.9561 | .00103199 | 3044.201 | 737458.24 |
| **970** | **940900** | **912673000** | **31.1448** | **9.8990** | **30210** | **3.9569** | **.00103093** | **3047.342** | **738981.13** |
| 971 | 942841 | 915498611 | 31.1609 | 9.9024 | 30257 | 3.9577 | .00102987 | 3050.484 | 740505.59 |
| 972 | 944784 | 918330048 | 31.1769 | 9.9058 | 30304 | 3.9585 | .00102881 | 3053.625 | 742031.62 |
| 973 | 946729 | 921167317 | 31.1929 | 9.9092 | 30351 | 3.9593 | .00102775 | 3056.767 | 743559.22 |
| 974 | 948676 | 924010424 | 31.2090 | 9.9126 | 30398 | 3.9602 | .00102669 | 3059.909 | 745088.39 |
| 975 | 950625 | 926859375 | 31.2250 | 9.9160 | 30444 | 3.9610 | .00102564 | 3063.050 | 746619.13 |
| 976 | 952576 | 929714176 | 31.2410 | 9.9194 | 30491 | 3.9618 | .00102459 | 3066.192 | 748151.44 |
| 977 | 954529 | 932574833 | 31.2570 | 9.9227 | 30538 | 3.9626 | .00102354 | 3069.333 | 749685.32 |
| 978 | 956484 | 935441352 | 31.2730 | 9.9261 | 30585 | 3.9634 | .00102249 | 3072.475 | 751220.78 |
| 979 | 958441 | 938313739 | 31.2890 | 9.9295 | 30632 | 3.9642 | .00102145 | 3075.617 | 752757.80 |
| **980** | **960400** | **941192000** | **31.3050** | **9.9329** | **30679** | **3.9650** | **.00102041** | **3078.758** | **754296.40** |
| 981 | 962361 | 944076141 | 31.3209 | 9.9363 | 30726 | 3.9658 | .00101937 | 3081.900 | 755836.59 |
| 982 | 964324 | 946966168 | 31.3369 | 9.9396 | 30773 | 3.9666 | .00101833 | 3085.041 | 757378.30 |
| 983 | 966289 | 949862087 | 31.3528 | 9.9430 | 30820 | 3.9674 | .00101729 | 3088.183 | 758921.61 |
| 984 | 968256 | 952763904 | 31.3688 | 9.9464 | 30867 | 3.9682 | .00101626 | 3091.325 | 760466.48 |
| 985 | 970225 | 955671625 | 31.3847 | 9.9497 | 30914 | 3.9691 | .00101523 | 3094.466 | 762012.93 |
| 986 | 972196 | 958585256 | 31.4006 | 9.9531 | 30961 | 3.9699 | .00101420 | 3097.608 | 763560.95 |
| 987 | 974169 | 961504803 | 31.4166 | 9.9565 | 31008 | 3.9707 | .00101317 | 3100.749 | 765110.54 |
| 988 | 976144 | 964430272 | 31.4325 | 9.9598 | 31055 | 3.9715 | .00101215 | 3103.891 | 766661.70 |
| 989 | 978121 | 967361669 | 31.4484 | 9.9632 | 31102 | 3.9723 | .00101112 | 3107.033 | 768214.44 |

Table 4.—Properties of Nnmbers—*Continued*

| $N$ | $N^2$ | $N^3$ | $\sqrt{N}$ | $\sqrt[3]{N}$ | $N^{3/2}$ | $\sqrt[5]{N}$ | $\frac{1}{N}$ | Circie ($N = D$) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Circum. | Area |
| **990** | **980100** | **970299000** | **31.4643** | **9.9666** | **31150** | **3.9731** | **.00101010** | **3110.174** | **769768.74** |
| 991 | 982081 | 973242271 | 31.4802 | 9.9699 | 31197 | 3.9739 | .00100908 | 3113.316 | 771324.61 |
| 992 | 984064 | 976191488 | 31.4960 | 9.9733 | 31244 | 3.9747 | .00100806 | 3116.457 | 772882.06 |
| 993 | 986049 | 979146657 | 31.5119 | 9.9766 | 31291 | 3.9755 | .00100705 | 3119.599 | 774441.07 |
| 994 | 988036 | 982107784 | 31.5278 | 9.9800 | 31339 | 3.9763 | .00100604 | 3122.740 | 776001.66 |
| 995 | 990025 | 985074875 | 31.5436 | 9.9833 | 31386 | 3.9771 | .00100503 | 3125.882 | 777563.82 |
| 996 | 992016 | 988047936 | 31.5595 | 9.9866 | 31433 | 3.9779 | .00100402 | 3129.024 | 779127.54 |
| 997 | 994009 | 991026973 | 31.5753 | 9.9900 | 31480 | 3.9787 | .00100301 | 3132.165 | 780692.84 |
| 998 | 996004 | 994011992 | 31.5911 | 9.9933 | 31528 | 3.9795 | .00100200 | 3135.307 | 782259.71 |
| 999 | 998001 | 997002999 | 31.6070 | 9.9967 | 31575 | 3.9803 | .00100100 | 3138.448 | 783828.15 |
| **1000** | **1000000** | **1000000000** | **31.6228** | **10.0000** | **31623** | **3.9811** | **.00100000** | **3141.593** | **785398.16** |

## CIRCULAR ARCS, CHORDS, SEGMENTS AND SECTORS

**Notation.**—$A$ = central angle, degrees; $C$ = chord of arc; $c$ = chord of $1/2$ arc; $D$ = diameter; $G$ = area of segment; $H$ = rise of arc or height of segment; $L$ = length of arc; $R$ = radius; $S$ = area of sector.

$$L \text{ (exact)} = 2\pi RN/360 = 0.017453RN$$

$$L \text{ (approx.)} = (8c - C)/3 = \{(2c \times 10H)/(60D - 27A)\} + 2c$$
$$= \{(\sqrt{C^2 + 4H^2} \times 10H^2)/(15C^2 + 33H^2)\} + 2c$$

$$C = 2\sqrt{c^2 - H^2} = \sqrt{D^2 - (D - 2H)^2} = (8c - 3L) = 2\sqrt{R^2 - (R - H)^2} = 2\sqrt{(D - H) \times H}$$

$$C/2 = \sqrt{H \times (D - H)}; \quad c = 1/2\sqrt{C^2 - 4H^2} = \sqrt{D \times H} = (3L + C)/8$$

$$D = c^2/H = (1/4\,C^2 + H^2)/H$$

$$H = c^2/D = 1/2\,(D - \sqrt{D^2 - C^2}) \text{ when } H < R; = 1/2\,(D + \sqrt{D^2 - C^2}) = \sqrt{c^2 - 1/4\,C^2} \text{ when } H > R$$

$$G = S - 1/2\,C\sqrt{R^2 - 1/4\,C^2} \text{ when } G < \text{semi-circle}; = S + 1/2\,C\sqrt{R^2 - 1/4\,C^2} \text{ when } S > \text{semi-circle}$$
$$= S - (C/2)(R - H) = 1/2\,\{(L - C)(D/2) + CH\}$$

$$S = 1/2\,LR = \pi AR^2/360 = (\pi R^2/360) \times 2[\sin^{-1}\{(C/2)/R\}]; \quad R = (C^2+4H^2)/8H$$

Table 5. Circular Arcs, Chords, and Segments. Radius = 1

| Central Angle in Degrees | Arc | | Chord, Length $C$ | $\frac{H}{C}$ | Area oi Seg-ment* $G$ | Central Angle in Degrees | Arc | | Chord, Length $C$ | $\frac{H}{C}$ | Area of Seg-ment* $G$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Length $L$ | Rise $H$ | | | | | Length $L$ | Rise $H$ | | | |
| 1 | 0.0175 | 0.0000 | 0.0175 | 0.0022 | 0.00000 | 31 | 0.5411 | 0.0364 | 0.5345 | 0.0680 | 0.01301 |
| 2 | .0349 | .0002 | .0349 | .0044 | .00000 | 32 | .5585 | .0387 | .5513 | .0703 | .01429 |
| 3 | .0524 | .0003 | .0524 | .0066 | .00001 | 33 | .5760 | .0412 | .5680 | .0725 | .01566 |
| 4 | .0698 | .0006 | .0698 | .0087 | .00003 | 34 | .5934 | .0437 | .5847 | .0747 | .01711 |
| 5 | .0873 | .0010 | .0872 | .0109 | .00006 | 35 | .6109 | .0463 | .6014 | .0770 | .01864 |
| 6 | .1047 | .0014 | .1047 | .0131 | .00010 | 36 | .6283 | .0489 | .6180 | .0792 | .02027 |
| 7 | .1222 | .0019 | .1221 | .0153 | .00015 | 37 | .6458 | .0517 | .6346 | .0814 | .02198 |
| 8 | .1396 | .0024 | .1395 | .0175 | .00023 | 38 | .6632 | .0545 | .6511 | .0837 | .02378 |
| 9 | .1571 | .0031 | .1569 | .0196 | .00032 | 39 | .6807 | .0574 | .6676 | .0859 | .02568 |
| **10** | **.1745** | **.0038** | **.1743** | **.0218** | **.00044** | **40** | **.6981** | **.0603** | **.6840** | **.0882** | **.02767** |
| 11 | .1920 | .0046 | .1917 | .0240 | .00059 | 41 | .7156 | .0633 | .7004 | .0904 | .02976 |
| 12 | .2094 | .0055 | .2091 | .0262 | .00076 | 42 | .7330 | .0664 | .7167 | .0927 | .03195 |
| 13 | .2269 | .0064 | .2264 | .0284 | .00097 | 43 | .7505 | .0696 | .7330 | .0949 | .03425 |
| 14 | .2443 | .0075 | .2437 | .0306 | .00121 | 44 | .7679 | .0728 | .7492 | .0972 | .03664 |
| 15 | .2618 | .0086 | .2611 | .0328 | .00149 | 45 | .7854 | .0761 | .7654 | .0995 | .03915 |
| 16 | .2793 | .0097 | .2783 | .0350 | .00181 | 46 | .8029 | .0795 | .7815 | .1017 | .04176 |
| 17 | .2967 | .0110 | .2956 | .0372 | .00217 | 47 | .8203 | .0829 | .7975 | .1040 | .04448 |
| 18 | .3142 | .0123 | .3129 | .0394 | .00257 | 48 | .8378 | .0865 | .8135 | .1063 | .04731 |
| 19 | .3316 | .0137 | .3301 | .0415 | .00302 | 49 | .8552 | .0900 | .8294 | .1086 | .05025 |
| **20** | **.3491** | **.0152** | **.3473** | **.0437** | **.00352** | **50** | **8727** | **.0937** | **.8452** | **.1108** | **.05331** |
| 21 | .3665 | .0167 | .3645 | .0459 | .00408 | 51 | .8901 | .0974 | .8610 | .1131 | .05649 |
| 22 | .3840 | .0184 | .3816 | .0481 | .00468 | 52 | .9076 | .1012 | .8767 | .1154 | .05978 |
| 23 | .4014 | .0201 | .3987 | .0503 | .00535 | 53 | .9250 | .1051 | .8924 | .1177 | .06319 |
| 24 | .4189 | .0219 | .4158 | .0526 | .00607 | 54 | .9425 | .1090 | .9080 | .1200 | .06673 |
| 25 | .4363 | .0237 | .4329 | .0548 | .00686 | 55 | .9599 | .1130 | .9235 | .1223 | .07039 |
| 26 | .4538 | .0256 | .4499 | .0570 | .00771 | 56 | .9774 | .1171 | .9389 | .1247 | .07417 |
| 27 | .4712 | .0276 | .4669 | .0592 | .00862 | 57 | .9848 | .1212 | .9543 | .1270 | .07808 |
| 28 | .4887 | .0297 | .4838 | .0614 | .00961 | 58 | 1.0123 | .1254 | .9696 | .1293 | .08212 |
| 29 | .5061 | .0319 | .5008 | .0636 | .01067 | 59 | 1.0297 | .1296 | .9848 | .1316 | .08629 |
| **30** | **.5236** | **.0341** | **.5176** | **.0658** | **.01180** | **60** | **1.0472** | **.1340** | **1.0000** | **.1340** | **.09059** |

*Area of segment of any radius = factor $G$ × square of radius

**Table 5.—Circular Arcs, Chords and Segments—*Continued***

| Central Angle in Degrees | Arc | | Chord, Length *C* | $\frac{H}{C}$ | Area of Segment * *G* | Central Angle in Degrees | Arc | | Chord, Length *C* | $\frac{H}{C}$ | Area of Segment * *G* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Length *L* | Rise *H* | | | | | Length *L* | Rise *H* | | | |
| 61 | 1.0647 | 0.1384 | 1.015 | 0.1363 | 0.09502 | 121 | 2.1118 | 0.5076 | 1.741 | 0.2916 | 0.62734 |
| 62 | 1.0821 | .1428 | 1.030 | .1387 | .09958 | 122 | 2.1293 | .5152 | 1.749 | .2945 | .64063 |
| 63 | 1.0996 | .1474 | 1.045 | .1410 | .10428 | 123 | 2.1468 | .5228 | 1.758 | .2975 | .65404 |
| 64 | 1.1170 | .1520 | 1.060 | .1434 | .10911 | 124 | 2.1642 | .5305 | 1.766 | .3004 | .66759 |
| 65 | 1.1345 | .1566 | 1.075 | .1457 | .11408 | 125 | 2.1817 | .5383 | 1.774 | .3034 | .68125 |
| 66 | 1.1519 | .1613 | 1.089 | .1481 | .11919 | 126 | 2.1991 | .5460 | 1.782 | .3064 | .69505 |
| 67 | 1.1694 | .1661 | 1.104 | .1505 | .12443 | 127 | 2.2166 | .5538 | 1.790 | .3094 | .70897 |
| 68 | 1.1868 | .1710 | 1.118 | .1529 | .12982 | 128 | 2.2340 | .5616 | 1.798 | .3124 | .72301 |
| 69 | 1.2043 | .1759 | 1.133 | .1553 | .13535 | 129 | 2.2515 | .5695 | 1.805 | .3155 | .73716 |
| **70** | **1.2217** | **.1808** | **1.147** | **.1576** | **.14102** | **130** | **2.2689** | **.5774** | **1.813** | **.3185** | **.75144** |
| 71 | 1.2392 | .1859 | 1.161 | .1601 | .14683 | 131 | 2.2864 | .5853 | 1.820 | .3216 | .76584 |
| 72 | 1.2566 | .1910 | 1.176 | .1625 | .15279 | 132 | 2.3038 | .5933 | 1.827 | .3247 | .78034 |
| 73 | 1.2741 | .1961 | 1.190 | .1649 | .15889 | 133 | 2.3213 | .6013 | 1.834 | .3278 | .79497 |
| 74 | 1.2915 | .2014 | 1.204 | .1673 | .16514 | 134 | 2.3387 | .6093 | 1.841 | .3309 | .80970 |
| 75 | 1.3090 | .2066 | 1.218 | .1697 | .17154 | 135 | 2.3562 | .6173 | 1.848 | .3341 | .82454 |
| 76 | 1.3265 | .2120 | 1.231 | .1722 | .17808 | 136 | 2.3736 | .6254 | 1.854 | .3373 | .83949 |
| 77 | 1.3439 | .2174 | 1.245 | .1746 | .18477 | 137 | 2.3911 | .6335 | 1.861 | .3404 | .85455 |
| 78 | 1.3614 | .2229 | 1.259 | .1771 | .19160 | 138 | 2.4086 | .6416 | 1.867 | .3436 | .86971 |
| 79 | 1.3788 | 2284 | 1.272 | .1795 | .19859 | 139 | 2.4260 | .6498 | 1.873 | .3469 | .88497 |
| **80** | **1.3963** | **.2340** | **1.286** | **.1820** | **.20573** | **140** | **2.4435** | **.6580** | **1.879** | **.3501** | **.90034** |
| 81 | 1.4137 | .2396 | 1.299 | .1845 | .21301 | 141 | 2.4609 | .6662 | 1.885 | .3534 | .91580 |
| 82 | 1.4312 | .2453 | 1.312 | .1869 | .22045 | 142 | 2.4784 | .6744 | 1.891 | .3566 | .93135 |
| 83 | 1.4486 | .2510 | 1.325 | .1894 | .22804 | 143 | 2.4958 | .6827 | 1.897 | .3599 | .94700 |
| 84 | 1.4661 | .2569 | 1.338 | .1919 | .23578 | 144 | 2.5133 | .6910 | 1.902 | .3633 | .96274 |
| 85 | 1.4835 | .2627 | 1.351 | .1944 | .24367 | 145 | 2.5307 | .6993 | 1.907 | .3666 | .97858 |
| 86 | 1.5010 | .2686 | 1.364 | .1970 | .25171 | 146 | 2.5482 | .7076 | 1.913 | .3700 | .99449 |
| 87 | 1.5184 | .2746 | 1.377 | .1995 | .25990 | 147 | 2.5656 | .7160 | 1.918 | .3734 | 1.0105 |
| 88 | 1.5359 | .2807 | 1.389 | .2020 | .26825 | 148 | 2.5831 | .7244 | 1.923 | .3768 | 1.0266 |
| 89 | 1.5533 | .2867 | 1.402 | .2046 | .27675 | 149 | 2.6005 | .7328 | 1.927 | .3802 | 1.0428 |
| **90** | **1.5708** | **.2929** | **1.414** | **.2071** | **.28540** | **150** | **2.6180** | **.7412** | **1.932** | **.3837** | **1.0590** |
| 91 | 1.5882 | .2991 | 1.427 | .2097 | .29420 | 151 | 2.6354 | .7496 | 1.936 | .3871 | 1.0753 |
| 92 | 1.6057 | .3053 | 1.439 | .2122 | .30316 | 152 | 2.6529 | .7581 | 1.941 | .3906 | 1.0917 |
| 93 | 1.6232 | .3116 | 1.451 | .2148 | .31226 | 153 | 2.6704 | .7666 | 1.945 | .3942 | 1.1082 |
| 94 | 1.6406 | .3180 | 1.463 | .2174 | .32152 | 154 | 2.6878 | .7750 | 1.949 | .3977 | 1.1247 |
| 95 | 1.6581 | .3244 | 1.475 | .2200 | .33093 | 155 | 2.7053 | .7836 | 1.953 | .4013 | 1.1413 |
| 96 | 1.6755 | .3309 | 1.486 | .2226 | .34050 | 156 | 2.7227 | .7921 | 1.956 | .4049 | 1.1580 |
| 97 | 1.6930 | .3374 | 1.498 | .2252 | .35021 | 157 | 2.7402 | .8006 | 1.960 | .4085 | 1.1747 |
| 98 | 1.7104 | .3439 | 1.509 | .2279 | .36008 | 158 | 2.7576 | .8092 | 1.963 | .4122 | 1.1915 |
| 99 | 1.7279 | .3506 | 1.521 | .2305 | .37009 | 159 | 2.7751 | .8178 | 1.967 | .4158 | 1.2084 |
| **100** | **1.7453** | **.3572** | **1.532** | **.2332** | **.38026** | **160** | **2.7925** | **.8264** | **1.970** | **.4195** | **1.2253** |
| 101 | 1.7628 | .3639 | 1.543 | .2358 | .39058 | 161 | 2.8100 | .8350 | 1.973 | .4233 | 1.2422 |
| 102 | 1.7802 | .3707 | 1.554 | .2385 | .40104 | 162 | 2.8274 | .8436 | 1.975 | .4270 | 1.2592 |
| 103 | 1.7977 | .3775 | 1.565 | .2412 | .41166 | 163 | 2.8449 | .8522 | 1.978 | .4308 | 1.2763 |
| 104 | 1.8151 | .3843 | 1.576 | .2439 | .42242 | 164 | 2.8623 | .8608 | 1.981 | .4346 | 1.2934 |
| 105 | 1.8326 | .3912 | 1.587 | .2466 | .43333 | 165 | 2.8798 | .8695 | 1.983 | .4385 | 1.3105 |
| 106 | 1.8500 | .3982 | 1.597 | .2493 | .44439 | 166 | 2.8972 | .8781 | 1.985 | .4424 | 1.3277 |
| 107 | 1.8675 | .4052 | 1.608 | .2520 | .45560 | 167 | 2.9147 | .8868 | 1.987 | .4463 | 1.3449 |
| 108 | 1.8850 | .4122 | 1.618 | .2548 | .46695 | 168 | 2.9322 | .8955 | 1.989 | .4502 | 1.3621 |
| 109 | 1.9024 | .4193 | 1.628 | .2575 | .47844 | 169 | 2.9496 | .9042 | 1.991 | .4542 | 1.3794 |
| **110** | **1.9199** | **.4264** | **1.638** | **.2603** | **.49008** | **170** | **2.9671** | **.9128** | **1.992** | **.4582** | **1.3967** |
| 111 | 1.9373 | .4336 | 1.648 | .2631 | .50187 | 171 | 2.9845 | .9215 | 1.994 | .4622 | 1.4140 |
| 112 | 1.9548 | .4408 | 1.658 | .2659 | .51379 | 172 | 3.0020 | .9302 | 1.995 | .4663 | 1.4314 |
| 113 | 1.9722 | .4481 | 1.668 | .2687 | .52586 | 173 | 3.0194 | .9390 | 1.996 | .4704 | 1.4488 |
| 114 | 1.9897 | .4554 | 1.677 | .2715 | .53806 | 174 | 3.0369 | .9477 | 1.997 | .4745 | 1.4662 |
| 115 | 2.0071 | .4627 | 1.687 | .2743 | .55041 | 175 | 3.0543 | .9564 | 1.998 | .4786 | 1.4836 |
| 116 | 2.0246 | .4701 | 1.696 | .2772 | .56289 | 176 | 3.0718 | .9651 | 1.9988 | .4828 | 1.5010 |
| 117 | 2.0420 | .4775 | 1.705 | .2800 | .57551 | 177 | 3.0892 | .9738 | 1.9993 | .4871 | 1.5184 |
| 118 | 2.0595 | .4850 | 1.714 | .2829 | .58827 | 178 | 3.1067 | .9825 | 1.9997 | .4913 | 1.5359 |
| 119 | 2.0769 | .4925 | 1.723 | .2858 | .60116 | 179 | 3.1241 | .9913 | 1.9999 | .4957 | 1.5533 |
| **120** | **2.0944** | **.5000** | **1.732** | **.2887** | **.61419** | **180** | **3.1416** | **1.0000** | **2.000** | **.5000** | **1.5708** |

* Area of segment of any radius = factor *G* × square of radius

## Table 6.—Areas of Segments of a Circle

Area = $F \times D^2$ where segment < semicircle; $= \frac{1}{4}\pi D^2 -$ (area of segment of rise $h$). $H$ = rise or height of segment; $D$ = diam. of circle; $F$ = factor corresponding to $H/D$. $h = (D - H)$

| H/D | F | H/D | F | H/D | F | H/D | F | H/D | F | H/D | F |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.000 | 0.00000 | 0.07 | 0.02417 | 0.14 | 0.06683 | 0.21 | 0.11990 | 0.28 | 0.18002 | 0.35 | 0.24498 |
| .001 | .00004 | .071 | .02468 | .141 | .06753 | .211 | .12071 | .281 | .18092 | .351 | .24593 |
| .002 | .00012 | .072 | .02520 | .142 | .06822 | .212 | .12153 | .282 | .18182 | .352 | .24689 |
| .003 | .00022 | .073 | .02571 | .143 | .06892 | .213 | .12235 | .283 | .18272 | .353 | .24784 |
| .004 | .00034 | .074 | .02624 | .144 | .06963 | .214 | .12317 | .284 | .18362 | .354 | .24880 |
| .005 | .00047 | .075 | .02676 | .145 | .07033 | .215 | .12399 | .285 | .18452 | .355 | .24976 |
| .006 | .00062 | .076 | .02729 | .146 | .07103 | .216 | .12481 | .286 | .18542 | .256 | .25071 |
| .007 | .00078 | .077 | .02782 | .147 | .07174 | .217 | .12563 | .287 | .18633 | .357 | .25167 |
| .008 | .00095 | .078 | .02836 | .148 | .07245 | .218 | .12646 | .288 | .18723 | .358 | .25263 |
| .009 | .00113 | .079 | .02889 | .149 | .07316 | .219 | .12729 | .289 | .18814 | .359 | .25359 |
| .01 | .00133 | .08 | .02943 | .15 | .07387 | .22 | .12811 | .29 | .18905 | .36 | .25455 |
| .011 | .00153 | .081 | .02998 | .151 | .07459 | .221 | .12894 | .291 | .18996 | .361 | .25551 |
| .012 | .00175 | .082 | .03053 | .152 | .07531 | .222 | .12977 | .292 | .19086 | .362 | .25647 |
| .013 | .00197 | .083 | .03108 | .153 | .07603 | .223 | .13060 | .293 | .19177 | .363 | .25743 |
| .014 | .00220 | .084 | .03163 | .154 | .07675 | .224 | .13144 | .294 | .19268 | .364 | .25839 |
| .015 | .00244 | .085 | .03219 | .155 | .07747 | .225 | .13227 | .295 | .19360 | .365 | .25936 |
| .016 | .00268 | .086 | .03275 | .156 | .07819 | .226 | .13311 | .296 | .19451 | .366 | .26032 |
| .017 | .00294 | .087 | .03331 | .157 | .07892 | .227 | .13395 | .297 | .19542 | .367 | .26128 |
| .018 | .00320 | .088 | .03387 | .158 | .07965 | .228 | .13478 | .298 | .19634 | .368 | .26225 |
| .019 | .00347 | .089 | .03444 | .159 | .08038 | .229 | .13562 | .299 | .19725 | .369 | .26321 |
| .02 | .00375 | .09 | .03501 | .16 | .08111 | .23 | .13646 | .3 | .19817 | .37 | .26418 |
| .021 | .00403 | .091 | .03559 | .161 | .08185 | .231 | .13731 | .301 | .19908 | .371 | .26514 |
| .022 | .00432 | .092 | .03616 | .162 | .08258 | .232 | .13815 | .302 | .20000 | .372 | .26611 |
| .023 | .00462 | .093 | .03674 | .163 | .08332 | .233 | .13900 | .303 | .20092 | .373 | .26708 |
| .024 | .00492 | .094 | .03732 | .164 | .08406 | .234 | .13984 | .304 | .20184 | .374 | .26805 |
| .025 | .00523 | .095 | .03791 | .165 | .08480 | .235 | .14069 | .305 | .20276 | .375 | .26901 |
| .026 | .00555 | .096 | .03850 | .166 | .08554 | .236 | .14154 | .306 | .20368 | .376 | .26998 |
| .027 | .00587 | .097 | .03909 | .167 | .08629 | .237 | .14239 | .307 | .20460 | .377 | .27095 |
| .028 | .00619 | .098 | .03968 | .168 | .08704 | .238 | .14324 | .308 | .20553 | .378 | .27192 |
| .029 | .00653 | .099 | .04028 | .169 | .08779 | .239 | .14409 | .309 | .20645 | .379 | .27289 |
| .03 | .00687 | .1 | .04087 | .17 | .08854 | .24 | .14494 | .31 | .20738 | .38 | .27386 |
| .031 | .00721 | .101 | .04148 | .171 | .08929 | .241 | .14580 | .311 | .20830 | .381 | .27483 |
| .032 | .00756 | .102 | .04208 | .172 | .09004 | .242 | .14666 | .312 | .20923 | .382 | .27580 |
| .033 | .00791 | .103 | .04269 | .173 | .09080 | .243 | .14751 | .313 | .21015 | .383 | .27678 |
| .034 | .00827 | .104 | .04330 | .174 | .09155 | .244 | .14837 | .314 | .21108 | .384 | .27775 |
| .035 | .00864 | .105 | .04391 | .175 | .09231 | .245 | .14923 | .315 | .21201 | .385 | .27872 |
| .036 | .00901 | .106 | .04452 | .176 | .09307 | .246 | .15009 | .316 | .21294 | .386 | .27969 |
| .037 | .00938 | .107 | .04514 | .177 | .09384 | .247 | .15095 | .317 | .21387 | .387 | .28067 |
| .038 | .00976 | .108 | .04576 | .178 | .09460 | .248 | .15182 | .318 | .21480 | .388 | .28164 |
| .039 | .01015 | .109 | .04638 | .179 | .09537 | .249 | .15268 | .319 | .21573 | .389 | .28262 |
| .04 | .01054 | .11 | .04701 | .18 | .09613 | .25 | .15355 | .32 | .21667 | .39 | .28359 |
| .041 | .01093 | .111 | .04763 | .181 | .09690 | .251 | .15441 | .321 | .21760 | .391 | .28457 |
| .042 | .01133 | .112 | .04826 | .182 | .09767 | .252 | .15528 | .322 | .21853 | .392 | .28554 |
| .043 | .01173 | .113 | .04889 | .183 | .09845 | .253 | .15615 | .323 | .21947 | .393 | .28652 |
| .044 | .01214 | .114 | .04953 | .184 | .09922 | .254 | .15702 | .324 | .22040 | .394 | .28750 |
| .045 | .01255 | .115 | .05016 | .185 | .10000 | .255 | .15789 | .325 | .22134 | .395 | .28848 |
| .046 | .01297 | .116 | .05080 | .186 | .10077 | .256 | .15876 | .326 | .22228 | .396 | .28945 |
| .047 | .01339 | .117 | .05145 | .187 | .10155 | .257 | .15964 | .327 | .22322 | .397 | .29043 |
| .048 | .01382 | .118 | .05209 | .188 | .10233 | .258 | .16051 | .328 | .22415 | .398 | .29141 |
| .049 | .01425 | .119 | .05274 | .189 | .10312 | .259 | .16139 | .329 | .22509 | .399 | .29239 |
| .05 | .01468 | .12 | .05338 | .19 | .10390 | .26 | .16226 | .33 | .22603 | .4 | .29337 |
| .051 | .01512 | .121 | .05404 | .191 | .10469 | .261 | .16314 | .331 | .22697 | .401 | .29435 |
| .052 | .01556 | .122 | .05469 | .192 | .10547 | .262 | .16402 | .332 | .22792 | .402 | .29533 |
| .053 | .01601 | .123 | .05535 | .193 | .10626 | .263 | .16490 | .333 | .22886 | .403 | .29631 |
| .054 | .01646 | .124 | .05600 | .194 | .10705 | .264 | .16578 | .334 | .22980 | .404 | .29729 |
| .055 | .01691 | .125 | .05666 | .195 | .10784 | .265 | .16666 | .335 | .23074 | .405 | .29827 |
| .056 | .01737 | .126 | .05733 | .196 | .10864 | .266 | .16755 | .336 | .23169 | .406 | .29926 |
| .057 | .01783 | .127 | .05799 | .197 | .10943 | .267 | .16843 | .337 | .23263 | .407 | .30024 |
| .058 | .01830 | .128 | .05866 | .198 | .11023 | .268 | .16932 | .338 | .23358 | .408 | .30122 |
| .059 | .01877 | .129 | .05933 | .199 | .11102 | .269 | .17020 | .339 | .23453 | .409 | .30220 |
| .06 | .01924 | .13 | .06000 | .2 | .11182 | .27 | .17109 | .34 | .23547 | .41 | .30319 |
| .061 | .01972 | .131 | .06067 | .201 | .11262 | .271 | .17198 | .341 | .23642 | .411 | .30417 |
| .062 | .02020 | .132 | .06135 | .202 | .11343 | .272 | .17287 | .342 | .23737 | .412 | .30516 |
| .063 | .02068 | .133 | .06203 | .203 | .11423 | .273 | .17376 | .343 | .23832 | .413 | .30614 |
| .064 | .02117 | .134 | .06271 | .204 | .11504 | .274 | .17465 | .344 | .23927 | .414 | .30712 |
| .065 | .02166 | .135 | .06339 | .205 | .11584 | .275 | .17554 | .345 | .24022 | .415 | .30811 |
| .066 | .02215 | .136 | .06407 | .206 | .11665 | .276 | .17644 | .346 | .24117 | .416 | .30910 |
| .067 | .02265 | .137 | .06476 | .207 | .11746 | .277 | .17733 | .347 | .24212 | .417 | .31008 |
| .068 | .02315 | .138 | .06545 | .208 | .11827 | .278 | .17823 | .348 | .24307 | .418 | .31107 |
| .069 | .02366 | .139 | .06614 | .209 | .11908 | .279 | .17912 | .349 | .24403 | .419 | .31205 |

*(Table continued on following page)*

### Table 6.—Area of Segments—*Continued*

| *H/D* | *F* | *H/D* | *F* | *H/D* | *F* | *H/D* | *F* | *H/D* | *F* | *H/D* | *F* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.42 | 0.31304 | 0.435 | 0.32788 | 0.45 | 0.34278 | 0.465 | 0.35773 | 0.48 | 0.37270 | 0.495 | 0.38770 |
| .421 | .31403 | .436 | .32887 | .451 | .34378 | .466 | .35873 | .481 | .37370 | .496 | .38870 |
| .422 | .31502 | .437 | .32987 | .452 | .34477 | .467 | .35972 | .482 | .37470 | .497 | .38970 |
| .423 | .31600 | .438 | .33086 | .453 | .34577 | .468 | .36072 | .483 | .37570 | .498 | .39070 |
| .424 | .31699 | .439 | .33185 | .454 | .34676 | .469 | .36172 | .484 | .37670 | .499 | .39170 |
| .425 | .31798 | .44 | .33284 | .455 | .34776 | .47 | .36272 | .485 | .37770 | .500 | .39270 |
| .426 | .31897 | .441 | .33384 | .456 | .34876 | .471 | .36372 | .486 | .37870 | | |
| .427 | .31996 | .442 | .33483 | .457 | .34975 | .472 | .36471 | .487 | .37970 | | |
| .428 | .32095 | .443 | .33582 | .458 | .35075 | .473 | .36571 | .488 | .38070 | | |
| .429 | .32194 | .444 | .33682 | .459 | .35175 | .474 | .36671 | .489 | .38170 | | |
| .43 | .32293 | .445 | .33781 | .46 | .35274 | .475 | .36771 | .49 | .38270 | | |
| .431 | .32392 | .446 | .33880 | .461 | .35374 | .476 | .36871 | .491 | .38370 | | |
| .432 | .32491 | .447 | .33980 | .462 | .35474 | .477 | .36971 | .492 | .38470 | | |
| .433 | .32590 | .448 | .34079 | .463 | .35573 | .478 | .37071 | .493 | .38570 | | |
| .434 | .32689 | .449 | .34179 | .464 | .35673 | .479 | .37171 | .494 | .38670 | | |

### Table 7.—Values of Degrees and Minutes in Radians

| Deg | Radians | Deg | Radians | Deg | Radians | Deg | Radians | Deg | Radians | Deg | Radians | Min | Radians | Min | Radians |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.01745 | 31 | 0.54105 | 61 | 1.06465 | 91 | 1.58825 | 121 | 2.11185 | 151 | 2.63545 | 1 | 0.00029 | 31 | 0.00902 |
| 2 | .03491 | 32 | .55851 | 62 | 1.08210 | 92 | 1.60570 | 122 | 2.12930 | 152 | 2.65290 | 2 | .00058 | 32 | .00931 |
| 3 | .05236 | 33 | .57596 | 63 | 1.09956 | 93 | 1.62316 | 123 | 2.14676 | 153 | 2.67035 | 3 | .00087 | 33 | .00960 |
| 4 | .06981 | 34 | .59341 | 64 | 1.11701 | 94 | 1.64061 | 124 | 2.16421 | 154 | 2.68781 | 4 | .00116 | 34 | .00989 |
| 5 | .08727 | 35 | .61087 | 65 | 1.13446 | 95 | 1.65806 | 125 | 2.18166 | 155 | 2.70526 | 5 | .00145 | 35 | .01018 |
| 6 | .10472 | 36 | .62832 | 66 | 1.15192 | 96 | 1.67552 | 126 | 2.19912 | 156 | 2.72271 | 6 | .00175 | 36 | .01047 |
| 7 | .12217 | 37 | .64577 | 67 | 1.16937 | 97 | 1.69297 | 127 | 2.21657 | 157 | 2.74017 | 7 | .00204 | 37 | .01076 |
| 8 | .13963 | 38 | .66323 | 68 | 1.18682 | 98 | 1.71042 | 128 | 2.23402 | 158 | 2.75762 | 8 | .00233 | 38 | .01105 |
| 9 | .15708 | 39 | .68068 | 69 | 1.20428 | 99 | 1.72788 | 129 | 2.25148 | 159 | 2.77507 | 9 | .00262 | 39 | .01134 |
| 10 | .17453 | 40 | .69813 | 70 | 1.22173 | 100 | 1.74533 | 130 | 2.26893 | 160 | 2.79253 | 10 | .00291 | 40 | .01164 |
| 11 | .19199 | 41 | .71559 | 71 | 1.23918 | 101 | 1.76278 | 131 | 2.28638 | 161 | 2.80998 | 11 | .00320 | 41 | .01193 |
| 12 | .20944 | 42 | .73304 | 72 | 1.25664 | 102 | 1.78024 | 132 | 2.30384 | 162 | 2.82743 | 12 | .00349 | 42 | .01222 |
| 13 | .22689 | 43 | .75049 | 73 | 1.27409 | 103 | 1.79769 | 133 | 2.32129 | 163 | 2.84489 | 13 | .00378 | 43 | .01251 |
| 14 | .24435 | 44 | .76795 | 74 | 1.29154 | 104 | 1.81514 | 134 | 2.33874 | 164 | 2.86234 | 14 | .00407 | 44 | .01280 |
| 15 | .26180 | 45 | .78540 | 75 | 1.30900 | 105 | 1.83260 | 135 | 2.35620 | 165 | 2.87979 | 15 | .00436 | 45 | .01309 |
| 16 | .27925 | 46 | .80285 | 76 | 1.32645 | 106 | 1.85005 | 136 | 2.37365 | 166 | 2.89725 | 16 | .00465 | 46 | .01338 |
| 17 | .29671 | 47 | .82031 | 77 | 1.34390 | 107 | 1.86750 | 137 | 2.39110 | 167 | 2.91470 | 17 | .00495 | 47 | .01367 |
| 18 | .31416 | 48 | .83776 | 78 | 1.36136 | 108 | 1.88496 | 138 | 2.40856 | 168 | 2.93215 | 18 | .00524 | 48 | .01396 |
| 19 | .33161 | 49 | .85521 | 79 | 1.37881 | 109 | 1.90241 | 139 | 2.42601 | 169 | 2.94961 | 19 | .00553 | 49 | .01425 |
| 20 | .34907 | 50 | .87267 | 80 | 1.39626 | 110 | 1.91986 | 140 | 2.44346 | 170 | 2.96706 | 20 | .00582 | 50 | .01454 |
| 21 | .36652 | 51 | .89012 | 81 | 1.41372 | 111 | 1.93732 | 141 | 2.46092 | 171 | 2.98451 | 21 | .00611 | 51 | .01484 |
| 22 | .38397 | 52 | .90757 | 82 | 1.43117 | 112 | 1.95477 | 142 | 2.47837 | 172 | 3.00197 | 22 | .00640 | 52 | .01513 |
| 23 | .40143 | 53 | .92502 | 83 | 1.44862 | 113 | 1.97222 | 143 | 2.49582 | 173 | 3.01942 | 23 | .00669 | 53 | .01542 |
| 24 | .41888 | 54 | .94248 | 84 | 1.46608 | 114 | 1.98968 | 144 | 2.51328 | 174 | 3.03687 | 24 | .00698 | 54 | .01571 |
| 25 | .43633 | 55 | .95993 | 85 | 1.48353 | 115 | 2.00713 | 145 | 2.53073 | 175 | 3.05433 | 25 | .00727 | 55 | .01600 |
| 26 | .45379 | 56 | .97738 | 86 | 1.50098 | 116 | 2.02458 | 146 | 2.54818 | 176 | 3.07178 | 26 | .00756 | 56 | .01629 |
| 27 | .47124 | 57 | .99484 | 87 | 1.51844 | 117 | 2.04204 | 147 | 2.56564 | 177 | 3.08923 | 27 | .00785 | 57 | .01658 |
| 28 | .48869 | 58 | 1.01299 | 88 | 1.53589 | 118 | 2.05949 | 148 | 2.58309 | 178 | 3.10669 | 28 | .00814 | 58 | .01687 |
| 29 | .50615 | 59 | 1.02974 | 89 | 1.55334 | 119 | 2.07694 | 149 | 2.60054 | 179 | 3.12414 | 29 | .00844 | 59 | .01716 |
| 30 | .52360 | 60 | 1.04720 | 90 | 1.57080 | 120 | 2.09440 | 150 | 2.61800 | 180 | 3.14159 | 30 | .00873 | 60 | .01745 |

### Table 8.—Values of Radians in Degrees

| Rad. | .00 | .01 | .02 | .03 | .04 | .05 | .06 | .07 | .08 | .09 |
|---|---|---|---|---|---|---|---|---|---|---|
| | Deg. | Deg. | Deg. | Deg. | Deg. | Deg. | Deg. | Deg. | Deg. | Deg. |
| 0.0 | 0.0000 | 0.5730 | 1.1459 | 1.7189 | 2.2918 | 2.8648 | 3.4377 | 4.0107 | 4.5837 | 5.1566 |
| .1 | 5.7296 | 6.3025 | 6.8755 | 7.4485 | 8.0214 | 8.5944 | 9.1673 | 9.7403 | 10.3132 | 10.8862 |
| .2 | 11.4591 | 12.0321 | 12.6051 | 13.1780 | 13.7510 | 14.3239 | 14.8969 | 15.4699 | 16.0428 | 16.6158 |
| .3 | 17.1887 | 17.7617 | 18.3346 | 18.9076 | 19.4806 | 20.0535 | 20.6265 | 21.1994 | 21.7724 | 22.3454 |
| .4 | 22.9183 | 23.4913 | 24.0642 | 24.6372 | 25.2101 | 25.7831 | 26.3561 | 26.9290 | 27.5020 | 28.0749 |
| .5 | 28.6479 | 29.2208 | 29.7938 | 30.3668 | 30.9397 | 31.1527 | 32.0856 | 32.6586 | 33.2316 | 33.8045 |
| .6 | 34.3775 | 34.9504 | 35.5234 | 36.0963 | 36.6693 | 37.2423 | 37.8152 | 38.3882 | 38.9611 | 39.5341 |
| .7 | 40.1070 | 40.6800 | 41.2530 | 41.8259 | 42.3989 | 42.9718 | 43.5448 | 44.1178 | 44.6907 | 45.2637 |
| .8 | 45.8366 | 46.4096 | 46.9825 | 47.5555 | 48.1285 | 48.7014 | 49.2744 | 49.8473 | 50.4203 | 50.9932 |
| .9 | 51.5662 | 52.1392 | 52.7121 | 53.2851 | 53.8580 | 54.4310 | 55.0039 | 55.5769 | 56.1499 | 56.7228 |

1 Radian = 57.29578 deg. | 2 Radians = 114.59156 deg. | 3 Radians = 171.88734 deg.

### Table 9.—Decimals of a Degree in Minutes and Seconds

| Decimal | .00 | | .01 | | .02 | | .03 | | .04 | | .05 | | .06 | | .07 | | .08 | | .09 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Min. | Sec. | Min. | Sec. | Min. | Sec | Min. | Sec. | Min. | Sec. | Min. | Sec. | Min. | Sec. | Min. | Sec. | Min. | Sec. | Min. | Sec. |
| 0.0 | 0 | 0 | 0 | 36 | 1 | 12 | 1 | 48 | 2 | 24 | 3 | 0 | 3 | 36 | 4 | 12 | 4 | 48 | 5 | 24 |
| .1 | 6 | 0 | 6 | 36 | 7 | 12 | 7 | 48 | 8 | 24 | 9 | 0 | 9 | 36 | 10 | 12 | 10 | 48 | 11 | 24 |
| .2 | 12 | 0 | 12 | 36 | 13 | 12 | 13 | 48 | 14 | 24 | 15 | 0 | 15 | 36 | 16 | 12 | 16 | 48 | 17 | 24 |
| .3 | 18 | 0 | 18 | 36 | 19 | 12 | 19 | 48 | 20 | 24 | 21 | 0 | 21 | 36 | 22 | 12 | 22 | 48 | 23 | 24 |
| .4 | 24 | 0 | 24 | 36 | 25 | 12 | 25 | 48 | 26 | 24 | 27 | 0 | 27 | 36 | 28 | 12 | 28 | 48 | 29 | 24 |
| .5 | 30 | 0 | 30 | 36 | 31 | 12 | 31 | 48 | 32 | 24 | 33 | 0 | 33 | 36 | 34 | 12 | 24 | 48 | 35 | 24 |
| .6 | 36 | 0 | 36 | 36 | 37 | 12 | 37 | 48 | 38 | 24 | 39 | 0 | 39 | 36 | 40 | 12 | 40 | 48 | 41 | 24 |
| .7 | 42 | 0 | 42 | 36 | 43 | 12 | 43 | 48 | 44 | 24 | 45 | 0 | 45 | 36 | 46 | 12 | 46 | 48 | 47 | 24 |
| .8 | 48 | 0 | 48 | 36 | 49 | 12 | 49 | 48 | 50 | 24 | 51 | 0 | 51 | 36 | 52 | 12 | 52 | 48 | 53 | 24 |
| .9 | 54 | 0 | 54 | 36 | 55 | 12 | 55 | 48 | 56 | 24 | 57 | 0 | 57 | 36 | 58 | 12 | 58 | 48 | 59 | 24 |

**Table 10.—Circumferences and Areas of Circles—Diameters in Feet and Inches ***

Circumferences in feet; areas in square feet

| Diam., feet | Diameter, inches | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
| | Circumference | | | | | | | | | | | |
| 0 | 0.0000 | 0.2618 | 0.5236 | 0.7854 | 1.0472 | 1.3090 | 1.5708 | 1.8326 | 2.0944 | 2.3562 | 2.6180 | 2.8798 |
| 1 | 3.1416 | 3.4034 | 3.6653 | 3.9270 | 4.1877 | 4.4507 | 4.7124 | 4.9741 | 5.2361 | 5.4978 | 5.7595 | 6.0215 |
| 2 | 6.2832 | 6.5450 | 6.8069 | 7.0686 | 7.3293 | 7.5923 | 7.8540 | 8.1157 | 8.3777 | 8.6394 | 8.9011 | 9.1631 |
| 3 | 9.4248 | 9.6866 | 9.9485 | 10.210 | 10.471 | 10.734 | 10.996 | 11.257 | 11.519 | 11.781 | 12.043 | 12.305 |
| 4 | 12.566 | 12.828 | 13.090 | 13.352 | 13.613 | 13.875 | 14.137 | 14.399 | 14.661 | 14.923 | 15.184 | 15.446 |
| 5 | 15.708 | 15.970 | 16.232 | 16.493 | 16.754 | 17.017 | 17.279 | 17.540 | 17.802 | 18.064 | 18.326 | 18.588 |
| 6 | 18.850 | 19.111 | 19.373 | 19.635 | 19.896 | 20.159 | 20.420 | 20.682 | 20.944 | 21.206 | 21.467 | 21.729 |
| 7 | 21.991 | 22.253 | 22.515 | 22.777 | 23.037 | 23.300 | 23.562 | 23.824 | 24.086 | 24.347 | 24.609 | 24.871 |
| 8 | 25.133 | 25.395 | 25.656 | 25.918 | 26.179 | 26.442 | 26.704 | 26.965 | 27.227 | 27.489 | 27.751 | 28.013 |
| 9 | 28.274 | 28.536 | 28.798 | 29.060 | 29.320 | 29.583 | 29.845 | 30.107 | 30.369 | 30.631 | 30.892 | 31.154 |
| 10 | 31.416 | 31.678 | 31.940 | 32.201 | 32.462 | 32.725 | 32.987 | 33.248 | 33.510 | 33.772 | 34.034 | 34.296 |
| 11 | 34.557 | 34.819 | 35.081 | 35.343 | 35.604 | 35.867 | 36.128 | 36.390 | 36.652 | 36.914 | 37.175 | 37.437 |
| 12 | 37.699 | 37.961 | 38.223 | 38.484 | 38.745 | 39.008 | 39.270 | 39.532 | 39.794 | 40.055 | 40.317 | 40.579 |
| 13 | 40.841 | 41.102 | 41.364 | 41.626 | 41.887 | 42.150 | 42.411 | 42.673 | 42.935 | 43.197 | 43.459 | 43.721 |
| 14 | 43.982 | 44.244 | 44.506 | 44.768 | 45.028 | 45.291 | 45.553 | 45.815 | 46.077 | 46.338 | 46.600 | 46.862 |
| 15 | 47.124 | 47.386 | 47.648 | 47.909 | 48.170 | 48.433 | 48.695 | 48.956 | 49.218 | 49.480 | 49.742 | 50.004 |
| 16 | 50.265 | 50.527 | 50.789 | 51.051 | 51.312 | 51.575 | 51.836 | 52.098 | 52.360 | 52.622 | 52.883 | 53.145 |
| 17 | 53.407 | 53.669 | 53.931 | 54.192 | 54.453 | 54.716 | 54.978 | 55.240 | 55.502 | 55.763 | 56.025 | 56.287 |
| 18 | 56.549 | 56.810 | 57.072 | 57.334 | 57.595 | 57.858 | 58.119 | 58.381 | 58.643 | 58.905 | 59.167 | 59.429 |
| 19 | 59.690 | 59.952 | 60.214 | 60.476 | 60.736 | 61.000 | 61.261 | 61.523 | 61.785 | 62.046 | 62.308 | 62.570 |
| 20 | 62.832 | 63.094 | 63.356 | 63.617 | 63.878 | 64.141 | 64.403 | 64.664 | 64.926 | 65.188 | 65.450 | 65.712 |
| 21 | 65.973 | 66.235 | 66.497 | 66.759 | 67.020 | 67.282 | 67.544 | 67.806 | 68.068 | 68.330 | 68.591 | 68.853 |
| 22 | 69.115 | 69.377 | 69.639 | 69.900 | 70.161 | 70.424 | 70.686 | 70.947 | 71.209 | 71.471 | 71.733 | 71.995 |
| 23 | 72.257 | 72.518 | 72.780 | 73.042 | 73.303 | 73.566 | 73.827 | 74.089 | 74.351 | 74.513 | 74.874 | 75.136 |
| 24 | 75.398 | 75.660 | 75.922 | 76.184 | 76.444 | 76.707 | 76.969 | 77.231 | 77.493 | 77.754 | 78.016 | 78.278 |
| 25 | 78.540 | 78.802 | 79.063 | 79.325 | 79.586 | 79.849 | 80.111 | 80.372 | 80.634 | 80.896 | 81.158 | 81.420 |
| 26 | 81.681 | 81.943 | 82.205 | 82.467 | 82.727 | 82.990 | 83.252 | 83.514 | 83.776 | 84.038 | 84 299 | 84.561 |
| 27 | 84.823 | 85.085 | 85.347 | 85.608 | 85.869 | 86.132 | 86.394 | 86.655 | 86.917 | 87.179 | 87.441 | 87.703 |
| 28 | 87.965 | 88.226 | 88.488 | 88.749 | 89.011 | 89.274 | 89.535 | 89.797 | 90.059 | 90.321 | 90.582 | 90.844 |
| 29 | 91.106 | 91.367 | 91.630 | 91.892 | 92.152 | 92.415 | 92.677 | 92.939 | 93.201 | 93.462 | 93.724 | 93.986 |
| 30 | 94.248 | 94.509 | 94.771 | 95.033 | 95.294 | 95.557 | 95.818 | 96.080 | 96.342 | 96.604 | 96.866 | 97.128 |
| | Area | | | | | | | | | | | |
| 0 | 0.0000 | 0.0055 | 0.0218 | 0.0491 | 0.0873 | 0.1364 | 0.1963 | 0.2673 | 0.3491 | 0.4418 | 0.5454 | 0.6600 |
| 1 | 0.7854 | 0.9218 | 1.069 | 1.227 | 1.396 | 1.576 | 1.767 | 1.969 | 2.182 | 2.405 | 2.640 | 2.885 |
| 2 | 3.142 | 3.409 | 3.687 | 3.976 | 4.276 | 4.587 | 4.909 | 5.241 | 5.585 | 5.940 | 6.305 | 6.681 |
| 3 | 7.069 | 7.467 | 7.876 | 8.296 | 8.727 | 9.168 | 9.621 | 10.08 | 10.56 | 11.04 | 11.54 | 12.05 |
| 4 | 12.57 | 13.10 | 13.64 | 14.19 | 14 75 | 15.32 | 15.90 | 16.50 | 17.10 | 17.72 | 18.35 | 18.99 |
| 5 | 19.63 | 20.29 | 20.97 | 21.65 | 22.34 | 23.04 | 23.76 | 24.48 | 25.22 | 25.97 | 26.73 | 27.49 |
| 6 | 28.27 | 29.07 | 29.87 | 30.68 | 31.50 | 32.34 | 33.18 | 34.04 | 34.91 | 35.78 | 36.67 | 37.57 |
| 7 | 38.48 | 39.41 | 40.34 | 41.28 | 42.24 | 43.20 | 44.18 | 45.17 | 46.16 | 47.17 | 48.19 | 49.22 |
| 8 | 50.27 | 51.32 | 52.38 | 53.46 | 54.54 | 55.64 | 56.75 | 57.86 | 58.99 | 60.13 | 61.28 | 62.44 |
| 9 | 63.62 | 64.80 | 66.00 | 67.20 | 68.42 | 69.64 | 70.88 | 72.13 | 73.39 | 74.66 | 75.94 | 77.24 |
| 10 | 78.54 | 79.85 | 81.18 | 82.52 | 83.86 | 85.22 | 86.59 | 87.97 | 89.36 | 90.76 | 92.18 | 93.60 |
| 11 | 95.03 | 96.48 | 97.93 | 99.40 | 100.9 | 102.4 | 103.9 | 105.4 | 106.9 | 108.4 | 110. 0 | 111.5 |
| 12 | 113.1 | 114.7 | 116.3 | 117.9 | 119.5 | 121.1 | 122.7 | 124.4 | 126.0 | 127.7 | 129.4 | 131.0 |
| 13 | 132.7 | 134.4 | 136.2 | 137.9 | 139.6 | 141.4 | 143.1 | 144.9 | 146.7 | 148.5 | 150.3 | 152.1 |
| 14 | 153.9 | 155.8 | 157.6 | 159.5 | 161.4 | 163.2 | 165.1 | 167.0 | 168.9 | 170.9 | 172.8 | 174.8 |
| 15 | 176.7 | 178.7 | 180.7 | 182.7 | 184.7 | 186.7 | 188.7 | 190.7 | 192.8 | 194.8 | 196.9 | 199.0 |
| 16 | 201.1 | 203.2 | 205.3 | 207.4 | 209.5 | 211.7 | 213.8 | 216.0 | 218.2 | 220.4 | 222.5 | 224.8 |
| 17 | 227.0 | 229.2 | 231.4 | 233.7 | 236.0 | 238.2 | 240.5 | 242.8 | 245.1 | 247.5 | 249.8 | 252.1 |
| 18 | 254.5 | 256.8 | 259.2 | 261.6 | 264.0 | 266.4 | 268.8 | 271.2 | 273.7 | 276.1 | 278.6 | 281.0 |
| 19 | 283.5 | 286.0 | 288.5 | 291.0 | 293.6 | 296.1 | 298.7 | 301.2 | 303.8 | 306.4 | 308.9 | 311.5 |
| 20 | 314.2 | 316.8 | 319.4 | 322.1 | 324.7 | 327.4 | 330.1 | 332.7 | 335.4 | 338.2 | 340.9 | 343.6 |
| 21 | 346.4 | 349.1 | 351.9 | 354.7 | 357.4 | 360.2 | 363.1 | 365.9 | 368.7 | 371.5 | 374.4 | 377.2 |
| 22 | 380.1 | 383.0 | 386.0 | 388.8 | 391.7 | 394.7 | 397.6 | 400.5 | 403.5 | 406.5 | 409.5 | 412.5 |
| 23 | 415.5 | 418.5 | 421.5 | 424.6 | 427.6 | 430.7 | 433.7 | 436.8 | 439.9 | 443.0 | 446.1 | 449.2 |
| 24 | 452.4 | 455.5 | 458.7 | 461.9 | 465.0 | 468.2 | 471.4 | 474.6 | 477.9 | 481.1 | 484.3 | 487.6 |
| 25 | 490.9 | 494.1 | 497.4 | 500.7 | 504.0 | 507.4 | 510.7 | 514.0 | 517.4 | 520.8 | 524.1 | 527.5 |
| 26 | 530.9 | 534.3 | 537.7 | 541.2 | 544.6 | 548.1 | 551.6 | 555.0 | 558.5 | 562.0 | 565.5 | 569.0 |
| 27 | 572.6 | 576.1 | 579.6 | 583.2 | 586.8 | 590.3 | 594.0 | 597.5 | 601.2 | 604.8 | 608.4 | 612.1 |
| 28 | 615.8 | 619.4 | 623.1 | 626.8 | 630.5 | 634.2 | 637.9 | 641.7 | 645.4 | 649.2 | 652.9 | 656.7 |
| 29 | 660.5 | 664.3 | 668.1 | 672.0 | 675.8 | 679.6 | 683.5 | 687.3 | 691.2 | 695.1 | 699.0 | 702.9 |
| 30 | 706.9 | 710.8 | 714.7 | 718.7 | 722.6 | 726.6 | 730.6 | 734.6 | 738.6 | 742.6 | 746.7 | 750.7 |

* For circumferences and areas of circles, diameters in inches, see Table 4, pp. 28–25 to 28–41.

**Table 11.—Natural Trigonometrical Functions**

| Deg. | Min. | Sine | Covers | Cosec | Tan | Cotan | Secant | Versin | Cosine | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 0 | 0.00000 | 1.00000 | Infinite | 0.00000 | Infinite | 1.0000 | 0.00000 | 1.00000 | 90 | 0 |
| | 15 | .00436 | .99564 | 229.18 | .00436 | 229.18 | 1.0000 | .00001 | .99999 | | 45 |
| | 30 | .00873 | .99127 | 114.59 | .00873 | 114.59 | 1.0000 | .00004 | .99996 | | 30 |
| | 45 | .01309 | .98691 | 76.397 | .01309 | 76.390 | 1.0001 | .00009 | .99991 | | 15 |
| 1 | 0 | .01745 | .98255 | 57.299 | .01745 | 57.290 | 1.0001 | .00015 | .99985 | 89 | 0 |
| | 15 | .02181 | .97819 | 45.840 | .02182 | 45.829 | 1.0002 | .00024 | .99976 | | 45 |
| | 30 | .02618 | .97382 | 38.202 | .02618 | 38.188 | 1.0003 | .00034 | .99966 | | 30 |
| | 45 | .03054 | .96946 | 32.746 | .03055 | 32.730 | 1.0005 | .00047 | .99953 | | 15 |
| 2 | 0 | .03490 | .96510 | 28.654 | .03492 | 28.636 | 1.0006 | .00061 | .99939 | 88 | 0 |
| | 15 | .03926 | .96074 | 25.471 | .03929 | 25.452 | 1.0008 | .00077 | .99923 | | 45 |
| | 30 | .04362 | .95638 | 22.926 | .04366 | 22.904 | 1.0009 | .00095 | .99905 | | 30 |
| | 45 | .04798 | .95202 | 20.843 | .04803 | 20.819 | 1.0011 | .00115 | .99885 | | 15 |
| 3 | 0 | .05234 | .94766 | 19.107 | .05241 | 19.081 | 1.0014 | .00137 | .99863 | 87 | 0 |
| | 15 | .05669 | .94331 | 17.639 | .05678 | 17.611 | 1.0016 | .00161 | .99839 | | 45 |
| | 30 | .06105 | .93895 | 16.380 | .06116 | 16.350 | 1.0019 | .00187 | .99813 | | 30 |
| | 45 | .06540 | .93460 | 15.290 | .06554 | 15.257 | 1.0021 | .00214 | .99786 | | 15 |
| 4 | 0 | .06976 | .93024 | 14.336 | .06993 | 14.301 | 1.0024 | .00244 | .99756 | 86 | 0 |
| | 15 | .07411 | .92589 | 13.494 | .07431 | 13.457 | 1.0028 | .00275 | .99725 | | 45 |
| | 30 | .07846 | .92154 | 12.745 | .07870 | 12.706 | 1.0031 | .00308 | .99692 | | 30 |
| | 45 | .08281 | .91719 | 12.076 | .08309 | 12.035 | 1.0034 | .00343 | .99656 | | 15 |
| 5 | 0 | .08716 | .91284 | 11.474 | .08749 | 11.430 | 1.0038 | .00381 | .99619 | 85 | 0 |
| | 15 | .09150 | .90850 | 10.929 | .09189 | 10.883 | 1.0042 | .00420 | .99580 | | 45 |
| | 30 | .09585 | .90415 | 10.433 | .09629 | 10.385 | 1.0046 | .00460 | .99540 | | 30 |
| | 45 | .10019 | .89981 | 9.9812 | .10069 | 9.9310 | 1.0051 | .00503 | .99497 | | 15 |
| 6 | 0 | .10453 | .89547 | 9.5668 | .10510 | 9.5144 | 1.0055 | .00548 | .99452 | 84 | 0 |
| | 15 | .10887 | .89113 | 9.1855 | .10952 | 9.1309 | 1.0060 | .00594 | .99406 | | 45 |
| | 30 | .11320 | .88680 | 8.8337 | .11393 | 8.7769 | 1.0065 | .00643 | .99357 | | 30 |
| | 45 | .11754 | .88246 | 8.5079 | .11836 | 8.4490 | 1.0070 | .00693 | .99307 | | 15 |
| 7 | 0 | .12187 | .87813 | 8.2055 | .12278 | 8.1443 | 1.0075 | .00745 | .99255 | 83 | 0 |
| | 15 | .12620 | .87380 | 7.9240 | .12722 | 7.8606 | 1.0081 | .00800 | .99200 | | 45 |
| | 30 | .13053 | .86947 | 7.6613 | .13165 | 7.5958 | 1.0086 | .00856 | .99144 | | 30 |
| | 45 | .13485 | .86515 | 7.4156 | .13609 | 7.3479 | 1.0092 | .00913 | .99086 | | 15 |
| 8 | 0 | .13917 | .86083 | 7.1853 | .14054 | 7.1154 | 1.0098 | .00973 | .99027 | 82 | 0 |
| | 15 | .14349 | .85651 | 6.9690 | .14499 | 6.8969 | 1.0105 | .01035 | .98965 | | 45 |
| | 30 | .14781 | .85219 | 6.7655 | .14945 | 6.6912 | 1.0111 | .01098 | .98902 | | 30 |
| | 45 | .15212 | .84788 | 6.5736 | .15391 | 6.4971 | 1.0118 | .01164 | .98836 | | 15 |
| 9 | 0 | .15643 | .84357 | 6.3924 | .15838 | 6.3138 | 1.0125 | .01231 | .98769 | 81 | 0 |
| | 15 | .16074 | .83926 | 6.2211 | .16286 | 6.1402 | 1.0132 | .01300 | .98700 | | 45 |
| | 30 | .16505 | .83495 | 6.0589 | .16734 | 5.9758 | 1.0139 | .01371 | .98629 | | 30 |
| | 45 | .16935 | .83065 | 5.9049 | .17183 | 5.8197 | 1.0147 | .01444 | .98556 | | 15 |
| 10 | 0 | .17365 | .82635 | 5.7588 | .17633 | 5.6713 | 1.0154 | .01519 | .98481 | 80 | 0 |
| | 15 | .17794 | .82206 | 5.6198 | .18083 | 5.5301 | 1.0162 | .01596 | .98404 | | 45 |
| | 30 | .18224 | .81776 | 5.4874 | .18534 | 5.3955 | 1.0170 | .01675 | .98325 | | 30 |
| | 45 | .18652 | .81348 | 5.3612 | .18986 | 5.2672 | 1.0179 | .01755 | .98245 | | 15 |
| 11 | 0 | .19081 | .80919 | 5.2408 | .19438 | 5.1446 | 1.0187 | .01837 | .98163 | 79 | 0 |
| | 15 | .19509 | .80491 | 5.1258 | .19891 | 5.0273 | 1.0196 | .01921 | .98079 | | 45 |
| | 30 | .19937 | .80063 | 5.0158 | .20345 | 4.9152 | 1.0205 | .02008 | .97992 | | 30 |
| | 45 | .20364 | .79636 | 4.9106 | .20800 | 4.8077 | 1.0214 | .02095 | .97905 | | 15 |
| 12 | 0 | .20791 | .79209 | 4.8097 | .21256 | 4.7046 | 1.0223 | .02185 | .97815 | 78 | 0 |
| | 15 | .21218 | .78782 | 4.7130 | .21712 | 4.6057 | 1.0233 | .02277 | .97723 | | 45 |
| | 30 | .21644 | .78356 | 4.6202 | .22169 | 4.5107 | 1.0243 | .02370 | .97630 | | 30 |
| | 45 | .22070 | .77930 | 4.5311 | .22628 | 4.4194 | 1.0253 | .02466 | .97534 | | 15 |
| 13 | 0 | .22495 | .77505 | 4.4454 | .23087 | 4.3315 | 1.0263 | .02563 | .97437 | 77 | 0 |
| | 15 | .22920 | .77080 | 4.3630 | .23547 | 4.2468 | 1.0273 | .02662 | .97338 | | 45 |
| | 30 | .23345 | .76655 | 4.2837 | .24008 | 4.1653 | 1.0284 | .02763 | .97237 | | 30 |
| | 45 | .23769 | .76231 | 4.2072 | .24470 | 4.0867 | 1.0295 | .02866 | .97134 | | 15 |
| 14 | 0 | .24192 | .75808 | 4.1336 | .24933 | 4.0108 | 1.0306 | .02970 | .97030 | 76 | 0 |
| | 15 | .24615 | .75385 | 4.0625 | .25397 | 3.9375 | 1.0317 | .03077 | .96923 | | 45 |
| | 30 | .25038 | .74962 | 3.9939 | .25862 | 3.8667 | 1.0329 | .03185 | .96815 | | 30 |
| | 45 | .25460 | .74540 | 3.9277 | .26328 | 3.7983 | 1.0341 | .03295 | .96705 | | 15 |
| 15 | 0 | .25882 | .74118 | 3.8637 | .26795 | 3.7320 | 1.0353 | .03407 | .96593 | 75 | 0 |
| | | Cosine | Versin | Secant | Cotan | Tan | Cosec | Covers | Sine | Deg. | Min. |

From 75° to 90° read from bottom of table upwards.

**Table 11.—Natural Trigonometrical Functions—*Continued***

| Deg. | Min. | Sine | Covers | Cosec | Tan | Cotan | Secant | Versin | Cosine | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 15 | 0 | 0.25882 | 0.74118 | 3.8637 | 0.26795 | 3.7320 | 1.0353 | 0.03407 | 0.96593 | 75 | 0 |
| | 15 | .26303 | .73697 | 3.8018 | .27263 | 3.6680 | 1.0365 | .03521 | .96479 | | 45 |
| | 30 | .26724 | .73276 | 3.7420 | .27732 | 3.6059 | 1.0377 | .03637 | .96363 | | 30 |
| | 45 | .27144 | .72856 | 3.6840 | .28203 | 3.5457 | 1.0390 | .03754 | .96246 | | 15 |
| 16 | 0 | .27564 | .72436 | 3.6280 | .28674 | 3.4874 | 1.0403 | .03874 | .96126 | 74 | 0 |
| | 15 | .27983 | .72017 | 3.5736 | .29147 | 3.4308 | 1.0416 | .03995 | .96005 | | 45 |
| | 30 | .28402 | .71598 | 3.5209 | .29621 | 3.3759 | 1.0429 | .04118 | .95882 | | 30 |
| | 45 | .28820 | .71180 | 3.4699 | .30096 | 3.3226 | 1.0443 | .04243 | .95757 | | 15 |
| 17 | 0 | .29237 | .70763 | 3.4203 | .30573 | 3.2709 | 1.0457 | .04370 | .95630 | 73 | 0 |
| | 15 | .29654 | .70346 | 3.3722 | .31051 | 3.2205 | 1.0471 | .04498 | .95502 | | 45 |
| | 30 | .30070 | .69929 | 3.3255 | .31530 | 3.1716 | 1.0485 | .04628 | .95372 | | 30 |
| | 45 | .30486 | .69514 | 3.2801 | .32010 | 3.1240 | 1.0500 | .04760 | .95240 | | 15 |
| 18 | 0 | .30902 | .69098 | 3.2361 | .32492 | 3.0777 | 1.0515 | .04894 | .95106 | 72 | 0 |
| | 15 | .31316 | .68684 | 3.1932 | .32975 | 3.0326 | 1.0530 | .05030 | .94970 | | 45 |
| | 30 | .31730 | .68270 | 3.1515 | .33459 | 2.9887 | 1.0545 | .05168 | .94832 | | 30 |
| | 45 | .32144 | .67856 | 3.1110 | .33945 | 2.9459 | 1.0560 | .05307 | .94693 | | 15 |
| 19 | 0 | .32557 | .67443 | 3.0715 | .34433 | 2.9042 | 1.0576 | .05448 | .94552 | 71 | 0 |
| | 15 | .32969 | .67031 | 3.0331 | .34921 | 2.8636 | 1.0592 | .05591 | .94409 | | 45 |
| | 30 | .33381 | .66619 | 2.9957 | .35412 | 2.8239 | 1.0608 | .05736 | .94264 | | 30 |
| | 45 | .33792 | .66208 | 2.9593 | .35904 | 2.7852 | 1.0625 | .05882 | .94118 | | 15 |
| 20 | 0 | .34202 | .65798 | 2.9238 | .36397 | 2.7475 | 1.0642 | .06031 | .93969 | 70 | 0 |
| | 15 | .34612 | .65388 | 2.8892 | .36892 | 2.7106 | 1.0659 | .06181 | .93819 | | 45 |
| | 30 | .35021 | .64979 | 2.8554 | .37388 | 2.6746 | 1.0676 | .06333 | .93667 | | 30 |
| | 45 | .35429 | .64571 | 2.8225 | .37887 | 2.6395 | 1.0694 | .06486 | .93514 | | 15 |
| 21 | 0 | .35837 | .64163 | 2.7904 | .38386 | 2.6051 | 1.0711 | .06642 | .93358 | 69 | 0 |
| | 15 | .36244 | .63756 | 2.7591 | .38888 | 2.5715 | 1.0729 | .06799 | .93201 | | 45 |
| | 30 | .36650 | .63350 | 2.7285 | .39391 | 2.5386 | 1.0748 | .06958 | .93042 | | 30 |
| | 45 | .37056 | .62944 | 2.6986 | .39896 | 2.5065 | 1.0766 | .07119 | .92881 | | 15 |
| 22 | 0 | .37461 | .62539 | 2.6695 | .40403 | 2.4751 | 1.0785 | .07282 | .92718 | 68 | 0 |
| | 15 | .37865 | .62135 | 2.6410 | .40911 | 2.4443 | 1.0804 | .07446 | .92554 | | 45 |
| | 30 | .38268 | .61732 | 2.6131 | .41421 | 2.4142 | 1.0824 | .07612 | .92388 | | 30 |
| | 45 | .38671 | .61329 | 2.5859 | .41933 | 2.3847 | 1.0844 | .07780 | .92220 | | 15 |
| 23 | 0 | .39073 | .60927 | 2.5593 | .42447 | 2.3559 | 1.0864 | .07950 | .92050 | 67 | 0 |
| | 15 | .39474 | .60526 | 2.5333 | .42963 | 2.3276 | 1.0884 | .08121 | .91879 | | 45 |
| | 30 | .39875 | .60125 | 2.5078 | .43481 | 2.2998 | 1.0904 | .08294 | .91706 | | 30 |
| | 45 | .40275 | .59725 | 2.4829 | .44001 | 2.2727 | 1.0925 | .08469 | .91531 | | 15 |
| 24 | 0 | .40674 | .59326 | 2.4586 | .44523 | 2.2460 | 1.0946 | .08645 | .91355 | 66 | 0 |
| | 15 | .41072 | .58928 | 2.4348 | .45047 | 2.2199 | 1.0968 | .08824 | .91176 | | 45 |
| | 30 | .41469 | .58531 | 2.4114 | .45573 | 2.1943 | 1.0989 | .09004 | .90996 | | 30 |
| | 45 | .41866 | .58134 | 2.3886 | .46101 | 2.1692 | 1.1011 | .09186 | .90814 | | 15 |
| 25 | 0 | .42262 | .57738 | 2.3662 | .46631 | 2.1445 | 1.1034 | .09369 | .90631 | 65 | 0 |
| | 15 | .42657 | .57343 | 2.3443 | .47163 | 2.1203 | 1.1056 | .09554 | .90446 | | 45 |
| | 30 | .43051 | .56949 | 2.3228 | .47697 | 2.0965 | 1.1079 | .09741 | .90259 | | 30 |
| | 45 | .43445 | .56555 | 2.3018 | .48234 | 2.0732 | 1.1102 | .09930 | .90070 | | 15 |
| 26 | 0 | .43837 | .56163 | 2.2812 | .48773 | 2.0503 | 1.1126 | .10121 | .89879 | 64 | 0 |
| | 15 | .44229 | .55771 | 2.2610 | .49314 | 2.0278 | 1.1150 | .10313 | .89687 | | 45 |
| | 30 | .44620 | .55380 | 2.2412 | .49858 | 2.0057 | 1.1174 | .10507 | .89493 | | 30 |
| | 45 | .45010 | .54990 | 2.2217 | .50404 | 1.9840 | 1.1198 | .10702 | .89298 | | 15 |
| 27 | 0 | .45399 | .54601 | 2.2027 | .50952 | 1.9626 | 1.1223 | .10899 | .89101 | 63 | 0 |
| | 15 | .45787 | .54213 | 2.1840 | .51503 | 1.9416 | 1.1248 | .11098 | .88902 | | 45 |
| | 30 | .46175 | .53825 | 2.1657 | .52057 | 1.9210 | 1.1274 | .11299 | .88701 | | 30 |
| | 45 | .46561 | .53439 | 2.1477 | .52612 | 1.9007 | 1.1300 | .11501 | .88499 | | 15 |
| 28 | 0 | .46947 | .53053 | 2.1300 | .53171 | 1.8807 | 1.1326 | .11705 | .88295 | 62 | 0 |
| | 15 | .47332 | .52668 | 2.1127 | .53732 | 1.8611 | 1.1352 | .11911 | .88089 | | 45 |
| | 30 | .47716 | .52284 | 2.0957 | .54295 | 1.8418 | 1.1379 | .12118 | .87882 | | 30 |
| | 45 | .48099 | .51901 | 2.0790 | .54862 | 1.8228 | 1.1406 | .12327 | .87673 | | 15 |
| 29 | 0 | .48481 | .51519 | 2.0627 | .55431 | 1.8040 | 1.1433 | .12538 | .87462 | 61 | 0 |
| | 15 | .48862 | .51138 | 2.0466 | .56003 | 1.7856 | 1.1461 | .12750 | .87250 | | 45 |
| | 30 | .49242 | .50758 | 2.0308 | .56577 | 1.7675 | 1.1490 | .12964 | .87036 | | 30 |
| | 45 | .49622 | .50378 | 2.0152 | .57155 | 1.7496 | 1.1518 | .13180 | .86820 | | 15 |
| 30 | 0 | .50000 | .50000 | 2.0000 | .57735 | 1.7320 | 1.1547 | .13397 | .86603 | 60 | 0 |
| | | Cosine | Versin | Secant | Cotan | Tan | Cosec | Covers | Sine | Deg. | Min. |

From 60° to 75° read from bottom of table upwards.

Table 11.—Natural Trigonometrical Functions—*Continued*

| Deg. | Min. | Sine | Covers | Cosec | Tan | Cotan | Secant | Versin | Cosine | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **30** | 0 | 0.50000 | 0.50000 | 2.0000 | 0.57735 | 1.7320 | 1.1547 | 0.13397 | 0.86603 | 60 | 0 |
| | 15 | .50377 | .49623 | 1.9850 | .58318 | 1.7147 | 1.1576 | .13616 | .86384 | | 45 |
| | 30 | .50754 | .49246 | 1.9703 | .58904 | 1.6977 | 1.1606 | .13837 | .86163 | | 30 |
| | 45 | .51129 | .48871 | 1.9558 | .59494 | 1.6808 | 1.1636 | .14059 | .85941 | | 15 |
| **31** | 0 | .51504 | .48496 | 1.9416 | .60086 | 1.6643 | 1.1666 | .14283 | .85717 | 59 | 0 |
| | 15 | .51877 | .48123 | 1.9276 | .60681 | 1.6479 | 1.1697 | .14509 | .85491 | | 45 |
| | 30 | .52250 | .47750 | 1.9139 | .61280 | 1.6319 | 1.1728 | .14736 | .85264 | | 30 |
| | 45 | .52621 | .47379 | 1.9004 | .61882 | 1.6160 | 1.1760 | .14965 | .85035 | | 15 |
| **32** | 0 | .52992 | .47008 | 1.8871 | .62487 | 1.6003 | 1.1792 | .15195 | .84805 | 58 | 0 |
| | 15 | .53361 | .46639 | 1.8740 | .63095 | 1.5849 | 1.1824 | .15427 | .84573 | | 45 |
| | 30 | .53730 | .46270 | 1.8612 | .63707 | 1.5697 | 1.1857 | .15661 | .84339 | | 30 |
| | 45 | .54097 | .45903 | 1.8485 | .64322 | 1.5547 | 1.1890 | .15896 | .84104 | | 15 |
| **33** | 0 | .54464 | .45536 | 1.8361 | .64941 | 1.5399 | 1.1924 | .16133 | .83867 | 57 | 0 |
| | 15 | .54829 | .45171 | 1.8238 | .65563 | 1.5253 | 1.1958 | .16371 | .83629 | | 45 |
| | 30 | .55194 | .44806 | 1.8118 | .66188 | 1.5108 | 1.1992 | .16611 | .83389 | | 30 |
| | 45 | .55557 | .44443 | 1.7999 | .66818 | 1.4966 | 1.2027 | .16853 | .83147 | | 15 |
| **34** | 0 | .55919 | .44081 | 1.7883 | .67451 | 1.4826 | 1.2062 | .17096 | .82904 | 56 | 0 |
| | 15 | .56280 | .43720 | 1.7768 | .68087 | 1.4687 | 1.2098 | .17341 | .82659 | | 45 |
| | 30 | .56641 | .43359 | 1.7655 | .68728 | 1.4550 | 1.2134 | .17587 | .82413 | | 30 |
| | 45 | .57000 | .43000 | 1.7544 | .69372 | 1.4415 | 1.2171 | .17835 | .82165 | | 15 |
| **35** | 0 | .57358 | .42642 | 1.7434 | .70021 | 1.4281 | 1.2208 | .18085 | .81915 | 55 | 0 |
| | 15 | .57715 | .42285 | 1.7327 | .70673 | 1.4150 | 1.2245 | .18336 | .81664 | | 45 |
| | 30 | .58070 | .41930 | 1.7220 | .71329 | 1.4019 | 1.2283 | .18588 | .81412 | | 30 |
| | 45 | .58425 | .41575 | 1.7116 | .71990 | 1.3891 | 1.2322 | .18843 | .81157 | | 15 |
| **36** | 0 | .58779 | .41221 | 1.7013 | .72654 | 1.3764 | 1.2361 | .19098 | .80902 | 54 | 0 |
| | 15 | .59131 | .40869 | 1.6912 | .73323 | 1.3638 | 1.2400 | .19356 | .80644 | | 45 |
| | 30 | .59482 | .40518 | 1.6812 | .73996 | 1.3514 | 1.2440 | .19614 | .80386 | | 30 |
| | 45 | .59832 | .40168 | 1.6713 | .74673 | 1.3392 | 1.2480 | .19875 | .80125 | | 15 |
| **37** | 0 | .60181 | .39819 | 1.6616 | .75355 | 1.3270 | 1.2521 | .20136 | .79864 | 53 | 0 |
| | 15 | .60529 | .39471 | 1.6521 | .76042 | 1.3151 | 1.2563 | .20400 | .79600 | | 45 |
| | 30 | .60876 | .39124 | 1.6427 | .76733 | 1.3032 | 1.2605 | .20665 | .79335 | | 30 |
| | 45 | .61222 | .38778 | 1.6334 | .77428 | 1.2915 | 1.2647 | .20931 | .79069 | | 15 |
| **38** | 0 | .61566 | .38434 | 1.6243 | .78129 | 1.2799 | 1.2690 | .21199 | .78801 | 52 | 0 |
| | 15 | .61909 | .38091 | 1.6153 | .78834 | 1.2685 | 1.2734 | .21468 | .78532 | | 45 |
| | 30 | .62251 | .37749 | 1.6064 | .79543 | 1.2572 | 1.2778 | .21739 | .78261 | | 30 |
| | 45 | .62592 | .37408 | 1.5976 | .80258 | 1.2460 | 1.2822 | .22012 | .77988 | | 15 |
| **39** | 0 | .62932 | .37068 | 1.5890 | .80978 | 1.2349 | 1.2868 | .22285 | .77715 | 51 | 0 |
| | 15 | .63271 | .36729 | 1.5805 | .81703 | 1.2239 | 1.2913 | .22561 | .77439 | | 45 |
| | 30 | .63608 | .36392 | 1.5721 | .82434 | 1.2131 | 1.2960 | .22838 | .77162 | | 30 |
| | 45 | .63944 | .36056 | 1.5639 | .83169 | 1.2024 | 1.3007 | .23116 | .76884 | | 15 |
| **40** | 0 | .64279 | .35721 | 1.5557 | .83910 | 1.1918 | 1.3054 | .23396 | .76604 | 50 | 0 |
| | 15 | .64612 | .35388 | 1.5477 | .84656 | 1.1812 | 1.3102 | .23677 | .76323 | | 45 |
| | 30 | .64945 | .35055 | 1.5398 | .85408 | 1.1708 | 1.3151 | .23959 | .76041 | | 30 |
| | 45 | .65276 | .34724 | 1.5320 | .86165 | 1.1606 | 1.3200 | .24244 | .75756 | | 15 |
| **41** | 0 | .65606 | .34394 | 1.5242 | .86929 | 1.1504 | 1.3250 | .24529 | .75471 | 49 | 0 |
| | 15 | .65935 | .34065 | 1.5166 | .87698 | 1.1403 | 1.3301 | .24816 | .75184 | | 45 |
| | 30 | .66262 | .33738 | 1.5092 | .88472 | 1.1303 | 1.3352 | .25104 | .74896 | | 30 |
| | 45 | .66588 | .33412 | 1.5018 | .89253 | 1.1204 | 1.3404 | .25394 | .74606 | | 15 |
| **42** | 0 | .66913 | .33087 | 1.4945 | .90040 | 1.1106 | 1.3456 | .25686 | .74314 | 48 | 0 |
| | 15 | .67237 | .32763 | 1.4873 | .90834 | 1.1009 | 1.3509 | .25978 | .74022 | | 45 |
| | 30 | .67559 | .32441 | 1.4802 | .91633 | 1.0913 | 1.3563 | .26272 | .73728 | | 30 |
| | 45 | .67880 | .32120 | 1.4732 | .92439 | 1.0818 | 1.3618 | .26568 | .73432 | | 15 |
| **43** | 0 | .68200 | .31800 | 1.4663 | .93251 | 1.0724 | 1.3673 | .26865 | .73135 | 47 | 0 |
| | 15 | .68518 | .31482 | 1.4595 | .94071 | 1.0630 | 1.3729 | .27163 | .72837 | | 45 |
| | 30 | .68835 | .31165 | 1.4527 | .94896 | 1.0538 | 1.3786 | .27463 | .72537 | | 30 |
| | 45 | .69151 | .30849 | 1.4461 | .95729 | 1.0446 | 1.3843 | .27764 | .72236 | | 15 |
| **44** | 0 | .69466 | .30534 | 1.4396 | .96569 | 1.0355 | 1.3902 | .28066 | .71934 | 46 | 0 |
| | 15 | .69779 | .30221 | 1.4331 | .97416 | 1.0265 | 1.3961 | .28370 | .71630 | | 45 |
| | 30 | .70091 | .29909 | 1.4267 | .98270 | 1.0176 | 1.4020 | .28675 | .71325 | | 30 |
| | 45 | .70401 | .29599 | 1.4204 | .99131 | 1.0088 | 1.4081 | .28981 | .71019 | | 15 |
| **45** | 0 | .70711 | .29289 | 1.4142 | .10000 | 1.0000 | 1.4142 | .29289 | .70711 | 45 | 0 |
| | | Cosine | Versin | Secant | Cotan | Tan | Cosec | Covers | Sine | Deg. | Min. |

From 45° to 60° read from bottom of table upwards.

### Table 12.—Logarithmic Trigonometric Functions

| Deg. | Sine | Cosec | Versin | Tangent | Cotan | Covers | Secant | Cosine | Deg |
|---|---|---|---|---|---|---|---|---|---|
| 0 | − ∞ | + ∞ | − ∞ | − ∞ | + ∞ | 10.00000 | 10.00000 | 10.00000 | 90 |
| 1 | 8.24186 | 11.75814 | 6.18271 | 8.24192 | 11.75808 | 9.99235 | 10.00007 | 9.99993 | 89 |
| 2 | 8.54282 | 11.45718 | 6.78474 | 8.54308 | 11.45692 | 9.98457 | 10.00026 | 9.99974 | 88 |
| 3 | 8.71880 | 11.28120 | 7.13687 | 8.71940 | 11.28060 | 9.97665 | 10.00060 | 9.99940 | 87 |
| 4 | 8.84358 | 11.15642 | 7.38667 | 8.84464 | 11.15536 | 9.96860 | 10.00106 | 9.99894 | 86 |
| 5 | 8.94030 | 11.05970 | 7.58039 | 8.94195 | 11.05805 | 9.96040 | 10.00166 | 9.99834 | 85 |
| 6 | 9.01923 | 10.98077 | 7.73863 | 9.02162 | 10.97838 | 9.95205 | 10.00239 | 9.99761 | 84 |
| 7 | 9.08589 | 10.91411 | 7.87238 | 9.08914 | 10.91086 | 9.94356 | 10.00325 | 9.99675 | 83 |
| 8 | 9.14356 | 10.85644 | 7.98820 | 9.14780 | 10.85220 | 9.93492 | 10.00425 | 9.99575 | 82 |
| 9 | 9.19433 | 10.80567 | 8.09032 | 9.19971 | 10.80029 | 9.92612 | 10.00538 | 9.99462 | 81 |
| 10 | 9.23967 | 10.76033 | 8.18162 | 9.24632 | 10.75368 | 9.91717 | 10.00665 | 9.99335 | 80 |
| 11 | 9.28060 | 10.71940 | 8.26418 | 9.28865 | 10.71135 | 9.90805 | 10.00805 | 9.99195 | 79 |
| 12 | 9.31788 | 10.68212 | 8.33950 | 9.32747 | 10.67253 | 9.89877 | 10.00960 | 9.99040 | 78 |
| 13 | 9.35209 | 10.64791 | 8.40875 | 9.36336 | 10.63664 | 9.88933 | 10.01128 | 9.98872 | 77 |
| 14 | 9.38368 | 10.61632 | 8.47282 | 9.39677 | 10.60323 | 9.87971 | 10.01310 | 9.98690 | 76 |
| 15 | 9.41300 | 10.58700 | 8.53243 | 9.42805 | 10.57195 | 9.86992 | 10.01506 | 9.98494 | 75 |
| 16 | 9.44034 | 10.55966 | 8.58814 | 9.45750 | 10.54250 | 9.85996 | 10.01716 | 9.98284 | 74 |
| 17 | 9.46594 | 10.53406 | 8.64043 | 9.48534 | 10.51466 | 9.84981 | 10.01940 | 9.98060 | 73 |
| 18 | 9.48998 | 10.51002 | 8.68969 | 9.51178 | 10.48822 | 9.83947 | 10.02179 | 9.97821 | 72 |
| 19 | 9.51264 | 10.48736 | 8.73625 | 9.53697 | 10.46303 | 9.82894 | 10.02433 | 9.97567 | 71 |
| 20 | 9.53405 | 10.46595 | 8.78037 | 9.56107 | 10.43893 | 9.81821 | 10.02701 | 9.97299 | 70 |
| 21 | 9.55433 | 10.44567 | 8.82230 | 9.58418 | 10.41582 | 9.80729 | 10.02985 | 9.97015 | 69 |
| 22 | 9.57358 | 10.42642 | 8.86223 | 9.60641 | 10.39359 | 9.79615 | 10.03283 | 9.96717 | 68 |
| 23 | 9.59188 | 10.40812 | 8.90034 | 9.62785 | 10.37215 | 9.78481 | 10.03597 | 9.96403 | 67 |
| 24 | 9.60931 | 10.39069 | 8.93679 | 9.64858 | 10.35142 | 9.77325 | 10.03927 | 9.96073 | 66 |
| 25 | 9.62595 | 10.37405 | 8.97170 | 9.66867 | 10.33133 | 9.76146 | 10.04272 | 9.95728 | 65 |
| 26 | 9.64184 | 10.35816 | 9.00521 | 9.68818 | 10.31182 | 9.74945 | 10.04634 | 9.95366 | 64 |
| 27 | 9.65705 | 10.34295 | 9.03740 | 9.70717 | 10.29283 | 9.73720 | 10.05012 | 9.94988 | 63 |
| 28 | 9.67161 | 10.32839 | 9.06838 | 9.72567 | 10.27433 | 9.72471 | 10.05407 | 9.94593 | 62 |
| 29 | 9.68557 | 10.31443 | 9.09823 | 9.74375 | 10.25625 | 9.71197 | 10.05818 | 9.94182 | 61 |
| 30 | 9.69897 | 10.30103 | 9.12702 | 9.76144 | 10.23856 | 9.69897 | 10.06247 | 9.93753 | 60 |
| 31 | 9.71184 | 10.28816 | 9.15483 | 9.77877 | 10.22123 | 9.68571 | 10.06693 | 9.93307 | 59 |
| 32 | 9.72421 | 10.27579 | 9.18171 | 9.79579 | 10.20421 | 9.67217 | 10.07158 | 9.92842 | 58 |
| 33 | 9.73611 | 10.26389 | 9.20771 | 9.81252 | 10.18748 | 9.65836 | 10.07641 | 9.92359 | 57 |
| 34 | 9.74756 | 10.25244 | 9.23290 | 9.82899 | 10.17101 | 9.64425 | 10.08143 | 9.91857 | 56 |
| 35 | 9.75859 | 10.24141 | 9.25731 | 9.84523 | 10.15477 | 9.62984 | 10.08664 | 9.91336 | 55 |
| 36 | 9.76922 | 10.23078 | 9.28099 | 9.86126 | 10.13874 | 9.61512 | 10.09204 | 9.90796 | 54 |
| 37 | 9.77946 | 10.22054 | 9.30398 | 9.87711 | 10.12289 | 9.60008 | 10.09765 | 9.90235 | 53 |
| 38 | 9.78934 | 10.21066 | 9.32631 | 9.89281 | 10.10719 | 9.58471 | 10.10347 | 9.89653 | 52 |
| 39 | 9.79887 | 10.20113 | 9.34802 | 9.90837 | 10.09163 | 9.56900 | 10.10950 | 9.89050 | 51 |
| 40 | 9.80807 | 10.19193 | 9.36913 | 9.92381 | 10.07619 | 9.55293 | 10.11575 | 9.88425 | 50 |
| 41 | 9.81694 | 10.18306 | 9.38968 | 9.93916 | 10.06084 | 9.53648 | 10.12222 | 9.87778 | 49 |
| 42 | 9.82551 | 10.17449 | 9.40969 | 9.95444 | 10.04556 | 9.51966 | 10.12893 | 9.87107 | 48 |
| 43 | 9.83378 | 10.16622 | 9.42918 | 9.96966 | 10.03034 | 9.50243 | 10.13587 | 9.86413 | 47 |
| 44 | 9.84177 | 10.15823 | 9.44818 | 9.98484 | 10.01516 | 9.48479 | 10.14307 | 9.85693 | 46 |
| 45 | 9.84949 | 10.15052 | 9.46671 | 10.00000 | 10.00000 | 9.46671 | 10.15052 | 9.84949 | 45 |
|  | Cosine | Secant | Covers | Cotan | Tangent | Versin | Cosec | Sine |  |

From 45° to 90° read from bottom of table upwards,

### Table 13.—Fractions Expressed as Decimals of an Inch and of 1 Foot

| Inches | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Decimal of 1 ft. | 0.0833 | 0.1667 | 0.2500 | 0.3333 | 0.4167 | 0.5000 | 0.5833 | 0.6667 | 0 7500 | 0 8333 | 0.9167 |

Fractions of 1 inch

| Fraction | Decimal |  | Fraction | Decimal |  | Fraction | Decimal |  | Fraction | Decimal |  |
|---|---|---|---|---|---|---|---|---|---|---|---|
|  | of 1 in. | of 1 ft. |  | of 1 in. | of 1 ft. |  | of 1 in. | of 1 ft. |  | of 1 in. | of 1 ft. |
| 1/64 | 0.015625 | 0.0013 | 17/64 | 0.265625 | 0.0221 | 33/64 | 0.515625 | 0.0430 | 49/64 | 0.765625 | 0 0638 |
| 1/32 | .03125 | .0026 | 9/32 | .28125 | .0234 | 17/32 | .53125 | .0443 | 25/32 | .78125 | .0651 |
| 3/64 | .046875 | .0039 | 19/64 | .296875 | .0247 | 35/64 | .546875 | .0456 | 51/64 | .796875 | .0664 |
| 1/16 | .0625 | .0052 | 5/16 | .3125 | .0260 | 9/16 | .5625 | .0469 | 13/16 | .8125 | .0677 |
| 5/64 | .078125 | .0065 | 21/64 | .328125 | .0273 | 37/64 | .578125 | .0482 | 53/64 | .828125 | .0690 |
| 3/32 | .09375 | .0078 | 11/32 | .34375 | .0286 | 19/32 | .59375 | .0495 | 27/32 | .84375 | .0703 |
| 7/64 | .109375 | .0091 | 23/64 | .359375 | .0299 | 39/64 | .609375 | .0508 | 55/64 | .859375 | .0716 |
| 1/8 | .125 | 0104 | 3/8 | .375 | .0313 | 5/8 | .625 | .0521 | 7/8 | .875 | .0729 |
| 9/64 | .140625 | .0117 | 25/64 | .390625 | .0326 | 41/64 | .640625 | .0534 | 57/64 | .890625 | .0742 |
| 5/32 | .15625 | .0130 | 13/32 | .40625 | .0339 | 21/32 | .65625 | .0547 | 29/32 | .90625 | .0755 |
| 11/64 | .171875 | .0143 | 27/64 | .421875 | .0352 | 43/64 | 671875 | .0560 | 59/64 | .921875 | .0768 |
| 3/16 | .1875 | .0156 | 7/16 | .4375 | .0365 | 11/16 | .6875 | .0573 | 15/16 | .9375 | .0781 |
| 13/64 | .203125 | .0169 | 29/64 | .453125 | .0378 | 45/64 | .703125 | .0586 | 61/64 | .953125 | .0794 |
| 7/32 | .21875 | .0182 | 15/32 | .46875 | .0391 | 23/32 | .71875 | .0599 | 31/32 | .96875 | .0807 |
| 15/64 | .234375 | .0195 | 31/64 | .484375 | .0404 | 47/64 | .734375 | .0612 | 63/64 | .984375 | .0820 |
| 1/4 | .25 | .0208 | 1/2 | .50 | .0417 | 3/4 | .75 | .0625 | .... | ........ | ...... |

Example: 9 19/32 in. = 0.7500 + 0.0495 = 0.7995 ft.

**Table 14.—Temperature Conversion—Centigrade to Fahrenheit**

| VALUES FOR INTERPOLATION IN TABLE | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Deg. C.... | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| Deg. F.... | 1.8 | 3.6 | 5.4 | 7.2 | 9.0 | 10.8 | 12.6 | 14.4 | 16.2 | 18.0 |

| Deg.→<br>C.↓ | 0 | 10 | 20 | 30 | 40 | 50 | 60 | 70 | 80 | 90 |
|---|---|---|---|---|---|---|---|---|---|---|
| | Degrees, Fahrenheit | | | | | | | | | |
| **−200** | −328 | −346 | −364 | −382 | −400 | −418 | −436 | −454 | ...... | ...... |
| **−100** | −148 | −166 | −184 | −202 | −220 | −238 | −256 | −274 | −292 | −310 |
| **−0** | +32 | +14 | −4 | −22 | −40 | −58 | −76 | −94 | −112 | −130 |
| **+0** | +32 | +50 | +68 | +86 | +104 | +122 | +140 | +158 | +176 | +194 |
| **100** | 212 | 230 | 248 | 266 | 284 | 302 | 320 | 338 | 356 | 374 |
| **200** | 392 | 410 | 428 | 446 | 464 | 482 | 500 | 518 | 536 | 554 |
| **300** | 572 | 590 | 608 | 626 | 644 | 662 | 680 | 698 | 716 | 734 |
| **400** | 752 | 770 | 788 | 806 | 824 | 842 | 860 | 878 | 896 | 914 |
| **500** | 932 | 950 | 968 | 986 | 1004 | 1022 | 1040 | 1058 | 1076 | 1094 |
| **600** | 1112 | 1130 | 1148 | 1166 | 1184 | 1202 | 1220 | 1238 | 1256 | 1274 |
| **700** | 1292 | 1310 | 1328 | 1346 | 1364 | 1382 | 1400 | 1418 | 1436 | 1454 |
| **800** | 1472 | 1490 | 1508 | 1526 | 1544 | 1562 | 1580 | 1598 | 1616 | 1634 |
| **900** | 1652 | 1670 | 1688 | 1706 | 1724 | 1742 | 1760 | 1778 | 1796 | 1814 |
| **1000** | 1832 | 1850 | 1868 | 1886 | 1904 | 1922 | 1940 | 1958 | 1976 | 1994 |
| **1100** | 2012 | 2030 | 2048 | 2066 | 2084 | 2102 | 2120 | 2138 | 2156 | 2174 |
| **1200** | 2192 | 2210 | 2228 | 2246 | 2264 | 2282 | 2300 | 2318 | 2336 | 2354 |
| **1300** | 2372 | 2390 | 2408 | 2426 | 2444 | 2462 | 2480 | 2498 | 2516 | 2534 |
| **1400** | 2552 | 2570 | 2588 | 2606 | 2624 | 2642 | 2660 | 2678 | 2696 | 2714 |
| **1500** | 2732 | 2750 | 2768 | 2786 | 2804 | 2822 | 2840 | 2858 | 2876 | 2894 |
| **1600** | 2912 | 2930 | 2948 | 2966 | 2984 | 3002 | 3020 | 3038 | 3056 | 3074 |
| **1700** | 3092 | 3110 | 3128 | 3146 | 3164 | 3182 | 3200 | 3218 | 3236 | 3254 |
| **1800** | 3272 | 3290 | 3308 | 3326 | 3344 | 3362 | 3380 | 3398 | 3416 | 3434 |
| **1900** | 3452 | 3470 | 3488 | 3506 | 3524 | 3542 | 3560 | 3578 | 3596 | 3614 |
| **2000** | 3632 | 3650 | 3668 | 3686 | 3704 | 3722 | 3740 | 3758 | 3776 | 3794 |
| **2100** | 3812 | 3830 | 3848 | 3866 | 3884 | 3902 | 3920 | 3938 | 3956 | 3974 |
| **2200** | 3992 | 4010 | 4028 | 4046 | 4064 | 4082 | 4100 | 4118 | 4136 | 4154 |
| **2300** | 4172 | 4190 | 4208 | 4226 | 4244 | 4262 | 4280 | 4298 | 4316 | 4334 |
| **2400** | 4352 | 4370 | 4388 | 4406 | 4424 | 4442 | 4460 | 4478 | 4496 | 4514 |
| **2500** | 4532 | 4550 | 4568 | 4586 | 4604 | 4622 | 4640 | 4658 | 4676 | 4694 |
| **2600** | 4712 | 4730 | 4748 | 4766 | 4784 | 4802 | 4820 | 4838 | 4856 | 4874 |
| **2700** | 4892 | 4910 | 4928 | 4946 | 4964 | 4982 | 5000 | 5018 | 5036 | 5054 |
| **2800** | 5072 | 5090 | 5108 | 5126 | 5144 | 5162 | 5180 | 5198 | 5216 | 5234 |
| **2900** | 5252 | 5270 | 5288 | 5306 | 5324 | 5342 | 5360 | 5378 | 5396 | 5414 |
| **3000** | 5432 | 5450 | 5468 | 5486 | 5504 | 5522 | 5540 | 5558 | 5576 | 5594 |
| **3100** | 5612 | 5630 | 5648 | 5666 | 5684 | 5702 | 5720 | 5738 | 5756 | 5774 |
| **3200** | 5792 | 5810 | 5828 | 5846 | 5864 | 5882 | 5900 | 5918 | 5936 | 5954 |
| **3300** | 5972 | 5990 | 6008 | 6026 | 6044 | 6062 | 6080 | 6098 | 6116 | 6134 |
| **3400** | 6152 | 6170 | 6188 | 6206 | 6224 | 6242 | 6260 | 6278 | 6296 | 6314 |
| **3500** | 6332 | 6350 | 6368 | 6386 | 6404 | 6422 | 6440 | 6458 | 6476 | 6494 |
| **3600** | 6512 | 6530 | 6548 | 6566 | 6584 | 6602 | 6620 | 6638 | 6656 | 6674 |
| **3700** | 6692 | 6710 | 6728 | 6746 | 6764 | 6782 | 6800 | 6818 | 6836 | 6854 |
| **3800** | 6872 | 6890 | 6908 | 6926 | 6944 | 6962 | 6980 | 6998 | 7016 | 7034 |
| **3900** | 7052 | 7070 | 7088 | 7106 | 7124 | 7142 | 7160 | 7178 | 7196 | 7214 |

EXAMPLES: 1347° C. = 1340° C. + 7° C. = 2444° F. + 12.6° F. = 2456.6° F.
−194° C. = − 190° C. + (− 4° C.) = − 310° F. + (− 7.2° F.) = − 317.2° F.
1852° F. = 1850° F. + 2° F. = 1010° C. + 1.11° C. = 1011.11° C.
−226° F. = − 220° F. + (− 6° F.) = − 134.44° C. + (− 3.33° C.) = − 137.77° C.

In the Fahrenheit thermometer the freezing-point of water is taken at 32°, and the boiling-point of water at mean atmospheric pressure at the sea-level, 14.7 lb. per sq. in., is taken at 212°, the distance between these two points being divided into 180°. In the Centigrade and Réaumur thermometers the freezing-point is taken at 0°. The boiling-point is 100° in the Centigrade scale, and 80° in the Réaumur.

| | | |
|---|---|---|
| 1 Fahrenheit degree | = $^{5}/_{9}$ deg. Centigrade | = $^{4}/_{9}$ deg. Réaumur. |
| 1 Centigrade degree | = $^{9}/_{5}$ deg. Fahrenheit | = $^{4}/_{5}$ deg. Réaumur. |
| 1 Réaumur degree | = $^{9}/_{4}$ deg. Fahrenheit | = $^{5}/_{4}$ deg. Centigrade. |
| Temperature Fahrenheit | = $^{9}/_{5}$ × temp. C. + 32° | = $^{9}/_{4}$ temp. R. + 32°. |
| Temperature Centigrade | = $^{5}/_{9}$ (temp. F. −32°) | = $^{5}/_{4}$ temp. R. |
| Temperature Réaumur | = $^{4}/_{5}$ temp. C. | = $^{4}/_{9}$ (temp. F. −32°) |

### Table 15.—Temperature Conversion—Fahrenheit to Centigrade

| Values for Interpolation in Table | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Deg. F... | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| Deg. C... | 0.555 | 1.111 | 1.666 | 2.222 | 2.777 | 3.333 | 3.888 | 4.444 | 5.000 | 5.555 |

| Deg. → <br> F ↓ | 0 | 10 | 20 | 30 | 40 | 50 | 60 | 70 | 80 | 90 |
|---|---|---|---|---|---|---|---|---|---|---|
| | Degrees, Centigrade | | | | | | | | | |
| −400 | −240.00 | −245.56 | −251.11 | −256.67 | −262.22 | −267.78 | −277.33 | ...... | ...... | ...... |
| −300 | −184.44 | −190.00 | −195.56 | −201.11 | −206.67 | −212.22 | −217.78 | −223.23 | −228.89 | −234.44 |
| −200 | −128.89 | −134.44 | −140.00 | −145.56 | −151.11 | −156.67 | −162.22 | −167.78 | −173.33 | −178.89 |
| −100 | −73.33 | −78.89 | −84.44 | −90.00 | −95.56 | −101.11 | −106.67 | −112.22 | −117.78 | −123.33 |
| −0 | −17.78 | −23.33 | −28.89 | −33.44 | −40.00 | −45.56 | −51.11 | −56.67 | −62.22 | −67.78 |
| +0 | −17.78 | −12.22 | −6.67 | −1.11 | +4.44 | +10.00 | +15.56 | +21.11 | +26.67 | +32.22 |
| 100 | 37.78 | 43.33 | 48.89 | 54.44 | 60.00 | 65.56 | 71.11 | 76.67 | 82.22 | 87.78 |
| 200 | 93.33 | 98.89 | 104.44 | 110.00 | 115.56 | 121.11 | 126.67 | 132.22 | 137.78 | 143.33 |
| 300 | 148.89 | 154.44 | 160.00 | 165.56 | 171.11 | 176.67 | 182.22 | 187.78 | 193.33 | 198.89 |
| 400 | 204.44 | 210.00 | 215.56 | 221.11 | 226.67 | 232.22 | 237.78 | 243.33 | 248.89 | 254.44 |
| 500 | 260.00 | 265.56 | 271.11 | 276.67 | 282.22 | 287.78 | 293.33 | 298.89 | 304.44 | 310.00 |
| 600 | 315.56 | 321.11 | 326.67 | 332.22 | 337.78 | 343.33 | 348.89 | 354.44 | 360.00 | 365.56 |
| 700 | 371.11 | 376.67 | 382.22 | 387.78 | 393.33 | 398.89 | 404.44 | 410.00 | 415.56 | 421.11 |
| 800 | 426.67 | 432.22 | 437.78 | 443.33 | 448.89 | 454.44 | 460.00 | 465.56 | 471.11 | 476.67 |
| 900 | 482.22 | 487.78 | 493.33 | 498.89 | 504.44 | 510.00 | 515.56 | 521.11 | 526.67 | 532.22 |
| 1000 | 537.78 | 543.33 | 548.89 | 554.44 | 560.00 | 565.56 | 571.11 | 576.67 | 582.22 | 587.78 |
| 1100 | 593.33 | 598.89 | 604.44 | 610.00 | 615.56 | 621.11 | 626.67 | 632.22 | 637.78 | 643.33 |
| 1200 | 648.89 | 654.44 | 660.00 | 655.56 | 671.11 | 676.67 | 682.22 | 687.78 | 693.33 | 698.89 |
| 1300 | 704.44 | 710.00 | 715.56 | 721.11 | 726.67 | 732.22 | 737.78 | 743.33 | 748.89 | 754.44 |
| 1400 | 760.00 | 765.56 | 771.11 | 776.67 | 782.22 | 787.78 | 793.33 | 798.89 | 804.44 | 810.00 |
| 1500 | 815.56 | 821.11 | 826.67 | 832.22 | 837.78 | 843.33 | 848.89 | 854.44 | 860.00 | 865.56 |
| 1600 | 871.11 | 876.67 | 882.22 | 887.78 | 893.33 | 898.89 | 904.44 | 910.00 | 915.56 | 921.11 |
| 1700 | 926.67 | 932.22 | 937.78 | 943.33 | 948.89 | 954.44 | 960.00 | 965.56 | 971.11 | 976.67 |
| 1800 | 982.22 | 987.78 | 993.33 | 998.89 | 1004.4 | 1010.0 | 1015.6 | 1021.1 | 1026.7 | 1032.2 |
| 1900 | 1037.8 | 1043.3 | 1048.9 | 1054.4 | 1060.0 | 1065.6 | 1071.1 | 1076.7 | 1082.2 | 1087.8 |
| 2000 | 1093.3 | 1098.9 | 1104.4 | 1110.0 | 1115.6 | 1121.1 | 1126.7 | 1132.2 | 1137.8 | 1143.3 |
| 2100 | 1148.9 | 1154.4 | 1160.0 | 1165.6 | 1171.1 | 1176.7 | 1182.2 | 1187.8 | 1193.3 | 1198.9 |
| 2200 | 1204.4 | 1210.0 | 1215.6 | 1221.1 | 1226.7 | 1232.2 | 1237.8 | 1243.3 | 1248.9 | 1254.4 |
| 2300 | 1260.0 | 1265.6 | 1271.1 | 1276.7 | 1282.2 | 1287.8 | 1293.3 | 1298.9 | 1304.4 | 1310.0 |
| 2400 | 1315.6 | 1321.1 | 1326.7 | 1332.2 | 1337.8 | 1343.3 | 1348.9 | 1354.4 | 1360.0 | 1365.6 |
| 2500 | 1371.1 | 1376.7 | 1382.2 | 1387.8 | 1393.3 | 1398.9 | 1404.4 | 1410.0 | 1415.6 | 1421.1 |
| 2600 | 1426.7 | 1432.2 | 1437.8 | 1443.3 | 1448.9 | 1454.4 | 1460.0 | 1465.6 | 1471.1 | 1476.7 |
| 2700 | 1482.2 | 1487.8 | 1493.3 | 1498.9 | 1504.4 | 1510.0 | 1515.6 | 1521.1 | 1526.7 | 1532.2 |
| 2800 | 1537.8 | 1543.3 | 1548.9 | 1554.4 | 1560.0 | 1565.6 | 1571.1 | 1576.7 | 1582.2 | 1587.8 |
| 2900 | 1593.3 | 1598.9 | 1604.4 | 1610.0 | 1615.6 | 1621.1 | 1626.7 | 1632.2 | 1637.8 | 1643.3 |
| 3000 | 1648.9 | 1654.4 | 1660.0 | 1665.6 | 1671.1 | 1676.7 | 1682.2 | 1687.8 | 1693.3 | 1698.9 |
| 3100 | 1704.4 | 1710.0 | 1715.6 | 1721.1 | 1726.7 | 1732.2 | 1737.8 | 1743.3 | 1748.9 | 1754.4 |
| 3200 | 1760.0 | 1765.6 | 1771.1 | 1776.7 | 1782.2 | 1787.8 | 1793.3 | 1798.9 | 1804.4 | 1810.0 |
| 3300 | 1815.6 | 1821.1 | 1826.7 | 1832.2 | 1837.8 | 1843.3 | 1848.9 | 1854.4 | 1860.0 | 1865.6 |
| 3400 | 1871.1 | 1876.7 | 1882.2 | 1887.8 | 1893.3 | 1898.9 | 1904.4 | 1910.0 | 1915.6 | 1921.1 |
| 3500 | 1926.7 | 1932.2 | 1937.8 | 1943.3 | 1948.9 | 1954.4 | 1960.0 | 1965.6 | 1971.1 | 1976.7 |
| 3600 | 1982.2 | 1987.8 | 1993.3 | 1998.9 | 2004.4 | 2010.0 | 2015.6 | 2021.1 | 2026.7 | 2032.2 |
| 3700 | 2037.8 | 2043.3 | 2048.9 | 2054.4 | 2060.0 | 2065.6 | 2071.1 | 2076.7 | 2082.2 | 2087.8 |
| 3800 | 2093.3 | 2098.9 | 2104.4 | 2110.0 | 2115.6 | 2121.1 | 2126.7 | 2132.2 | 2137.8 | 2143.3 |
| 3900 | 2148.9 | 2154.4 | 2160.0 | 2165.6 | 2171.1 | 2176.7 | 2182.2 | 2187.8 | 2193.3 | 2198.9 |
| 4000 | 2204.4 | 2210.0 | 2215.6 | 2221.1 | 2226.7 | 2232.2 | 2237.8 | 2243.3 | 2248.9 | 2254.4 |
| 4100 | 2260.0 | 2265.6 | 2271.1 | 2276.7 | 2282.2 | 2287.8 | 2293.3 | 2298.9 | 2304.4 | 2310.0 |
| 4200 | 2315.6 | 2321.1 | 2326.7 | 2332.2 | 2337.8 | 2343.3 | 2348.9 | 2354.4 | 2360.0 | 2365.6 |
| 4300 | 2371.1 | 2376.7 | 2382.2 | 2387.8 | 2393.3 | 2398.9 | 2404.4 | 2410.0 | 2415.6 | 2421.1 |
| 4400 | 2426.7 | 2432.2 | 2437.8 | 2443.3 | 2448.9 | 2454.4 | 2460.0 | 2465.6 | 2471.1 | 2476.7 |
| 4500 | 2482.2 | 2487.8 | 2493.3 | 2498.9 | 2504.4 | 2510.0 | 2515.6 | 2521.1 | 2526.7 | 2532.2 |
| 4600 | 2537.8 | 2543.3 | 2548.9 | 2554.4 | 2560.0 | 2565.6 | 2571.1 | 2576.7 | 2582.2 | 2587.8 |
| 4700 | 2593.3 | 2598.9 | 2604.4 | 2610.0 | 2615.6 | 2621.1 | 2626.7 | 2632.2 | 2637.8 | 2643.3 |
| 4800 | 2648.9 | 2654.4 | 2660.0 | 2665.6 | 2671.1 | 2676.7 | 2682.2 | 2687.8 | 2693.3 | 2698.9 |
| 4900 | 2704.4 | 2710.0 | 2715.6 | 2721.1 | 2726.7 | 2732.2 | 2737.8 | 2743.3 | 2748.9 | 2754.4 |
| 5000 | 2760.0 | 2765.6 | 2771.1 | 2776.7 | 2782.2 | 2787.8 | 2793.3 | 2798.9 | 2804.4 | 2810.0 |
| 5100 | 2815.6 | 2821.1 | 2826.7 | 2832.2 | 2837.8 | 2843.3 | 2848.9 | 2854.4 | 2860.0 | 2865.6 |
| 5200 | 2871.1 | 2876.7 | 2882.2 | 2887.8 | 2893.3 | 2898.9 | 2904.4 | 2910.0 | 2915.6 | 2921.1 |
| 5300 | 2926.7 | 2932.2 | 2937.8 | 2943.3 | 2948.9 | 2954.4 | 2960.0 | 2965.6 | 2971.1 | 2976.7 |
| 5400 | 2982.2 | 2987.8 | 2993.3 | 2998.9 | 3004.4 | 3010.0 | 3015.6 | 3021.1 | 3026.7 | 3032.2 |
| 5500 | 3037.8 | 3043.3 | 3048.9 | 3054.4 | 3060.0 | 3065.6 | 3071.1 | 3076.7 | 3082.2 | 3087.8 |
| 5600 | 3093.3 | 3098.9 | 3104.4 | 3110.0 | 3115.6 | 3121.1 | 3126.7 | 3132.2 | 3137.8 | 3143.3 |
| 5700 | 3148.9 | 3154.4 | 3160.0 | 3165.6 | 3171.1 | 3176.7 | 3182.2 | 3187.8 | 3193.3 | 3198.9 |
| 5800 | 3204.4 | 3210.0 | 3215.6 | 3221.1 | 3226.7 | 3232.2 | 3237.8 | 3243.3 | 3248.9 | 3254.4 |
| 5900 | 3260.0 | 3265.6 | 3271.1 | 3276.7 | 3282.2 | 3287.8 | 3293.3 | 3298.9 | 3304.4 | 3310.0 |

## MEASURES AND WEIGHTS COMMONLY USED IN THE UNITED STATES AND GREAT BRITAIN

### Abbreviations

bbl. = barrel
bu. = bushel
chn. = chain
circm. = circumference
cir. in. = circular inch
cir. mil. = circular mil
cwt. = hundred weight
deg. = degree
drm. = drachm (dram)
dwt. = pennyweight
fath. = fathom
ft. = foot
fl. drm. = fluid drachm
fl. oz. = fluid ounce
furl. = furlong
gal. = gallon (U.S.)
grn. = grain
hghd. = hogshead
hr. = hour
Imp. = Imperial (bu., gal.)
in. = inch
lea = league
m. pace = military pace
min. = minute
naut. mi. = nautical mile
pnch = puncheon
quad. = quadrant
quin. = quintal
qr. = quarter
qt. = quart
scr. = scruple
sec. = second
sq. = square (in., ft., etc.)
stat. mi. = statute mile
trce. = tierce
yd. = yard

### Table 16.— Measures of Length

**Long Measure**

12 in. = 1 ft.
3 ft. = 1 yd.
5 1/2 yd. = 16 1/2 ft. = 1 rod, pole or perch
40 poles = 220 yd. = 1 furl.
8 furl. = 1760 yd. = 5280 ft. = 1 mile
4 in. = 1 hand
9 in. = 1 span
2 1/2 ft. = 1 m. pace.

**Land, Surveyors, or Gunter's Measure**

7.92 in. = 1 link
100 links = 66 ft. = 4 rods = 1 chn.
10 chn. = 220 yd. = 1 furl.
8 furl. = 80 chn. = 1 mile.

**Nautical Measure**

6080.26 ft. = 1.15156 stat. mi. = 1 naut. mi.
3 naut. mi. = 1 lea.
60 naut. mi. = 1 deg. (at equator)
6 ft. = 1 fath.
120 fath. = 1 cable
1 naut. mi. per hr. = 1 knot.

### Table 17.— Measures of Area

**Square Measure**

144 sq. in. = 183.35 cir. in. = 1 sq. ft.
9 sq. ft. = 1 sq. yd.
30 1/4 sq. yd. = 272 1/4 sq. ft. = 1 sq. rod
160 sq. rods = 10 chn. }
4840 sq. yd. = 43560 sq. ft. } = 1 acre
640 acres = 27,878,400 sq. ft. = 1 sq. mile
1 acre = $(208.71\ \text{ft.})^2$
1 cir. in. = { area of circle 1 in. diam. = 0.7854 sq. in.
1 cir. mil = { area of circle 0.001 in. diam. = $0.0_5 7854$ sq. in.
1,000,000 cir. mil = 1 cir. in.

**Board Measure**

144 cu. in. = 12 × 12 × 1 in. = 1 board ft.

The above definition of a board foot applies to rough green lumber. American practice uses a yard standard of 29/32 in. thick and an industrial standard of 7/8 in. thick for rough dry lumber; for dressed lumber the corresponding thicknesses are 25/32 in. and 13/16 in. For example, a rough dry piece 12 × 12 × 29/32 in. contains 1 board ft. Board ft. in round timber = $(1/4\, c \times d \times l)/144$. $c$ = mean circm., $d$ = diam., $l$ = length, all in ft.

### Table 18.—Measures of Weight

(The grain is the same in all systems.)

**Avoirdupois Weight**

16 drm. = 437.5 grn. = 1 oz.
16 oz. = 7000 grn. = 1 lb.
14 lb. = 1 stone
28 lb. = 2 stone = 1 qr.
100 lb. = 1 quin.
4 qr. = 112 lb. = 1 cwt.
20 cwt. = 2240 lb. = 1 long ton
2000 lb. = 1 short ton.

**Troy Weight (for gold and silver)**

24 grn. = 1 dwt.
20 dwt. = 480 grn. = 1 oz.
12 oz. = 5760 grn. = 1 lb.
1 Assay ton = 29,167 milligrams
= { Troy oz. per 2000 lb. ton avoirdupois

**Carat Weight (for precious stones)**

1 carat = 3.086 grn. = 0.200 gram

**Apothecaries' Weight**

20 grn. = 1 scr. ℈
3 scr. = 60 grn. = 1 drm. ʒ
8 drm. = 480 grn. = 1 oz. ℥
12 oz. = 5760 grn. = 1 lb.

### Table 19.—Measures of Volume

**Cubic Measure**

1728 cu. in. = 1 cu. ft.
27 cu. ft. = 1 cu. yd.
1 cord of wood = { 4 × 4 × 8 ft. = 128 cu. ft.
1 perch masonry = { 16 1/2 × 1 1/2 × 1 ft. = 23 3/4 cu. ft.

**Liquid Measure**

4 gills = 1 pint
2 pints = 1 qt.
4 qt. = 231 cu. in. = 8.3356 lb. $H_2O$
= 1 U. S. gal.
= 0.8327 Imp. gal.
1 Imp. gal. = 277.420 cu. in.
= 10 lb. $H_2O$
1.20094 U. S. gal. = 1 Imp. gal.
7.4805 U. S. gal. = 1 cu. ft.

**Old Liquid Measure**

31 1/2 gal. = 1 bbl.
42 gal. = 1 trce.
2 bbl. = 63 gal. = 1 hghd.
84 gal. = 2 trce. = 1 pchn.
2 hghd. = 126 gal. = 1 pipe or butt
2 pipes = 3 pchn. = 1 tun.

**Apothecaries' Fluid Measure**

60 minims = 1 fl. drm.
8 drm. = 1 fl. oz.
1 fl. oz. = 1/128 U. S. gal. = 1.8047 cu. in. = 456.3 grn. $H_2O$ at 39° F.
16 fl. oz. = 1 pint
1 fl. oz. (British) = 1.732 cu. in.
= 437.5 grn. $H_2O$ at 62° F.

**Dry Measure**

2 pints = 1 qt.
8 qts. = 1 peck
4 pecks = 2150.42 cu. in.
= 1 bu. (struck)
1 1/4 bu. (struck) = 1 bu. (heaped)
105 qts. (struck) = 7056 cu. in. = 1 bbl.
8 Imp. gal. = 2218.192 cu. in.
= 1 Imp. bu.
8 Imp. bu. = 1 qr.

**Shipping Measure**

100 cu. ft. = 1 Register ton *
40 cu. ft. = 31.143 U. S. bu.
= 1 shipping ton (U. S.) †
42 cu. ft. = 32.719 Imp. bu.
= 1 shipping ton (British) †

* For measurement of entire internal capacity. † For measurement of cargo.

**Miner's Inch**

A variable unit, defined by statute in different states as the quantity of water that will flow through an orifice of 1 sq. in. area, under a head ranging from 4 1/2 to 6 in. U. S. Reclamation Service defines the miner's inch as 1 cu. in. per sec. The law governing the locality in which the measurement is to be made should be consulted.

### Table 20.—Circular Measure

60 sec. = 1 min.
60 min. = 1 deg.
90 deg. = 1 quad.
360 deg. = circm.
57.2957795 deg. = 1 radian
= 57° 17′ 44.806″
1 radian = angle of arc equal to radius.

### Table 21.—Time

60 sec. = 1 min.
60 min. = 1 hr.
24 hr. = 1 day
7 days = 1 week
52 weeks = 1 year
1 year (exact) = 365 days, 5 hr., 48 min., 45.6 sec.

A year divisible by 4 is a leap year and contains 366 days, except centesimal years, of which only those divisible by 400 are leap years.

365.24734 mean solar days = 1 yr.
= 366.008515 sidereal days
24 hr. mean solar time
= 23 hr. 56 min., 49.9 sec. sidereal time
1 mean solar day
= 1 mean sidereal day + 3 min., 10.1 sec.

## THE METRIC SYSTEM

The fundamental standard of length in the metric system is the International Standard Meter, derived from the Metre des Archives, Paris. Its length is defined as the distance between two lines at 0° C. on a platinum-iridium bar deposited at the International Bureau of Weights and Measures near Paris. 1 meter = 39.37 in.

The fundamental standard of mass is the International Standard Kilogram, which is a mass of platinum-iridium, deposited at the International Bureau, whose weight *in vacuo* is the same as that of the Kilogram des Archives.

The fundamental unit of volume is the **liter,** which is defined as the volume of a kilogram of water at the temperature of maximum density, 4° C., under a pressure of 76 cm. of mercury. The liter originally was defined as a cubic decimeter, but is slightly greater. The generally accepted relation between the cubic decimeter and the liter is 1 liter = 1.000027 cu. decimeters.

The unit of force is the **dyne,** which is the force which acting for 1 second on a mass of one gram will produce a velocity of 1 centimeter per second. 1 dyne = (1/980.665) gram. 1,000,000 dynes = 1.020 kilograms force = 2.248 lb. force. 1 lb. force = 0.4536 kilogram force = 444,800 dynes.

The unit of heat is the **gram-calorie,** which is the quantity of heat required to raise 1 gram of water 1° C. See p. 3–17.

### Table 22.—Units of Metric Measures*

| Length | | | Area | | |
|---|---|---|---|---|---|
| Unit | Symbol | Value in Meters | Unit | Symbol | Value in Sq. Meters |
| Micron | $\mu$ | 0.000 001 | | | |
| Millimeter | mm. | 0.001 | Sq. millimeter | sq. mm. | 0.000 001 |
| Centimeter | cm. | 0.01 | Sq. centimeter | sq. cm. | 0.000 1 |
| Decimeter | dm. | 0.1 | Sq. decimeter | sq. dm. | 0.01 |
| Meter (unit) | m. | 1.0 | Sq. meter (centiare) | sq. m. | 1.0 |
| Dekameter | dkm. | 10.0 | Sq. dekameter (are) | sq. dkm. | 100.0 |
| Hectometer | hm. | 100.0 | Hectare | ha. | 10,000.0 |
| Kilometer | km. | 1,000.0 | Sq. kilometer | sq. km. | 1,000,000.0 |
| Myriameter | Mm. | 10,000.0 | | | |
| Megameter | .... | 1,000,000.0 | | | |

| Cubic Measure | | | Volume | | |
|---|---|---|---|---|---|
| Unit | Symbol | Value in Cubic Meters | Unit | Symbol | Value in Liters |
| Cubic kilometer | cu. km. | 1,000,000,000 | Milliliter | ml. | 0.001 |
| Cubic hectometer | cu. hm. | 1,000,000 | Centiliter | cl. | 0.01 |
| Cubic dekameter | cu. dkm. | 1,000 | Deciliter | dl. | 0.1 |
| Cubic meter | cu. m. | 1 | Liter (unit) | l. | 1.0 |
| Cubic decimeter | cu. dm. | 0.001 | Dekaliter | dkl. | 10.0 |
| Cubic centimeter | cu. cm. | 0.000 001 | Hectoliter | hl. | 100.0 |
| Cubic millimeter | cu. mm. | 0.000 000 001 | Kiloliter | kl. | 1,000.0 |
| Cubic micron | $\mu^3$ | $0.0_{17}1$ | | | |

| Weight | | | | | |
|---|---|---|---|---|---|
| Unit | Symbol | Value in Grams | Unit | Symbol | Value in Grams. |
| Microgram | $\gamma$ | 0.000 001 | Gram (unit) | g. | 1.0 |
| Milligram | mg. | 0.001 | Dekagram | dkg. | 10.0 |
| Centigram | cg. | 0.01 | Hectogram | hg. | 100.0 |
| Decigram | dg. | 0.1 | Kilogram | kg. | 1,000.0 |
| Gram (unit) | g. | 1.0 | Myriagram | Mg. | 10,000.0 |
| | | | Quintal | q. | 100,000.0 |
| | | | Ton | t. | 1,000,000.0 |

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* A subscript after a figure indicates the number of times it is repeated. Thus $0.0_38 = 0.0008$

### Table 23.—Metric Equivalents of Length *

| English to Metric | | | Log. | Metric to English | | | Log. |
|---|---|---|---|---|---|---|---|
| 1 inch | = 25.4 | mm. | 1.404834 | 1 mm. | = 0.03937 | in. | $\bar{2}.595165$ |
| | = 2.54 | cm. | 0.404834 | | = 0.003281 | ft. | $\bar{3}.515984$ |
| | = 0.0254 | m. | $\bar{2}.404834$ | | = 0.001094 | yd. | $\bar{3}.038863$ |
| 1 foot | = 304.800 | mm. | 2.484015 | 1 cm. | = 0.3937 | in. | $\bar{1}.595165$ |
| | = 30.480 | cm. | 1.484015 | | = 0.03281 | ft. | $\bar{2}.515984$ |
| | = 0.3048 | m. | $\bar{1}.484015$ | | = 0.01094 | yd. | $\bar{2}.038863$ |
| 1 yard | = 91.4402 | cm. | 1.961137 | 1 meter | = 39.37 | in. | 1.595162 |
| | = 0.9144 | m. | $\bar{1}.961137$ | | = 3.2808 | ft. | 0.515984 |
| | = $0.0_3914$ | km. | $\bar{4}.961137$ | | = 1.0936 | yd. | 0.038863 |
| 1 rod | = 502.9211 | cm. | 2.701500 | | = 0.1988 | rd. | $\bar{1}.298500$ |
| | = 5.0292 | m. | 0.701500 | | = 0.04971 | chain | $\bar{2}.696440$ |
| | = 0.005029 | km. | $\bar{3}.701500$ | | = $0.0_36214$ | mi. | $\bar{4}.793350$ |
| 1 chain | = 2011.68 | cm. | 3.303559 | 1 kilometer | = 3280.833 | ft. | 3.515984 |
| | = 20.1168 | m. | 1.303559 | | = 1093.611 | yd. | 3.038863 |
| | = 0.02012 | km. | $\bar{2}.303559$ | | = 198.838 | rods | 2.298500 |
| 1 mile | = 1609.344 | m. | 3.206650 | | = 49.7095 | chains | 1.696440 |
| | = 1.6093 | km. | 0.206650 | | = 0.6214 | mi. | $\bar{1}.793350$ |

### Table 24.—Metric Equivalents of Area *

| English to Metric | | | Log. | Metric to English | | | Log. |
|---|---|---|---|---|---|---|---|
| 1 cir. mil | = $0.0_35067$ | sq. mm. | $\bar{4}.704751$ | 1 sq. mm. | = 1,973.55 | cir. mils | 3.295249 |
| 1 sq. in. | = 645.163 | sq. mm. | 2.809669 | | = 0.001550 | sq. in. | $\bar{3}.190331$ |
| | = 6.4516 | sq. cm. | 0.809669 | | = $0.0_410764$ | sq. ft. | $\bar{5}.031968$ |
| | = $0.0_36452$ | sq. m. | $\bar{4}.809669$ | | = $0.0_51196$ | sq. yd. | $\bar{6}.077726$ |
| 1 sq. ft. | = 92,903.41 | sq. mm. | 4.968032 | 1 sq. cm. | = 0.1550 | sq. in. | $\bar{1}.190331$ |
| | = 929.0341 | sq. cm. | 2.968032 | | = 0.001076 | sq. ft. | $\bar{3}.031968$ |
| | = 0.0929 | sq. m. | $\bar{2}.968032$ | | = $0.0_31196$ | sq. yd. | $\bar{4}.077726$ |
| | = $0.0_5929$ | hectare | $\bar{6}.968032$ | 1 sq. m. | = 1,549.9969 | sq. in. | 3.190331 |
| | = $0.0_7929$ | sq. km. | $\bar{8}.968032$ | | = 10.7639 | sq. ft. | 1.031968 |
| 1 sq. yd. | = 836,130.74 | sq. mm. | 5.922274 | | = 1.1960 | sq. yd. | 0.077726 |
| | = 8,361.307 | sq. cm. | 3.922274 | | = 0.002471 | sq. chain | $\bar{3}.392882$ |
| | = 0.83613 | sq. m. | $\bar{1}.922274$ | | = $0.0_32471$ | acre | $\bar{4}.392882$ |
| | = $0.0_4836$ | hectare | $\bar{5}.922274$ | | = $0.0_63861$ | sq. mi. | $\bar{7}.586700$ |
| | = $0.0_6836$ | sq. km. | $\bar{7}.922274$ | 1 hectare | = 107,638.7 | sq. ft. | 5.031968 |
| 1 sq. chain | = 404.686 | sq. m. | 2.607118 | | = 11,959.85 | sq. yd. | 4.077726 |
| | = 0.04047 | hectare | $\bar{2}.607118$ | | = 24.710 | sq. chain | 1.392882 |
| | = $0.0_34047$ | sq. km. | $\bar{4}.607118$ | | = 2.4710 | acres | 0.392882 |
| 1 acre | = 4,046.86 | sq. m. | 3.607118 | | = 0.003861 | sq. mi. | $\bar{3}.586700$ |
| | = 0.4047 | hectare | $\bar{1}.607118$ | 1 sq. km. | = 10,763,867.36 | sq. ft. | 7.031968 |
| | = 0.004047 | sq. km. | $\bar{3}.607118$ | | = 1,195,985.26 | sq. yd. | 6.077726 |
| 1 sq. mile | = 2,589,998 | sq. m. | 6.413300 | | = 2,471.050 | sq. chains | 3.392882 |
| | = 259.0 | hectare | 2.413300 | | = 247.1045 | acres | 2.392882 |
| | = 2.590 | sq. km. | 0.413300 | | = 0.3861 | sq. mi. | $\bar{1}.586700$ |

### Table 25.—Metric Equivalents of Weight or Mass *

| English to Metric | | | Log. | Metric to English | | | Log. |
|---|---|---|---|---|---|---|---|
| 1 grain | = 64 797 | mg. | 1.811568 | 1 milligram | = 0.01543 | grain | $\bar{2}.188433$ |
| | = 0.0648 | gm. | $\bar{2}.811568$ | | = $0.0_43215$ | oz. Troy | $\bar{5}.507191$ |
| 1 oz. Troy or | | | | | = $0.0_43527$ | oz. avoir. | $\bar{5}.547454$ |
| apothecary | = 31,103.5 | mg. | 4.492809 | 1 gram | = 15.4324 | grains | 1.188433 |
| | = 31.103 | gm. | 1.492809 | | = 0.03215 | oz. Troy | $\bar{2}.507191$ |
| | = 0.03110 | kg. | $\bar{2}.492809$ | | = 0.03527 | oz. avoir. | $\bar{2}.547454$ |
| 1 oz. avoirdupois | = 28,349.5 | mg. | 4.452546 | | = $0.0_22679$ | lb. Troy | $\bar{3}.428010$ |
| | = 28.3495 | gm. | 1.452546 | | = $0.0_22205$ | lb. avoir. | $\bar{3}.343334$ |
| | = 0.02835 | kg. | $\bar{2}.452546$ | 1 kilogram | = 32.1508 | oz. Troy | 1.507191 |
| 1 pound Troy or | | | | | = 35.2740 | oz. avoir. | 1.547454 |
| apothecary | = 373.2417 | gm. | 2.571990 | | = 2.6792 | lb. Troy | 0.428010 |
| | = 0.3732 | kg. | $\bar{1}.571990$ | | = 2.2046 | lb. avoir. | 0.343334 |
| 1 pound avoirdupois | = 453.5925 | gm. | 2.656666 | | = $0.0_21102$ | short ton | $\bar{3}.043304$ |
| | = 0.4536 | kg. | $\bar{1}.656666$ | | = $0.0_39842$ | long ton | $\bar{4}.993086$ |
| | = $0.0_34536$ | metric ton | $\bar{4}.656666$ | 1 metric ton | = 2204.62 | lb. avoir. | 3.343334 |
| 1 short ton | = 907.2 | kg. | 2.957696 | | = 1.1023 | short tons | 0.043304 |
| | = 0.9072 | metric ton | $\bar{1}.957696$ | | = 0.9842 | long ton | $\bar{1}.993086$ |
| 1 long ton | = 1,016.06 | kg. | 3.006914 | | | | |
| | = 1.0161 | metric ton | 0.006914 | | | | |

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* A subscript after a figure indicates the number of times it is repeated. Thus $0.0_38 = 0.0008$

## Table 26.—Metric Equivalents of Capacity or Volume*

| English to Metric | | Log. | Metric to English | | Log. |
|---|---|---|---|---|---|
| 1 cu. in. | = 16,387.17 cu. mm. | 4.214504 | 1 cu. mm. | = $0.0_46102$ cu. in. | $\bar{5}.785496$ |
| | = 16.3872 cu. cm. | 1.214504 | | = $0.0_32705$ fluid dr. | $\bar{4}.432185$ |
| | = 0.016387 l. | $\bar{2}.214504$ | | = $0.0_43381$ fluid oz. | $\bar{5}.529094$ |
| | = $0.0_316387$ hl. | $\bar{4}.214504$ | 1 cu. cm. | = 0.06102 cu. in. | $\bar{2}.785496$ |
| | = $0.0_416387$ cu. m. | $\bar{5}.214504$ | | = $0.0_43531$ cu. ft. | $\bar{5}.547951$ |
| 1 cu. ft. | = 28,317.08 cu. cm. | 4.452049 | | = $0.0_51308$ cu. yd. | $\bar{6}.116589$ |
| | = 28.3169 l. | 1.452049 | | = $0.0_42838$ bushel | $\bar{5}.452972$ |
| | = 0.2832 hl. | $\bar{1}.452049$ | | = 0.2705 fluid dr. | $\bar{1}.432185$ |
| | = 0.02832 cu.m. | $\bar{2}.452049$ | | = 0.03381 fluid oz. | $\bar{2}.529094$ |
| 1 cu. yd. | = 764,559.5 cu. cm. | 5.883411 | | = 0.001057 quart | $\bar{3}.023944$ |
| | = 764.5595 l. | 2.883411 | | = $0.0_32642$ gallon | $\bar{4}.421884$ |
| | = 7.6456 hl. | 0.883411 | 1 liter | = 61.02398 cu. in. | 1.785496 |
| | = 0.7646 cu. m. | $\bar{1}.883411$ | | = 0.035313 cu. ft. | $\bar{2}.547951$ |
| 1 bushel | = 35,239.3 cu. cm. | 4.547027 | | = 0.0013079 cu. yd. | $\bar{3}.116589$ |
| | = 35.2393 l. | 1.547027 | | = 0.028377 bushel | $\bar{2}.452972$ |
| | = 0.3524 hl. | $\bar{1}.547027$ | | = 1.0567 quart | 0.023944 |
| | = 0.03524 cu. m. | $\bar{2}.547027$ | | = 0.2642 gallon | $\bar{1}.421884$ |
| 1 fluid drachm | = 3,696.7 cu. mm. | 3.567815 | 1 hectoliter | = 6,102.398 cu. in. | 3.785496 |
| | = 3.6967 cu. cm. | 0.567815 | | = 3.5313 cu. ft. | 0.547951 |
| 1 fluid oz. | = 29,573.7 cu. mm. | 4.470906 | | = 0.13079 cu. yd. | $\bar{1}.116589$ |
| | = 29.5737 cu. cm. | 1.470906 | | = 2.8377 bushels | 0.452972 |
| 1 quart | = 946.359 cu. cm. | 2.976056 | 1 cu. meter | = 61,023.38 cu. in. | 4.785496 |
| | = 0.9464 l. | $\bar{1}.976056$ | | = 35.3133 cu. ft. | 1.547951 |
| | = $0.0_3946$ cu. m. | $\bar{4}.976056$ | | = 1.3079 cu. yd. | 0.116589 |
| 1 U. S. gallon | = 3,785.43 cu. cm. | 3.578116 | | = 28.3773 bushels | 1.452972 |
| | = 3.7854 l. | 0.578116 | | = 1,056.682 quarts | 3.023944 |
| | = 0.003785 cu. m. | $\bar{3}.578116$ | | = 264.170 gallons | 2.421884 |

## Table 27.—Metric Equivalents of Density*

| English to Metric | | Log. | Metric to English | | Log. |
|---|---|---|---|---|---|
| 1 lb. per cu. in. | = 27.680 gm. per cu. cm. | 1.442162 | 1 gm. per cu. cm. | = 0.03613 lb. per cu. in. | 2.557838 |
| | = 27,679.7 kg. per cu. m. | 4.442162 | | = 62.430 lb. per cu. ft. | 1.795381 |
| 1 lb. per cu. ft. | = 0.01602 gm. per cu. cm. | $\bar{2}.204619$ | | = 8.3454 lb. per U. S. gal. | 0.921450 |
| | = 16.0183 kg. per cu. m. | 1.204619 | 1 kg. per cu. m. | = $0.0_43613$ lb. per cu. in. | $\bar{5}.557838$ |
| | = 0.01602 metric ton per cu. m. | $\bar{2}.204619$ | | = 0.062430 lb. per cu. ft. | $\bar{2}.795381$ |
| 1 lb. per cu. yd. | = 0.5933 kg. per cu. m. | $\bar{1}.773254$ | | = 1.6856 lb. per cu. yd. | 0.226746 |
| | = $0.0_35933$ metric tons per cu. m. | $\bar{4}.773254$ | | = $0.0_28345$ lb. per U. S. gal. | $\bar{3}.921450$ |
| 1 lb. per U. S. gal. | = 0.1198 gm. per cu. cm. | $\bar{1}.078550$ | 1 metric ton per cu. m | = 62.4286 lb. per cu. ft. | 1.795381 |
| | = 119.826 kg. per cu. m. | 2.078550 | | = 1685.487 lb. per cu. yd. | 3.226746 |
| 1 short ton per cu. yd. | = 1.1865 metric tons per cu. m. | 0.074285 | | = 0.8428 short ton per cu. yd. | $\bar{1}.925715$ |
| 1 long ton per cu. yd. | = 1.3289 metric tons per cu. m. | 0.123502 | | = 0.7525 long ton per cu. yd. | $\bar{1}.876498$ |

## Table 28.—Metric Equivalents of Velocity*

| English to Metric | | Log. | Metric to English | | Log. |
|---|---|---|---|---|---|
| 1 in. per sec., | = 2.54001 cm. per sec. | 0.404834 | 1 cm. per sec. | = 0.3937 in. per sec. | $\bar{1}.595165$ |
| | = 0.0254 m. per sec. | $\bar{2}.404834$ | | = 0.03281 ft. per sec. | $\bar{2}.515984$ |
| | = 1.5240 m. per min. | 0.182985 | | = 1.9685 ft. per min. | 0.294136 |
| 1 ft. per sec. | = 30.4801 cm. per sec. | 1.484015 | | = 0.02237 mi. per hr. | $\bar{2}.349653$ |
| | = 0.3048 m. per sec. | $\bar{1}.484015$ | | = 0.01943 knot | $\bar{2}.288367$ |
| | = 18.2880 m. per min. | 1.262166 | 1 m. per sec. | = 39.37 in. per sec. | 1.595165 |
| | = 1.0973 km. per hr. | 0.040318 | | = 3.2808 ft. per sec. | 0.515984 |
| 1 ft. per min. | = 0.5080 cm. per sec. | $\bar{1}.705864$ | | = 196.8500 ft. per min. | 2.294136 |
| | = 0.00508 m. per sec. | $\bar{3}.705864$ | | = 2.2369 mi. per hr. | 0.349653 |
| | = 0.3048 m. per min. | $\bar{1}.484015$ | | = 1.9426 knots | 0.288367 |
| | = 0.018288 km. per hr. | $\bar{2}.262166$ | 1 m. per min. | = 0.6562 in. per sec. | $\bar{1}.817014$ |
| 1 mile per hr. | = 44.704 cm. per sec. | 1.650348 | | = 0.05468 ft. per sec. | $\bar{2}.737833$ |
| | = 0.4470 m. per sec. | $\bar{1}.650348$ | | = 3.2808 ft. per min. | 0.515984 |
| | = 26.8222 m. per min. | 1.428495 | | = 0.03728 mi. per hr. | $\bar{2}.571503$ |
| | = 1.6093 km. per hr. | 0.206650 | | = 0.03238 knot | $\bar{2}.510218$ |
| 1 knot | = 51.4791 cm. per sec. | 1.711631 | 1 km. per hr. | = 0.9113 ft. per sec. | $\bar{1}.959681$ |
| | = 0.5148 m. per sec. | $\bar{1}.711631$ | | = 54.6806 ft. per min. | 1.737833 |
| | = 30.8875 m. per min. | 1.489782 | | = 0.62138 mi. per hr. | $\bar{1}.793357$ |
| | = 1.8532 km. per hr. | 0.267933 | | = 0.5396 knot | $\bar{1}.732072$ |

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* A subscript after a figure indicates the number of times it is repeated. Thus $0.0_38 = 0.0008$

## Table 29.—Metric Equivalents of Pressure*

| English to Metric | | Log. | Metric to English | | Log. |
|---|---|---|---|---|---|
| 1 lb. per sq. in. | = 0.7031 gm. per sq. mm. | $\bar{1}.846996$ | 1 gm. per sq. mm. | = 1.4223 lb. per sq. in. | 0.153004 |
| | = 70.3066 gm. per sq. cm. | 2.846996 | | = 204.8170 lb. per sq. ft. | 2.311366 |
| | = 0.07031 kg. per sq. cm. | $\bar{2}.846996$ | 1 gm. per sq. cm. | = 0.01422 lb. per sq. in. | $\bar{2}.153004$ |
| | = 0.7031 metric ton per sq. m. | $\bar{1}.846996$ | | = 2.0482 lb. per sq. ft. | 0.311366 |
| | = 0.07031 metric atmosphere † | $\bar{2}.846996$ | 1 kg. per sq. cm. | = 14.2234 lb. per sq. in. | 1.153004 |
| 1 lb. per sq. ft. | = 0.004882 gm. per sq. mm. | $\bar{3}.688598$ | | = 2048.1696 lb. per sq. ft. | 3.311366 |
| | = 0.4882 gm. per sq. cm. | $\bar{1}.688598$ | 1 metric ton per sq. m. | = 1.4223 lb. per sq. in. | 0.153004 |
| | = $0.0_34882$ kg. per sq. cm. | $\bar{4}.688598$ | | = 204.8170 lb. per sq. ft. | 2.311366 |
| | = 0.004882 metric ton per sq. m. | $\bar{3}.688598$ | | = 0.1024 ton per sq. ft. § | $\bar{1}.010342$ |
| | = $0.0_34882$ metric atmosphere † | $\bar{4}.688598$ | 1 metric atmosphere † | = 14.2234 lb. per sq. in. | 1.153004 |
| 1 ton per sq. ft. † | = 9.7648 metric tons per sq. m. | 0.989663 | | = 2048.1696 lb. per sq. ft. | 3.311366 |
| | = 0.9765 metric atmosphere † | $\bar{1}.989663$ | | = 1.0241 tons per sq. ft. | 0.010342 |
| 1 atmosphere ‡ | = 1.0335 metric atmospheres † | 0.014310 | | = 0.9676 atmosphere ‡ | $\bar{1}.985696$ |

† 1 metric atmosphere = 1 kg. per sq. cm. ‡ 1 atmosphere = 14.7 lb. per sq. in. § 1 ton = 2000 lb.

## Table 30.— Metric Equivalents of Force *

| English to Metric | | Log. | Metric to English | | Log. |
|---|---|---|---|---|---|
| 1 lb. | = 444,800 dynes | 5.648165 | 1 dyne | = $0.0_52248$ lb. | $\bar{6}.351796$ |
| | = 453.6 grams | 2.656673 | | = $0.0_47233$ poundal | $\bar{5}.859318$ |
| | = 0.04448 joule per cm. | $\bar{2}.648165$ | | = $0.0_21020$ gram | $\bar{3}.008600$ |
| | = 0.4536 kg. | $\bar{1}.656673$ | | = $0.0_51020$ kg. | $\bar{6}.008600$ |
| | = 32.17 poundals | 1.507451 | | = $0.0_61$ joule per cm. | $\bar{7}.000000$ |
| 1 poundal | = 13,826 dynes | 4.140696 | 1 gram | = $0.0_22205$ lb. | $\bar{3}.343409$ |
| | = 14.10 grams | 1.149219 | | = 0.07093 poundal | $\bar{2}.850830$ |
| | = $0.0_213826$ joule per cm. | $\bar{3}.140696$ | | = 980.7 dynes | 2.991536 |
| | = 0.01410 kg. | $\bar{2}.149219$ | | = 0.001 kg. | $\bar{3}.000000$ |
| | = 0.03108 lb. | $\bar{2}.492481$ | | = $0.0_49807$ joule per cm. | $\bar{5}.991536$ |
| | | | 1 joule per cm. | = 22.48 lb. | 1.351796 |
| | | | | = 723.3 poundals | 2.859318 |
| | | | | = 10,000,000 dynes | 7.000000 |
| | | | | = 10,200 grams | 4.008600 |
| | | | | = 10.20 kg. | 1.008600 |
| | | | 1 kg. | = 2.205 lb. | 0.343409 |
| | | | | = 70.93 poundals | 1.850830 |
| | | | | = 980,700 dynes | 5.991536 |
| | | | | = 1000 grams | 3.000000 |
| | | | | = 0.09807 joule per cm. | $\bar{2}.991536$ |

## Table 31.—Metric Equivalents of Power, Work and Energy*

| English to Metric | | Log | Metric to English | | Log. |
|---|---|---|---|---|---|
| 1 ft.-lb. | = 0.1383 kg.-m. | $\bar{1}.140683$ | 1 kilogram-meter | = 7.2331 ft.-lb. | 0.859318 |
| | = $0.0_63766$ kw.-hr. | $\bar{7}.575880$ | | = $0.0_53653$ Hp.-hr. | $\bar{6}.562651$ |
| | = $0.0_65121$ cheval-vapeur-hr. ‡ | $\bar{7}.709698$ | | = $0.0_29303$ B.t.u. | $\bar{3}.968622$ |
| | = $0.0_33241$ kg.-calorie | $\bar{4}.510660$ | 1 kilogram-meter per sec. | = 7.2331 ft.-lb. per sec. | 0.859318 |
| 1 ft.-lb. per sec. | = 0.1383 kg.-m. per sec. | $\bar{1}.140683$ | | = 0.01315 Hp. | $\bar{2}.118929$ |
| | = $0.0_21383$ poncelet † | $\bar{3}.140683$ | 1 kilogram-meter per min. | = 7.2331 ft.-lb. per min. | 0.859318 |
| | = $0.0_21843$ cheval-vapeur ‡ | $\bar{3}.265619$ | | = $0.0_32180$ Hp. | $\bar{4}.340809$ |
| | = $0.0_33241$ kg.-cal. per sec. | $\bar{4}.510660$ | 1 kilowatt-hour | = 2,655,180 ft.-lb. | 6.424082 |
| 1 ft.-lb. per min. | = 0.1383 kg.-m. per min. | $\bar{1}.140683$ | | = 1.3341 Hp.-hr. | 0.125189 |
| | = $0.0_21383$ poncelet † | $\bar{3}.140683$ | | = 3411.5 B.t.u. | 3.532946 |
| | = $0.0_43072$ cheval-vapeur ‡ | $\bar{5}.487469$ | 1 cheval-vapeur ‡ | = 542.48 ft.-lb. per sec. | 2.734384 |
| 1 Hp. | = 76.0398 kg.-m. per sec. | 1.881040 | | = 32,548.57 ft.-lb. per min. | 4.512532 |
| | = 4562.384 kg.-m. per min. | 3.659191 | | = 0.9863 Hp. | $\bar{1}.994018$ |
| | = 0.7604 poncelet † | $\bar{1}.881040$ | 1 cheval-vapeur-hour | = 1,952,914 ft.-lb. | 6.290683 |
| | = 1.01390 cheval-vapeur ‡ | 0.016616 | | = 2513.56 B.t.u. | 3.400294 |
| 1 Hp.-hr. | = 273,742.9 kg.-m. | 5.437436 | 1 Poncelet † | = 723.31 ft.-lb. per sec. | 2.859318 |
| | = 0.7456 kw.-hr. | $\bar{1}.872516$ | | = 43,398.10 ft.-lb. per min. | 4.637471 |
| | = 641.6895 kg.-cal. | 2.807325 | | = 1.3151 Hp. | 0.118929 |
| 1 B.t.u. | = 0.2520 kg.-cal. | $\bar{1}.401401$ | 1 kilogram-calorie | = 3085.60 ft.-lb. | 3.489360 |
| | = 107.4925 kg.-m. | 2.031123 | | = $0.0_21558$ Hp.-hr. | $\bar{3}.192675$ |
| | = $0.0_32931$ kw.-hr. | $\bar{4}.467016$ | | = 3.9686 B.t.u. | 0.598367 |
| | = $0.0_33978$ cheval-vapeur-hr. ‡ | $\bar{4}.599711$ | 1 kilogram-calorie per sec. | = 3085.60 ft.-lb. per sec. | 3.489360 |
| | | | | = 5.610 Hp. | 0.748982 |

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* A subscript after a figure indicates the number of times it is repeated. Thus $0.0_38 = 0.0008$.
† 1 poncelet = 100 kg.-m. per sec. ‡ 1 cheval-vapeur = 75 kg.-m. per sec. = 1 metric horsepower.

# INDEX

**Note: The double numbers refer to both section and page numbers. For example, 13–09 indicates page 09 of Section 13.**

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